

UNIVERSIDAD POLITÉCNICA DE MADRID

Escuela Técnica Superior de Ingenieros Industriales



**SPOT-WELDED FLANGES CONTACT MODELING FOR THE ENHANCEMENT OF
LINEAR FINITE ELEMENT MODELS ACCURACY. APPLICATION TO THE MODAL
BEHAVIOR CORRELATION IMPROVEMENT OF A VEHICLE STRUCTURE**

DOCTORAL THESIS

Luis Guilherme Mariano Viana Martins
M.Sc. Automotive Engineering

2023

Departamento de Ingeniería Mecánica
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El presidente,

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Los vocales,

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A mi padre, mi primer maestro, con quien entre tantas cosas, aprendí el gusto por estudiar. Sé que te alegrarías mucho por esto.

RESUMEN

Las simulaciones estructurales de ingeniería han requerido modelos computacionales cada vez más complejos para sustituir a los ensayos físicos con precisión. Este trabajo se dedica especialmente al análisis numérico de estructuras de carrocerías de vehículos, cuyo tamaño y complejidad hacen inviable el uso de modelos no lineales de alta precisión, debido a los inviables esfuerzos computacionales implicados.

El propósito de este estudio es encontrar una nueva metodología para modelar uniones soldadas por puntos en modelos lineales de elementos finitos de estructuras de carrocería de vehículos de paredes delgadas, mejorando la precisión del modelo sin aumentar su complejidad. Utilizando un conjunto de modelos no lineales simplificados, se ajustan las propiedades de rigidez y amortiguamiento de estas uniones soldadas, ajustes estos que, en seguida, se utilizan en un modelo lineal que representa toda la estructura de la carrocería del vehículo.

Los resultados ajustados obtenidos son comparados con los ensayos físicos, desde lo que se puede percibir una clara mejora en la precisión de las respuestas modales y de frecuencia proporcionadas por el modelo lineal de elementos finitos, aunque manteniendo su nivel de complejidad inicial.

En la secuencia, el modelo ajustado se utiliza en un análisis de optimización para reducir el peso de la estructura, disminuyendo también su coste, y además posibilitando reducciones importantes en el uso de los recursos naturales y en las emisiones de carbono.

ABSTRACT

Structural engineering simulations have required increasingly complex computational models to replace physical tests accurately. This work focuses on the numerical analysis of vehicle body structures, whose size and complexity make the use of very accurate nonlinear models unfeasible due to the prohibitive computational costs involved.

The purpose of this study is to find a new approach to model spot-welded joints in linear finite element models of thin-wall vehicle body structures, improving the accuracy of the model without increasing its complexity. Using a set of simplified nonlinear models, the stiffness and damping properties of these welded joints are fitted, and those adjusted values are used in a linear model of the entire vehicle body structure.

The results are compared with experimental tests, showing a clear improvement in the accuracy of the modal and frequency responses provided by the linear finite element model, but keeping its initial complexity level.

The adjusted model is then used in an optimization analysis to reduce the structure's weight, leading to interesting cost savings and important reductions in the use of natural resources and carbon emissions.

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LIST OF ACRONYMS

DOF	Degree Of Freedom
EMA	Experimental Model Analysis
FE	Finite Element
FEA	Finite Element Analysis
FEM	Finite Element Model
FFT	Fast Fourier Transform
FRAC	Frequency Response Assurance Criterion
FRF	Frequency Response Function
MAC	Modal Assurance Criterion
MAE	Medium Absolute Error
MIG	Metal Inert Gas
MSE	Medium Squared Error
PSD	Power Spectrum Diagram

1. INTRODUCTION

The level of correlation between the behavior of a physical system and the results obtained from a finite element (FE) model is essential in many engineering areas. The accuracy of results from finite element analysis (FEA) increases with the correlation levels, thus reducing the demand for prototypes and providing high savings in time and resources for engineering projects. This statement is especially relevant in the automotive industry [1–3], where the use of FE models is necessary and determinant to accelerate product development cycles and increase the cost and time efficiency of new vehicle projects.

From this perspective, the validation of analytical simulation models became a decisive factor in developing new products, especially since the use of advanced computational resources became economically viable.

