SCIENTIA CUM INDUSTRIA, V. 6, N. 1, PP. 29 — 37, 2018

Comparison between vertical acceleration data from acquired signals and multibody model for an off-road vehicle

Cristian Padilha Fontoura*, Alan Alves dos Passos*, Deives Roberto Bareta*, Elton João Zanol*

Abstract

SAE Mini Baja competitions require efforts in developing a reliable vehicle project that enables their teams to manage time and resources wisely. Vehicle simulations are one the best ways to deal with these conditions and prevent failure during a test. This work outlines the methodology that was carried out for validating the multibody dynamics model of a Mini Baja vehicle through vertical acceleration data acquisition. The data was acquired with the vehicle in different sets of obstacles, based on those seen in previously held competitions. Simulation was done through ADAMS/Car, with the vehicle's multibody model being simulated in different three-dimensional roads, counterpart to those where data acquisition took place. Simulation data, when compared to acquired acceleration signals for most of the obstacles, exhibited equivalence. Additional data computation revealed that the spectra in the frequency domain presented most severe loads concentrated between 0 and 20 Hz, incoming mostly from road unevenness. Gathering such data, by the presented approach can assist future analyses and guide the Baja Team in defining an improved project by predicting its dynamic behavior.

Keywords

Multibody vehicle simulation, acceleration data acquisition, multibody model validation

I. Introduction

Road profile characterization is an important analytical tool that engineering has been using in recent decades, especially with the advent and consolidation of software capable of performing simulations with high degrees of reliability. Profile measurement is a common method for evaluation of roads and highways and is often done to obtain the International Roughness Index (IRI) [1], which is defined as the mathematical property of a two-dimensional lane profile; the profile represents a longitudinal slice of the lane showing how the elevations vary along the longitudinal distance along a path travelled [2]. The influence of the road surface is reflected in costs to society and the environment, such as fuel consumption, wear and tear on vehicle components, and the safety and comfort of occupants of a vehicle [3-5]. Measurements of such profiles, though, have also a high cost and are often a time-consuming approach, as suggested by [6]. A computational model helps linking the surface roughness profile of a track with the various impacts on a vehicle.

The knowledge of the irregularities imposed by the pavement on the vehicle has been applied directly in the automotive industry, especially with dynamics simulation as such method is allied with agility in the execution of projects [7]. The use of simulations is a prevailing method for the assessment of vehicle projects; however, prediction of loading and reliable structural analysis can only be achieved if there is a faithful model of the vehicle and accurate road data, which fully represent the disturbance absorbed by the car [8]. As for Mini Baja competitions, these are led by engineering students and require time and financial support. Consistency in the project in this case is one of the most important factors, to prevent failure during competition and within the deadlines.

A difficulty seen in previous studies is the use of such tools, signal filtering and multibody approach, for off-road vehicles. Taking all of this into account, the objective of this work is to compare acceleration data of experimental nature with acceleration data resulting from multibody dynamics simulation of the vehicle-road interaction, so that the multibody model can validate data acquisition and vice-versa, with a similar methodology adopted by [9-13].

To reach the main goal, which is validating the multibody approach, the following actions were taken:

- Definition of the obstacles to be characterized; as seen in previous competitions;
- Instrumentation of the vehicle;
- Collection of experimental acceleration data;
- Filtering of the obtained data;
- Simulation of the vehicle in multibody dynamics software and comparison of postprocessing data with experimental signals.

II. Materials and Methods

A – Model description

So that the multibody model can be faithful to the real vehicle, the latter's information must be well known. The car had its sprung and unsprung masses weighed, the tire and springs tested, and a study on the vehicle's center of gravity (hereafter referred as CG) was undertaken. A simplified

* Universidade de Caxias do Sul, Caxias do Sul, RS
E-mails: cpfontoura@ucs.br, aapassos@ucs.br, drbareta@ucs.br, ejzanol@ucs.br

Data de envio: 02/01/2018
Data de aceite: 09/02/2018

http://dx.doi.org/10.18226/23185279.v6iss1p29

model was chosen, to make this an easily assessed analysis that can be undertaken by students involved in the project. The model comprises of both suspensions and a concentrated mass body where the CG is.

An important variable in automotive studies is the characterization of tires. The behavior of an impact-absorbing tire directly influences how the results can be interpreted. The tire vertical damping coefficient was neglected, as this is a suitable consideration [14]. In order to obtain the stiffness of the tires, a compression load vertically applied to the tires was performed on a EMIC DL200000 testing machine, with reading made through a Tesc software. The stiffness values were obtained from the inclination of force-deformation curves and based on mass distribution at each tire [15-17]. Measurements were made with the two types of tire that are used by the Mini Baja team. Figure 1 shows a photograph of the equipment. The test was also done in different tire inflation pressures. The shock absorber springs were characterized in a similar fashion.

(a) (b)

Figure 1 – Compression test for (a) rear tyre and (b) front tyre

The tire inflation pressure used during the data acquisition was 12 psi. The CG was studied through SolidWorks, with a full vehicle CAD model, as seen in Figure 2.

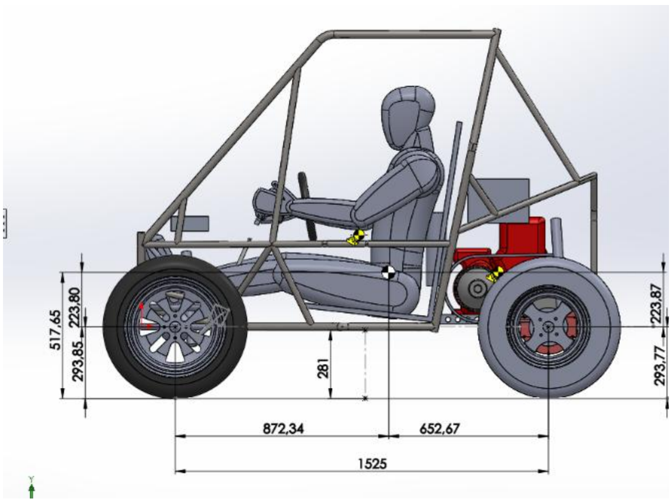


Figure 2 – Study of the car's CG, through CAD model (lateral view)

Table 1 brings the most essential specifications needed to create the vehicle multibody model (MBM), such as both tire stiffnesses, geometrical properties and masses involved.

Table 1 – Specification of the vehicle

Description	Value
Total mass of vehicle, with pilot	254 kg
Sprung mass	180 kg
Mass distribution (front axle)	43%
Basewheel	1. 25 m
CG's height	0.518 m
Distance CG – front axle	0.872 m
Distance CG – rear axle	0.653 m
Front tire stiffness	69 kN/m
Rear tire stiffness	117 kN/m

MSC ADAMS/Car is a multibody dynamics software well used in performing different types of studies applied to automotive engineering. In the ADAMS/Car environment, specific tools for automotive components and subsystems are available. The suspension models were based on models from the software's library. These models comprise all the joints and links in the suspension, as well as springs, dampers and tires. The mass distribution between the axles, the tire stiffness, the distance between the axles and the masses involved are some of the specifications that are also input into the model. For this vehicle, the front suspension has a double wishbone configuration, while the rear suspension has a multi-link configuration.

The information on the obstacles studied was added in the ADAMS virtual environment through the Road-Profile Builder tool. By importing vectors containing horizontal position values and vertical displacements, it is possible to obtain a three-dimensional profile of the track in question. The coefficient of friction of the tracks was considered 0.7 for the asphalt obstacles and 0.6 for the gravel and earth road obstacles. Figure 3 shows the modelled slope, with the Baja model inserted in it, ready to start running the simulation.

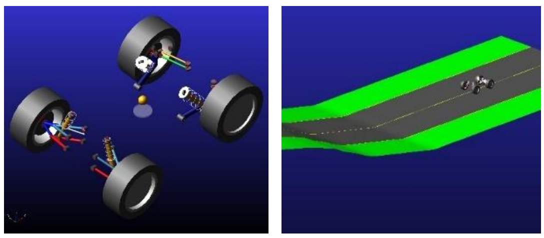

(a) (b)

Figure 3 – Full-vehicle model (a) in a built profile (b)

With the modelling of the tracks and the vehicle in question, the Smart Driver tool, within Full-Vehicle Analysis, can be used to run the vehicle simulation on the chosen tracks. The speed of the vehicle is configured, as well as the simulation time and the number of iterations desired. Another method used for the simulation of the suspensions alone is the Test Rig, which is composed of vertical actuators that impose movement to the subsystem of the suspension. Figure 4 shows the suspensions supported on the actuators for this type of simulation.

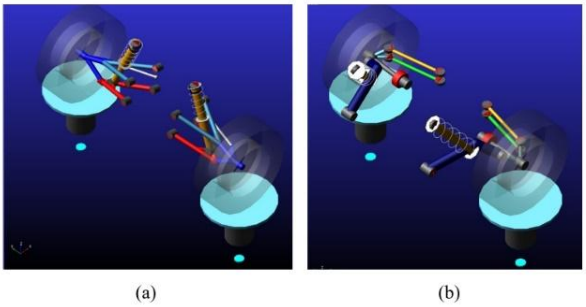


Figure 4 – Test Rig on front suspension (a) and rear suspension (b)

B – Experimental setup

The experimental setup and data filtering methodology adopted has been used in the industry and in research and was adapted onto the needs of this study.