But despite of this continuous improvement of FEA [2,4–8], the accuracy level of linear models remains limited by the lack of nonlinear phenomena representation, such as friction and contact reaction forces. In addition, the complexity of nonlinear models needed to represent those phenomena limits its use to specific boundary conditions. Otherwise, it would be unfeasible to solve them considering the computational resources currently available. For this reason, linear models are traditionally mainly used in these situations.

In the case of a vehicle body subjected to a vibrational analysis, the structure is typically composed of a set of thin-wall stamped sheet metal parts connected by spot welds at their edges. As a consequence of these constructive properties, nonlinear contact between flanges of adjacent sheet metal parts will always occur.

Considering the high complexity given by the deep detail level required, the building and analysis of the simulation models are performed by using the FE technique, taking advantage of its flexibility in design representation and its ability to simulate distinct mechanical elements.

On the other hand, to perform vibration analyses, these FE models have to use the eigenvalue approach [9,10] at least up to 100 Hz, to capture the highest energy modes of the structure, leading to an undesirable combination of a large nonlinear model with a very demanding computational effort.

In this work, a new modelling methodology to increase the accuracy of linear FEA is presented, thus avoiding exceeding the computational resources currently available. To this end, the correlation level obtained with both the linear and the nonlinear models of the whole vehicle structure are compared.

Taking a vehicle body structure, a set of bench tests is performed to capture its modal behavior and its frequency response functions (FRFs). A reference FE model of the same vehicle body structure is also built to simulate an identical set of analysis to the field tests performed on the bench test (Figure 1). The results obtained from the reference FE model of the vehicle body structure are compared with those obtained from the bench tests, and the correlation coefficients between them are computed.

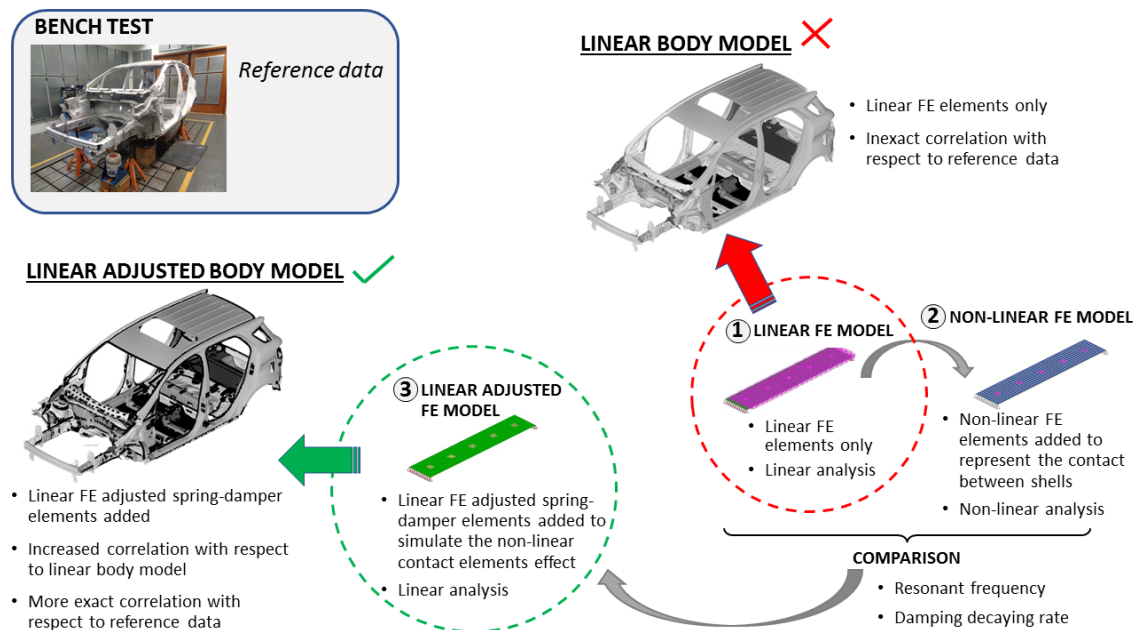


Figure 1. Scheme followed to develop the adjusted linear FE model of the vehicle body structure.

After that, three FE test models are built to represent a small part of the structure welded flange. First, a FE test model is built using only linear parameters, thus avoiding nonlinear behaviors in the contact area between the flanges of adjacent panels. Then a second nonlinear FE test model is built using the first model as a base, adding to the first model the nonlinear contacts of the flanged areas.