To obtain reliable and useful data, uniaxial accelerometers (608A11, IMC Sensors) were coupled at three points in the vehicle. As a form of simplification, only one side of the vehicle was instrumented. One of the accelerometers was coupled near the vehicle's CG, which collected the sprung mass accelerations. As the CG is in the vicinities the pilot's body, the accelerometer was attached to a side bar, disregarding the transversal position and considering its vertical and longitudinal position. Two other accelerometers were installed in the knuckles at the center of the wheel, one in the front axle and the other in the rear axle, for the accelerations imposed on the unsprung masses of each suspension. Several related works have made or suggested the use of this type of experimental configuration, such as [18, 19]. Figure 5 shows the location of accelerometers fitted on the vehicle wheels and the CG. The accelerometers were calibrated with a 394C06 PCB Piezotronics handheld shaker of 1g of acceleration and working frequency of 159.2 Hz.

Figure 5 – Fitting of the accelerometer in the vehicle

C – Data filtering

The acceleration data were collected by mobile signal amplifiers and saved directly to the recorder for further filtering, being both HBM equipment. The data recorder was a QuantumX CX22B-W model and the amplifier was a QuantumX MX440B. As each channel represents a point on the vehicle, the data was separated and underwent a smoothing process to remove noise and anomalies through a Savitzky-Golay filter. However, to obtain data as a function of displacements of the unsprung mass, the double integration was performed with raw discrete data, since that data passed through MATLAB's Butterworth filter.

For the double integration, the motion equation of acceleration for unsprung mass of a quarter-car model (Eq. 1) was considered.

$$m_u\ddot{x}_u = [-k_s\,(x_s - x_u) + c_s(\dot{x}_s - \dot{x}_u) - k_u\,(x_u - y)] \qquad (1)$$

In which:

m_u unsprung mass
$\ddot{x}_u$ unsprung mass acceleration
k_s sprung mass stiffness

x_s	sprung mass displacement
c_s	sprung mass damping coefficient
$\dot{x}_s$	sprung mass velocity
$\dot{x}_u$	unsprung mass velocity
k_u	tire stiffness
x_u	unsprung mass
y	random signal input

As the acceleration $\ddot{x}_u$ is the second derivative of the displacement and has a direct relation with a given random signal y (the road, in this case), a double integration returns displacement values.

The data acquisition was performed separately in each obstacle, all with triplicate measurements, to ensure the consistency of the results, with the vehicle passing over them at different speeds. The acquisition rate was 600 Hz. The recording equipment was coupled to the rear of the vehicle, near the tank, with a bracket specially made for supporting it.

Discrete Fourier Transform (DFT), as seen in Equation 2, was used to convert data into the frequency domain:

$$F_n = \sum_{i=0}^{N-1} \ddot{x}_i e^{-\frac{2\pi j}{N} ni} \quad (2)$$

The route obstacles, in which the data was collected, were partially chosen based on Mini Baja competitions of previous years. They are seen in Figure 6.

Speed bump with a maximum height of 11 cm

Equally distanced raised pavement markers

Flat portion of gravel road

Hillside with 12 degrees of inclination

Figure 6 – Obstacles chosen for data acquisition

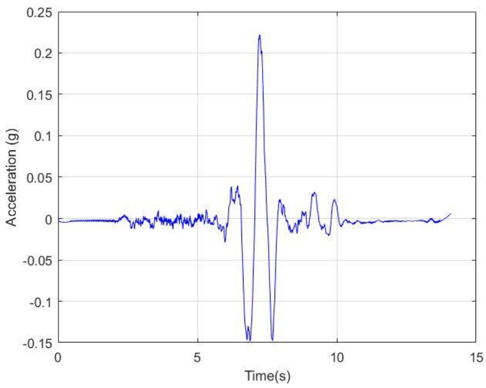
III. Results and Discussion

A – Comparison of Acceleration Data

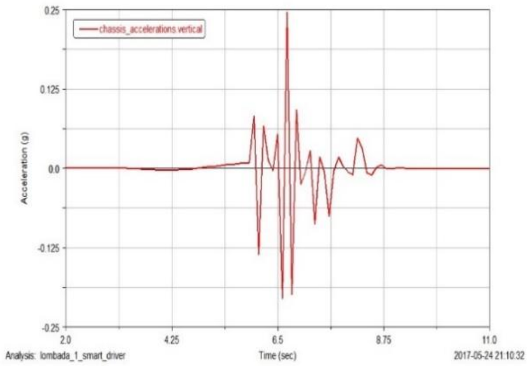
With the data obtained from the accelerometer coupled near the vehicle's CG, actual acceleration data can be compared with the CG'S virtual response data for the full vehicle multibody analysis. The data of the accelerometers in each of the suspension's wheel knuckle was used to compare the simulation data in the Test Rig mode, where simulation isolates the suspension. For full vehicle analyses, results are given in function of g. For isolated suspensions, results are in m/s².

A1 – Crossing a speed bump

The vehicle's CG acceleration signals collected during speed bump crossing are seen in Figure 7a, with a maximum peak of approximately 0.225 g. There are also two negative peaks of approximately -0.15 g. The results of the simulation, shown in Figure 7b, returned maximum peak values of approximately 0.25 g at the highest positive peak and for negative peaks, values are close to -0.15 g. The analysis of speed bump crossing showed it to be one that demands most of the dampers as seen similarly in [20]. This was clear to notice when unsprung mass accelerations are also compared with the CG's signals, showing a great difference in absolute values.



(a)

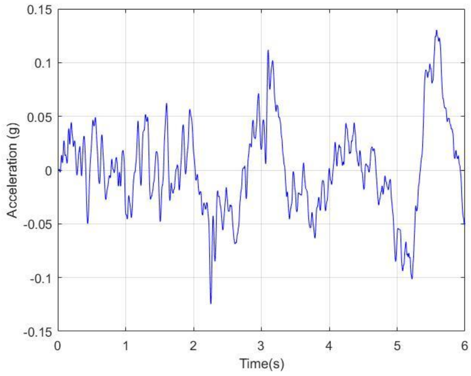


(b)

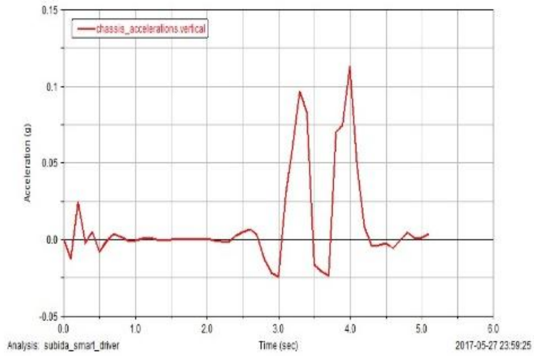
Figure 7 – Acceleration signal acquired on a speed bump (a) and MBM result (b)

A2 – Transition from flat to slope

In the transition from flat road to slope, the experimental values, seen in Figure 8a, showed peaks of 0.1 g and 0.15 g of acceleration, coming from mean values of the flat road of 0.05 g. The simulation did not return mean values close to 0.05 g, as shown in Figure 8b, since the roughness on the virtual flat road is not considered. Peaks, however, of 0.1 g and 0.11 g were also observed.



(a)



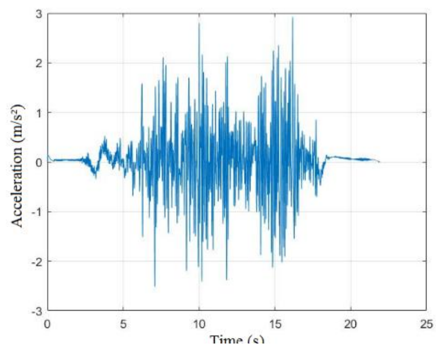
(b)

Figure 8 – Acceleration signal acquired during transition at a slope (a) and MBM result (b)

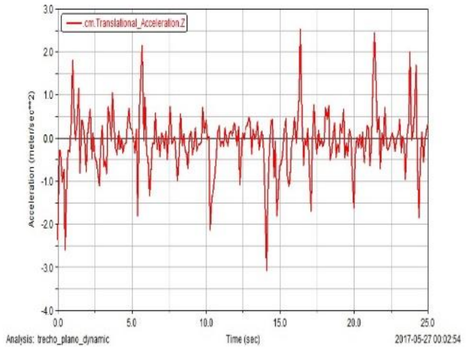
A3 – Flat gravel road

For the flat gravel road, simulations were performed with a load case input into the wheels, using the Test Rig. The load case was generated with mean elevations of the road in a vertical motion quasi-static state for 25 seconds.

Figure 9a shows that the imposed accelerations were quite random, even with the application of filters, the signal presents a noisy characteristic, consistent with the condition of the track. The highest peaks seen for the acquired signals were of 3.0 m/s². The simulation results in Figure 9b, agreeing with this, showed a similar result, with highest peaks of 2.5 m/s².