The difference of behavior between both models is computed. A third linear FE test model is also built using the first model as a base but now adding linear spring-damper elements to simulate the nonlinear behaviour at the contact areas. The elastic constants of these spring-damper elements are adjusted using the differences found between the responses of the first and second FE test models.

Finally, an adjusted FE model is built of the whole vehicle body structure, adding to the reference FE model of the body structure the linear spring-damper elements previously adjusted with the help of the FE test models. Then the same set of analysis carried out with the bench test and the

reference FE model are performed with this adjusted model, comparing its results with the bench test results, and computing the correlation coefficients.

As a result of the adjusted FE model analysis, the body structure shows an improvement of 10.6% in the correlation coefficient when compared to the results of the initial reference model. The modal behavior coefficients and the visual comparison of the FRF curves also shows a higher correlation given by the adjusted model, confirming the validation of the adjustment methodology proposed here.

During the state-of-the-art review, a knowledge gap is detected regarding the representation of nonlinear contacts in large finite element models subjected to frequency domain analysis. To overcome this gap, this work has the objective of finding a new approach to model spot-welded flanged joints into linear finite element models of thin-wall vehicle body structures, using a nonlinear model behavior as a reference. In this way, it can improve the accuracy of the modal and the frequency response function results without increasing the complexity of the FE model analysis.

2. LITERATURE REVIEW

The relevance of the dynamic behavior of a mechanical system to the success of its simulation has been discussed over time by distinct authors. Furthermore, the finite element models used to represent these systems have been continuously improved. The parallel evolution of these two factors has been fundamental for increasing the accuracy of simulation models, and therefore, their ability to predict results.

In [4], Baker, M. compared different methodologies for test analysis model reductions as Static Reduction, Modal Reduction, Improved Reduced System and Hybrid Reduction; concluding that each method applicability is dependent of the dynamic characteristics of each model. The author also identified issues related to the modal test results acquiring, especially due to missing data associated to the modal reduction process.

So, an optimized selection of the model parameters is considered, as well as the use of computer graphic representation, with the aim of helping the modal identification and characterization of the model eigenmodes.

In fact, the analysis of the correlation by comparing the modal shapes of finite element models against experimental tests proves to be an efficient and practical tool to evaluate the ability of these models to simulate the dynamic behavior of a structure. In addition to this, the use of the modal assurance criterion (MAC), a mathematical method to compare the eigenmodes of a mechanical system, also allows to quantify the level of correlation between finite element models and experimental tests.

The combination of these methods has been successfully used in different studies with simplified models. Its application can be used, as an example, to evaluate the effects that a calibration mass has on the modal behavior of a system or to better understand the effects that adjustments in the parameters of a model mesh can have on the results of its simulation.

In [11], S. V. Modak, T. K. Kundra and B. C. Nakra used a constrained optimization method for updating the FE model of a structure. The analysis is based on minimizing a mathematical objective function, in this case represented by the norm-based error between the measured and the analytical versions of the eigenfrequencies of the identified mode shapes. The modal assurance criterion (MAC) coefficient is used to evaluate the correlation level obtained.

For this, a steel bolted testing structure is build and exited by an impact hammer, having the accelerations acquired in 16 different locations of the structure with the aim of measure the eigenfrequencies and identify the mode shapes of the structure.

Then, a FE model is build and submitted to the same loads already used, having also the accelerations calculated in the model at the same 16 locations of the structure, with the aim of calculating the eigenfrequencies and identify the mode shapes using the FE model of the structure (Table 1).

<i>Correlation of measured and FE-model-based modal data</i>				
Mode no.	Measured frequency (Hz)	FE-model predictions		MAC-value
		Frequency (Hz)	% Error	
1	34.95	43.05	23.17	0.9901
2	104.02	123.67	18.89	0.9470
3	133.96	185.21	38.26	0.9265
4	317.52	385.17	21.30	0.9054
5	980.16	1020.06	4.07	0.7299
6	1057.8	1084.79	2.55	0.8040
7	1531.45	1925.76	25.74	0.8798

Table 1. Comparison between the measured and the FE-model mode shapes [11]

Next, the model is updated according to the constrained optimization method proposed, by using the first five eigenvalues only in the updating process (Table 2).