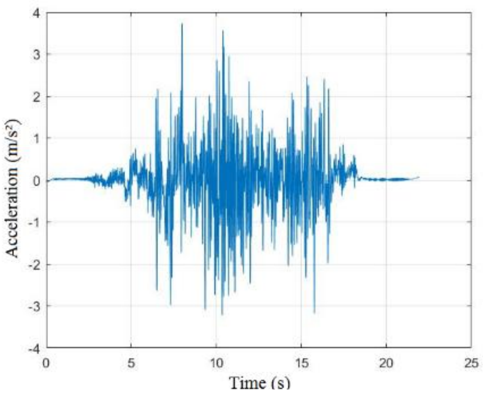
(a)



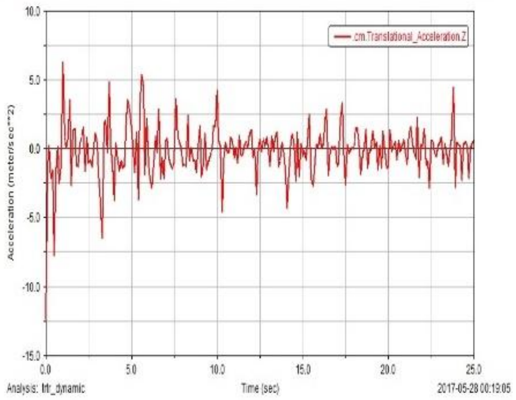
(b)

Figure 9 – Acceleration signal acquired during gravel road (a) and MBM (b), front suspension

As for the rear suspension, average values were higher, as expected. In general, the loads had greater magnitude in the rear suspension. One of the main reasons is that most of the vehicle's weight is concentrated on the rear part. In all tests, this result could be observed. This pattern was repeated for the gravel road and the highest peaks were between 2 and 4 m/s². Figures 10a and 10b show, respectively, the results of accelerations collected and those of the simulation. The latter showed higher peaks, of approximately 5 m/s², though.



(a)



(b)

Figure 10 – Acceleration signal acquired during gravel road (a) and MBM (b), rear suspension

The results could have been better converging in case real road displacement mapping was used. As the creation of three-dimensional roads with high roughness is complex and time-consuming method, especially when data is acquired at high rates, meaning that for an acquisition rate of 600 Hz each second of road would correspond to 600 points per second. A way to better deal with this would be collecting data at a lower rate, as the mapping of the lane won't be much affected. Then, with a double integration and the use of a proper transference function, carry out a lane reconstruction.

B – Double integration of a signal

The track with raised pavement markers, due to the repetition of obstacles of same known elevation, was double integrated, in order to obtain displacement data. This method is suggested by [19]. After the double integration performed through MATLAB, the acceleration signal exhibited the behavior seen in the graph of Figure 11. The displacements computed in the rear wheel were all in a range of 40 to 50 mm for each marker, with a behavior showing the periodicity expected, as the markers have a height of approximately 45 mm.

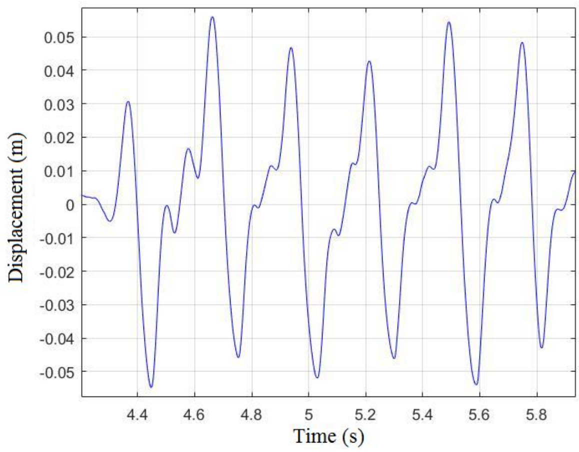


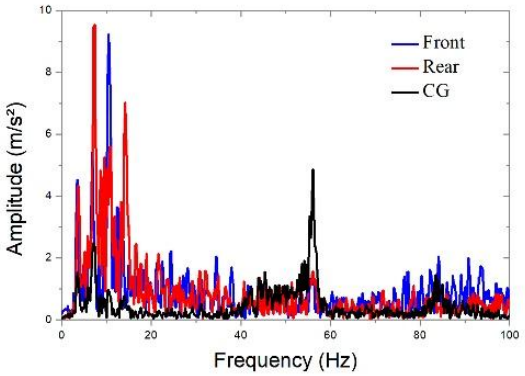
Figure 11 – Displacement during route with raised pavement markers

The double integration of an acceleration signal is an interesting mathematical tool that reproduces integrally the displacements suffered in the track by a certain part of the vehicle and its validation with well-defined displacements was obtained successfully. These results can be further used to carry out a road reconstruction [21]. This again proves that the double integration is another interesting method to obtain the displacement mapping of a lane. Likewise, the displacements in the CG of a vehicle are of a great deal of importance, to qualify issues such as perceived comfort and car handling.

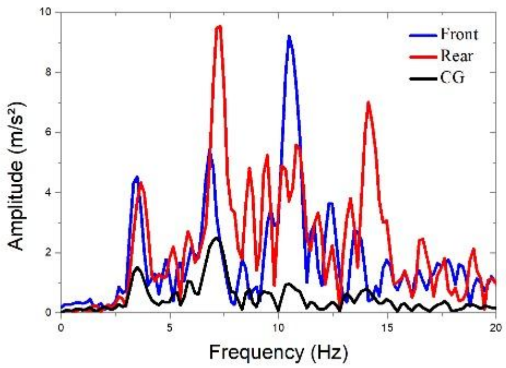
C – Frequency domain

For further evaluation of the data, a frequency domain approach was also considered, as it helps validate data [22].

With the application of the FFT to the acceleration signal, it was possible to obtain data in the frequency domain. The analysis was in the range of 0 to 100 Hz. The obstacle that resulted in higher amplitudes in the unsprung mass was the bump track. In all the obstacles, it was noticed that the sprung mass presents peaks in the region near 60 Hz. One of the hypotheses is that this is due to the vibration caused by the engine, because it is supported directly on the structure of the vehicle. Figure 12a shows the amplitudes of acceleration in the frequency domain for both suspensions and the car's CG passing over the markers on the track. Figure 12b shows the detail for frequencies between 0 and 20 Hz. Figure 13 shows the frequency domain data for the engine support bar during idling at high revving. A noticeable peak at around 60 Hz could sustain the hypothesis of the engine provoking the peaks at the structure. Moreover, revolutions per minute at the engine measured with a tachometer, revealed a high revving of 3800 RPM, or angular velocity of 63.33 Hz.



(a)



(b)

Figure 12 – Frequency domain response during bumpy track (a) and an inset from 0-20 Hz

As seen, the engine support bar has a peak near 63.33 Hz. The car body shows a peak at approximately 57 Hz. This is because the engine, while performing the bumpy maneuver, is not at its peak rotation.

Another interesting point is that the frequency of the highest accelerations of all signals is in the 0 to 20 Hz range, or ride range for road unevenness ($0.5 < \nu < 20$ Hz), something discerned in other studies [23-27]. Apart from that, human discomfort, when traveling in vehicles, has been proved to be more sensitive in low frequencies (from 3 to 10 Hz), above the vehicle's eigenfrequency, and higher acceleration, such as the data presented by [28] in a subjective discomfort assessment and in multibody approaches [7, 29, 30]. In this case, the accelerations in low frequencies perceived in the CG were relatively high, implying some level of discomfort to the pilot. These signals, though, show that there is a great dissipation of the imposed loads, comparing the accelerations measured in the unsprung masses with the sprung mass.

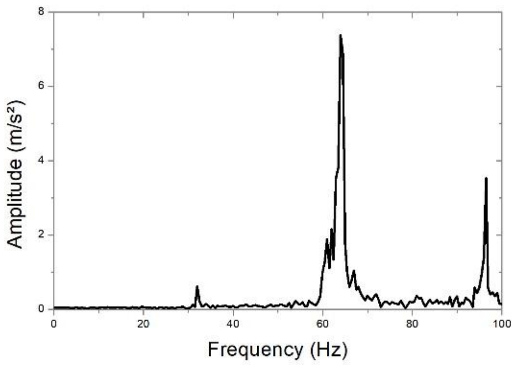


Figure 13 – Frequency domain response at the engine support, during vehicle idling

IV. Conclusions

As a final statement, the simulations had a very coherent result, proving that the virtual model has a high degree of reliability. Off-road virtual tracks, however, showed some difference between real road condition signals, as the former aren't as well detailed as the latter. The acceleration values captured in the obstacles were high for the unsprung masses, which makes these types of obstacle some of the most demanding of the Mini Baja's suspensions during dynamic tests.

The adopted accelerometer-based technique proved to be satisfactory for obtaining data and, in addition, the comparison of data shows that signals of virtual origin from the MBM are very similar to those of experimental origin, proving that the creation of suspension models and its simulation with virtual roads in ADAMS Car result in coherent results. The triumph of this modelling is in its adjustability and versatility, being able to generate a wide range of analyses, such as structural analysis in transient state, modal analysis of several components that make up the vehicle suspension, as well as a thorough analysis of vehicle dynamics.