<i>Correlation of the measured and the updated-model-based modal data</i>				
Mode no.	Measured frequency (Hz)	Updated-model predictions		Mac-value
		Frequency (Hz)	% Error	
1	34.95	34.86	-0.25	0.9954
2	104.02	103.36	-0.63	0.9638
3	133.96	134.03	0.05	0.9410
4	317.52	320.63	0.98	0.9370
5	980.16	980.21	0.00	0.4231
6	1057.8	1003.27	-5.15	0.5517
7	1531.45	1509.78	-1.41	0.9629

Table 2. Comparison between the measured and the updated FE-model modal shapes [11]

As a result, looking at the first five modes only and without imposing any bounds on the updating variables, it can be noticed that the error in natural frequencies has been reduced to less than 1%. Also, except for the fifth mode, the MAC coefficients have improved, indicating that the correlation between the mode shapes increased.

Next, a concentrated mass is add at different positions of the model in order to evaluate the effects of potential design modifications made to the structure. So, a local reinforcement is add to the structure with this same purpose. In both cases the updated FE-model showed improved capacity to predict the modal behavior of the structure.

In [12], Bais RS, Gupta AK, Nacra BC used a calibrated lumped mass to adjust an analytical FE model of a drilling machine structure, in order to improve the model correlation accuracy calculations respect to the experimental test results.

First, the drilling machine structure (Figure 2) eigenfrequencies and mode shapes are measured with the help of a accelerometers meshing.

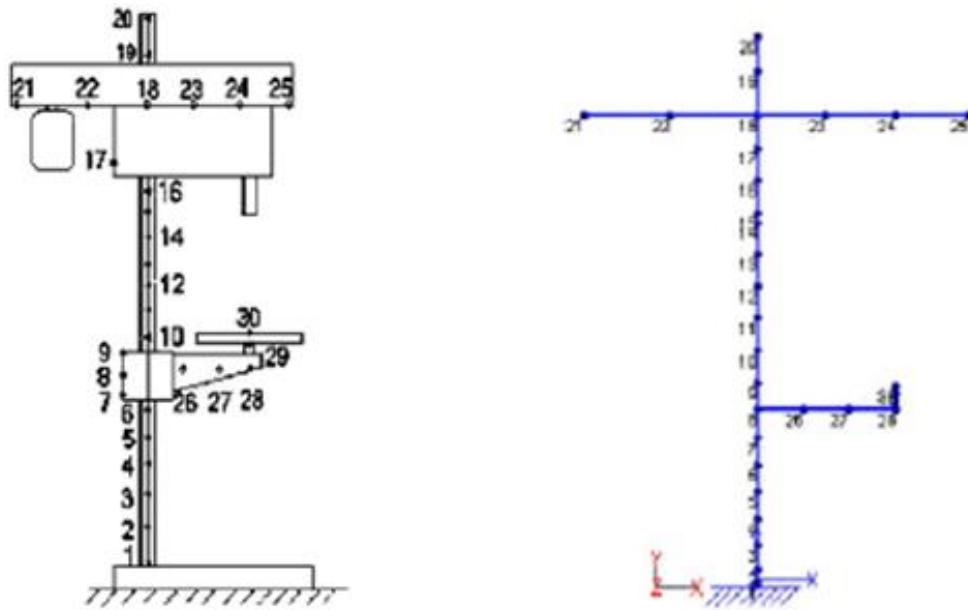


Figure 2. Drilling machine structure and simplified FE representation [12]

Then a FE model is built to represent this structure, and its eigenfrequencies and mode shapes are calculated and compared (Table 3).

Mode number	Study 1			Study 2		
	Analytical model	Experimental model	MAC values	Analytical model	Experimental model	MAC
Model1	10.29 Hz	8.67 HZ	0.953	11.20 Hz	8.79 Hz	0.946
Mode 2	65.14 Hz	47.34 HZ	0.901	63.37 Hz	44.40 Hz	0.906

Table 3. Comparison between the experimental and FE analytical model eigenfrequencies [12]

In the sequence the FE model is adjusted with a calibrated mass and its eigenfrequencies and mode shapes are re-calculated (Table 4)

Mode no.	Measured frequencies	Sensitivity method
1	8.37 Hz	8.40 Hz
2	46.05 Hz	43.20 Hz

Table 4. Experimental and FE analytical model eigenfrequencies with mass adjustment [12]

Using this adjustment as a new starting point, both drilling machine structure and FE model are submitted to updates with the addition of a set of two masses at two locations, one at a time.