Despite all the imposed simplifications, the MBM used also allows the modification of parameters such as structural bar lengths, CG coordinates, masses involved, tire stiffness, among others, thus helping the team with improved time management of its project. A closer approximation of the model can still be achieved by using the cage structure, pilot representation, as well as the inclusion of flexible elements that are part of the Mini Baja vehicle, to make it as close to reality as possible. As aforementioned, the procedures adopted for these tests can be used for other types of studies, whether it is deepening concepts of vertical vehicle dynamics, comparison between different vehicle configuration, modal and structural analyses, and comfort study with an improved methodology, with severer obstacles, variation of parameters, such as tire inflation and spring stiffness and the use of triaxial accelerometers.

ACKNOWLEDGMENTS

The authors thank the Universidade de Caxias do Sul Baja team for their support in the completion of this work and LAMEC – The Laboratory of Mechanical Testing at Universidade de Caxias do Sul.

V. BIBLIOGRAPHY

[1] A. González, et al. "The use of vehicle acceleration measurements to estimate road roughness". *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility*, vol. 46, no. 6, pp. 483–499, 2008. (http://doi.org/10.1080/00423110701485050)

[2] M.W. Sayers and S.M. Karamihas. *The Little Book of Profiling*. Ann Arbor, MI, USA: UMTRI, 1998. (http://www.umtri.umich.edu/content/LittleBook98R.pdf)

[3] L. Frýba. "Dynamic Interaction of Vehicles with Tracks and Roads". *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility*, vol. 16, no. 3, pp. 129–138, 1987. (http://doi.org/10.1080/00423118708968874)

[4] L. Gagnon, M.J. Richard and G. Doré. "A multibody dynamics model to assess the impact of road unevenness on the efficiency of a semitrailer truck". *Int. J. Vehicle Systems Modelling and Testing*, vol. 10, no. 1, pp. 1–28, 2015. (http://doi.org/10.1504/IJVSMT.2015.067519)

[5] O. P. Joshi, et al. "Investigating Effect of Road Roughness and Vehicle Speed on the Dynamic Response of the seven degrees-of-freedom Vehicle Model by using Artificial Road Profile". *International Journal of Current Engineering and Technology,* , vol. 5, no. 4, pp. 2596–2602, 2015. (http://inpressco.com/wp-content/uploads/2015/07/Paper542596-2602.pdf)

[6] P. R. Pawar and M. R. Saraf. "Measurement of Road Profile and Study its Effect on Vehicle Durability and Ride,". *SAE Technical Paper* 2009.

[7] X. Ning, C. Zhao and J. Shen. "Dynamic analysis of car suspension using ADAMS/car for development of a software interface for optimization". *Procedia Engineering*, vol. 16, no. 1, pp. 333–341, 2011. (http://doi.org/10.1016/j.proeng.2011.08.1092)

[8] C.B. Patel, P.P. Gohil and B. Borhade. "Modelling and vibration analysis of a road profile measuring system". *International journal of automotive and mechanical engineering*, vol. 1, no. 1, pp. 13–28, 2010. (http://dx.doi.org/10.15282/ijame.1.2010.2.0002)

[9] L. Yongjie, et al. "Numerical and experimental investigation on stochastic dynamic load of a heavy duty vehicle". *Applied Mathematical Modelling*, vol. 34, no. 10, pp. 2698–2710, 2010. (http://doi.org/10.1016/j.apm.2009.12.006)

[10] N. Amati, et al. "Dynamic modelling and experimental validation of three wheeled tilting vehicles". *Vehicle Systems Dynamics*, vol. 49, no. 6, pp. 889–914, 2011. (http://doi.org/10.1080/00423114.2010.503277)

[11] M. Doumiati, et al. "Estimation of road profile for vehicle dynamics motion: experimental validation'". In: *American Control Conference (ACC)*, San Francisco. San Francisco: IEEE, 2011. pp. 5237–5242. (http://doi.org/10.1109/ACC.2011.5991595)

[12] T. Murakami and T. Shiiba. "Road Profile Estimation From Vehicle State With Multibody Dynamic Analysis". In: *5th Annual Dynamic Systems and Control Conference joint with the JSME 2012 11th Motion and Vibration Conference*, American Society of Mechanical Engineers, 2012. pp. 577-581. (http://dx.doi.org/10.1115/DSCC2012-MOVIC2012-8637)

[13] S. Kushairi, et al. "Tractor-trailer modelling and validation". *Int. J. Heavy Vehicle Systems*, vol. 21, no. 1, pp. 64–82, 2014. (http://dx.doi.org/10.1504/IJHVS.2014.057802)

[14] M. Blundell and D Harty. *The Multibody Systems Approach to Vehicle Dynamics*. 1. Oxford, England: Butterworth-Heinemann, 2004.