As a result, considering these FE model adjustments performed to the drilling machine structure and FE model, similar eigenfrequency results and mode shapes are obtained, confirming the effectiveness of the proposed methodology.

In [13], Steenackers G. and Guillaume. P describe a response surface method, based on the analysis of a model in which the estimation of one of the parameters is performed through an automated model updating procedure.

For this, an aluminum plate specimen is manufactured and tested by using a calibrated impact hammer and an accelerometer set. The first 10 plate eigenfrequencies are measured considering free-free boundary conditions. Comparative results are shown in Table 5.

Test modes (Hz)	F.E. 12110 (Hz)
404.33	409.59
820.51	843.84
1115.16	1128.42
1702.86	1757.37
2185.34	2206.21
2735.45	2803.82
3589.07	3606.02
3948.22	4032.3
5065.04	5062.9
5356.18	5410.85

Table 5. Comparison between test and FE eigenfrequencies [13]

A FE model of the plate is also built, and the first 10 plate eigenfrequencies are calculated using the same experimental test boundary conditions, so the FE results can be compared with the bench tests.

On the sequence, a FE model update parameter is developed based on the comparison between the measured and calculated results. For this, the plate length (L), the plate material Young's modulus (E), and the plate width (W) parameters are updated, in this order, according with three steps:

- Calculate the polynomial expression found as a result of the interpolation between the measured and the analytical updated eigenfrequencies values (Figure 3, left);
- Compare the measured and the calculated eigenfrequencies values with the aim of minimize the cost function;
- Estimate the new parameters L, E, and W by solving the least-squared problem.

First order length (L) and the second order width (W) polynomial interpolation function is also shown (Figure 3, right).

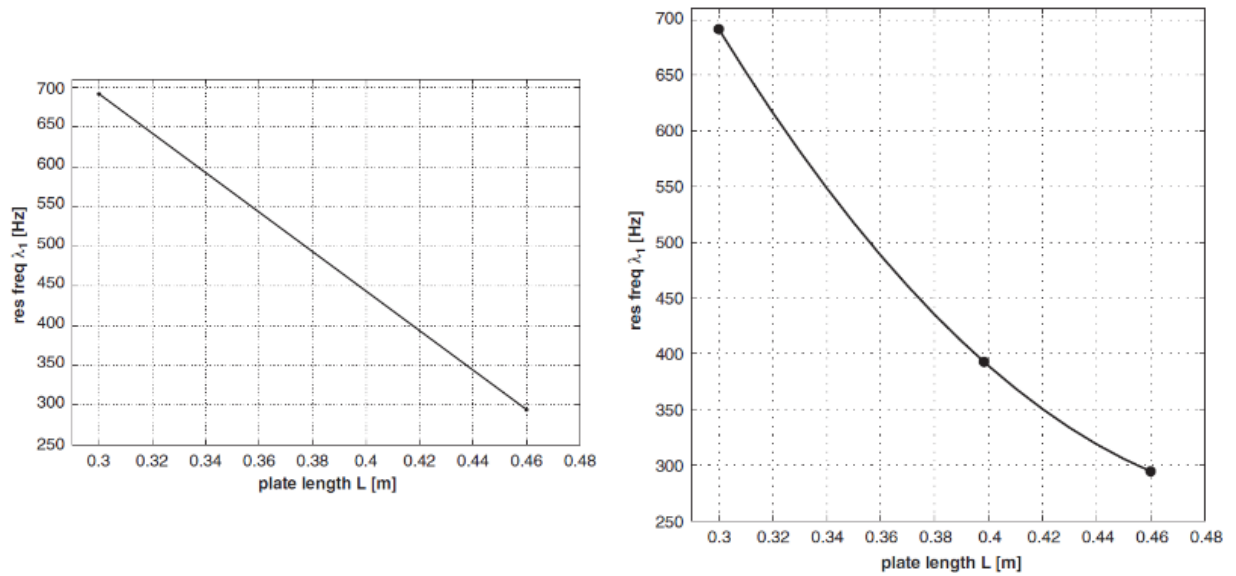


Figure 3. First (left) and second (right) order interpolation of plate resonance values [13]

Then, considering the L, E, and W updated values the eigenvalues model are re-calculated. As a result, they show very close values when compared with the measured eigenmodes (Table 6), despite of the model accuracy decreases after frequencies above 5000Hz.

Mode no.	Mean (Hz)	Standard deviation (Hz)
1	403.81	0.51
2	820.55	0.10
3	1115.14	0.10
4	1703.72	1.01
5	2185.41	0.36
6	2734.84	0.67
7	3589.51	0.71
8	3948.29	0.27
9	5062.26	3.70
10	5370.27	8.38

Table 6. Comparison between measured and updated eigenvalues [13]

So, in addition to the evaluation of eigenmode shapes and MAC coefficients, other studies that have continued to advance also evaluate the frequency response functions, but still using low and medium complexity models.

In these cases, the accuracy of the proposed stiffness and damping adjustment solutions are also evaluated by the comparison between the transfer functions.

In [14] Lin R.M. and Zhu J. proposes a method to improve the accuracy of a FE model of a structure by using the response of a base excitation data to update the analytical FRF.

For this, a cantilever beam is built and excited, having its experimental response measured. Then, a FE model of the cantilever beam is modeled, and its analytical response functions acquired.

After this, the analytical response functions already obtained are updated by the proposed method and compared with the updated results (Figure 4).

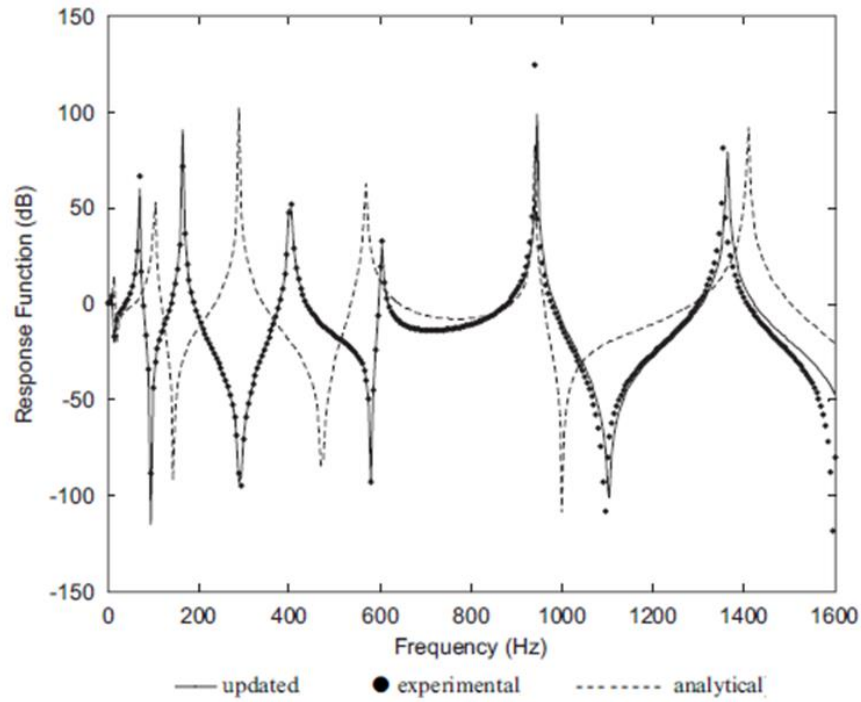


Figure 4. Comparison between measured and updated eigenvalues [14]

On the sequence three other cases are studied: a truss with complete coordinate data available, a truss with uncomplete coordinate data available (Figure 5), and the same cantilever beam already studied, but this time considering the addition of a hypothetical function measurement noise.