[15] F. Orengo, M. H. Ray and C. A. Plaxico. "Modeling Tire Blow-Out in Roadside Hardware Simulations Using LS-DYNA". In: *International Mechanical Engineering Congress and Exposition*, Washington, USA. Washington, USA: ASME, 2003. pp. 15–21. (http://doi.org/10.1115/IMECE2003-55057)

[16] K. M. Billal, et al. "Virtual Validation & Test Correlation of Wheel Using Abaqus Solver". In: *Simulia India Regional User Meeting*, Chrysler Technology Center. Chrysler Technology Center, 2014. pp. 1–15.

[17] M. J. Stallmann, P. S. Els and C. M. Bekker. "Parameterization and modelling of large off-road tyres for ride analyses: Part 1–Obtaining parameterization data". *Journal of Terramechanics*, vol. 55, pp. 73– 84, 2014. (http://doi.org/10.1016/j.jterra.2014.05.004)

[18] M. Nybacka. "Validation of ADAMS/Car simulations and dynamic performance in multi-link suspensions". 2005. f. Master's thesis – Luleå Tekniska Universitet, Luleå, Sweden, 2005. (http://www.diva-portal.org/smash/get/diva2:1031663/FULLTEXT01.pdf)

[19] H. Imine, Y. Delanne and N.K. M'Sirdi. "Road profile input estimation in vehicle dynamics simulation". *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility*, vol. 44, no. 4, pp. 285–303, 2006. (http://doi.org/10.1080/00423110500333840)

[20] R. R. Teixeira, S. R. D. S. Moreira and S. M. O. Tavares. "Multibody dynamics simulation of an electric bus". *Procedia Engineering,*, vol. 114, no. 1, pp. 470–477, 2015. (http://doi.org/10.1016/j.proeng.2015.08.094)

[21] P. Gáspár, Z. Szabó and J. Bokor. "Parameter identification of a suspension system and road disturbance estimation". *Int. J. Vehicle Systems Modelling and Testing*, vol. 2, no. 2, pp. 128–137, 2007. (http://doi.org/10.1504/IJVSMT.2007.013891)

[22] E. Kutluay and H. Winner. "Validation of vehicle dynamics simulation models – a review". *Vehicle System Dynamics: International Journal of Vehicle Mechanics and Mobility*, vol. 52, no. 2, pp. 186–200, 2014. (http://doi.org/10.1080/00423114.2013.868500)

[23] E. Pennestrì, P.P. Valentini and L. Vita. "Comfort analysis of car occupants: comparison between multibody and finite element models". *Int. J. Vehicle Systems Modelling and Testing*, vol. 1, no. 1/2/3, pp. 68–78, 2005. (http://doi.org/10.1504/IJVSMT.2005.008573)

[24] V. Cossalter, et al. "Frequency-domain method for evaluating the ride comfort of a motorcycle". *Vehicle Systems Dynamics*, vol. 44, no. 4, pp. 339–355, 2006. (http://doi.org/10.1080/00423110500420712)

[25] T. Bruns and E. Schäfer. "Modelling and identification of an all-terrain vehicle". *Int. J. Vehicle Systems Modelling and Testing*, vol. 2, no. 3, pp. 276–295, 2007. (http://doi.org/10.1504/IJVSMT.2007.016238)

[26] K.V. Gangadharan, C. Sujatha and V. Ramamurti. "Dynamic response of railroad vehicles: a frequency domain approach". *Int. J. Heavy Vehicle Systems*, vol. 15, no. 1, pp. 65–81, 2008. (http://doi.org/10.1504/IJHVS.2008.017984)

[27] Ł. Konieczny, R. Burdzik and T. Figlus. "Use of MBS (ADAMS/CAR) software in simulations of vehicle suspension systems". *Transport Problems*,

vol. 9, no. 1, pp. 111-118, 2014. (http://transportproblems.polsl.pl/pl/Archiwum/2014/zeszyt1/2014t9z1_12.pdf)

[28] T. Ibicek and A.N. Thite. "In Situ Measurement of Discomfort Curves for Seated Subjects in a Car on the Four-Post Rig". *Advances in Acoustics and Vibration***,** vol. 2014, no. 1, pp. 1–14, 2014. (http://dx.doi.org/10.1155/2014/239178)

[29] F. Y. Yi and S. K. Zhang. "Ride Comfort Simulation under Random Road Based on Multi-Body Dynamics". In: *3rd International Workshop on Intelligent Systems and Applications (ISA)*, IEEE, 2011. pp. 1-3. (http://doi.org/10.1109/ISA.2011.5873455)

[30] B. Jiang "The Comparative Analysis of Car Ride Comfort with Two Different Front Suspensions". In: *International Conference on Mechanical Engineering and Control Automation*, 2017. pp. 295–300. (http://dx.doi.org/10.12783/dtetr/icmeca2017/11948)