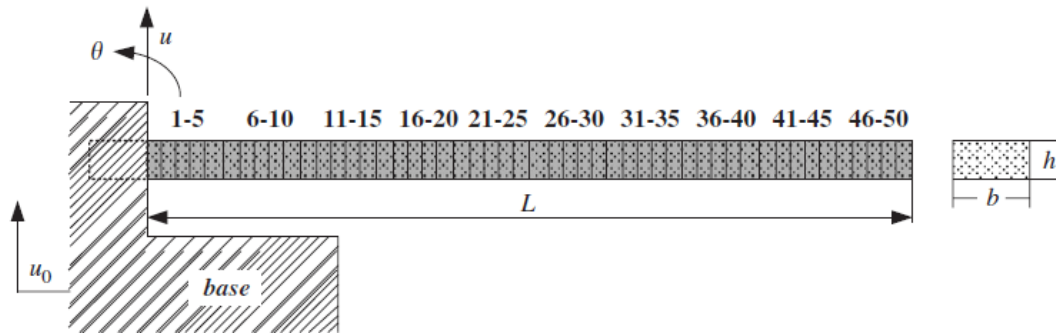


Figure 5. Refined FE mesh used for experimental model of a cantilever beam attached to a base [14]

As a result, although the studied cases are considered relatively simple, the method demonstrated high potential to successfully update the analytical response functions, even under some noise disturbance conditions.

In [7] Arora V., Singh S.P. and Kundra T.K. proposes a FRF updating process that, besides of the stiffness and mass matrices, also include damping matrices, with the aim of improve the analytical results accuracy.

The process is split in two steps. In the first, an FRF-based structure method is used to update the real part of the stiffness and mass matrices of the structure, but without proceeding with any modal extraction.

On the sequence, the updated stiffness and mass matrices of the structure are used to identify the damping matrix of the structure with the use of a direct method.

Two case studies are performed to measure the proposed method accuracy. In the first case, a simple fixed-fixed beam is excited, and the analytical results are compared with a simulated experimental test data (Figure 6, left). So, the analytical FRF is updated according with the first and second steps proposed, and then compared with the simulated-experimental test data (Figure 6, right).

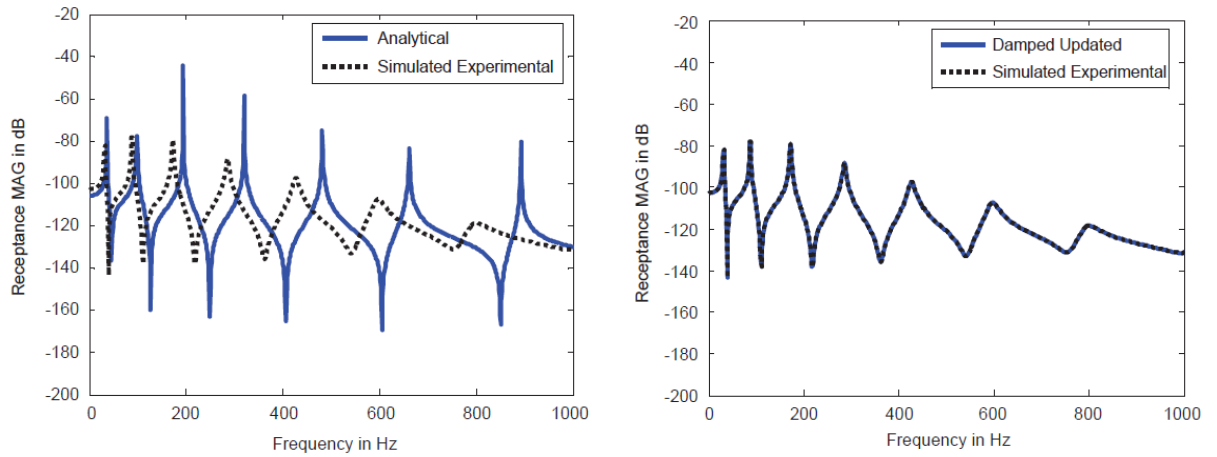


Figure 6. Overlay of simulated experimental and analytical first case FRFs before (left) and after (right) updating [14]

In the second case, an experimental FRF of a F-shape structure is obtained from a test. In this case, this structure have higher complexity level than the first case, having more degree of freedoms than that. So, the analytical FRF is compared with the FRF acquired from the experimental test.

As same as already done with the first case, the second case analytical FRF is also updated according with the first and second steps proposed, and then compared with the simulated-experimental test data. The proposed method shows very similar results in the first case where the structure have very low complexity (Figure 7, left). In the second case, less accurate results are noticed due to the higher complexity level of the structure tested (Figure 7, right).

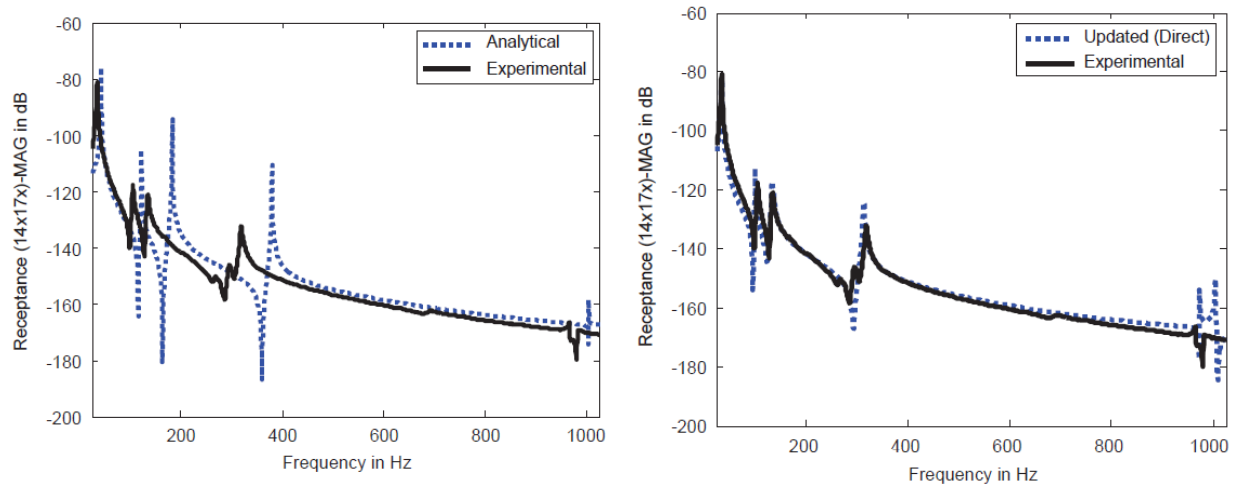


Figure 7. Overlay of simulated experimental and analytical second case FRFs before (left) and after (right) updating [14]

In [15] Lin R.M., and Ewins D.J. proposes an iterative method based on the comparison between two sets of FRFs acquired from an experimental test and a mathematical analysis is presented.

The proposal is to use the real part of the FRFs to update both the mass and the stiffness matrices, without proceeding with a modal extraction.

Two case studies are used to verify the method effectiveness. The first case is based in a numerical 8 degree of freedom (DOF) mass-spring model system. The second case is based in a 216 DOF FEM representation of a standardized truss structure (Figure 8), commonly used to assess the applicability of new analysis methods.

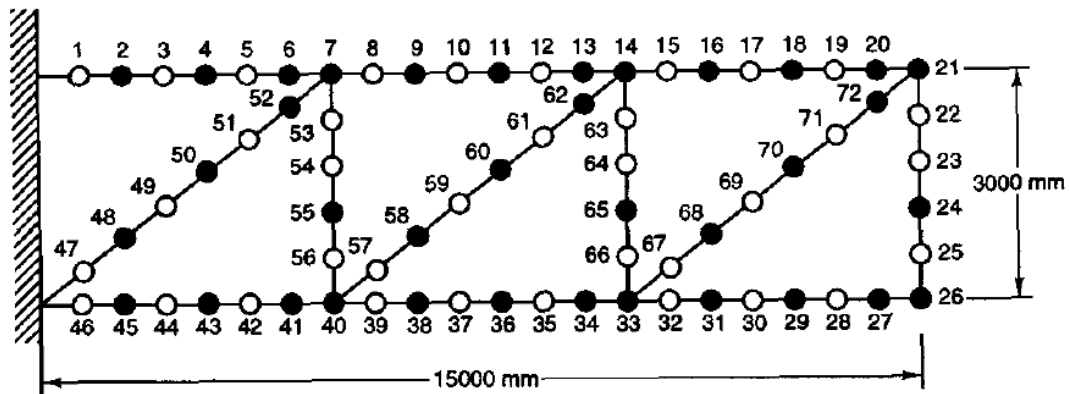


Figure 8. Standardized truss structure FEM [15]

The method show high similarity level between the experimental data and the analytical results obtained from the standardized 216 DOF FEM representation (Figure 9).

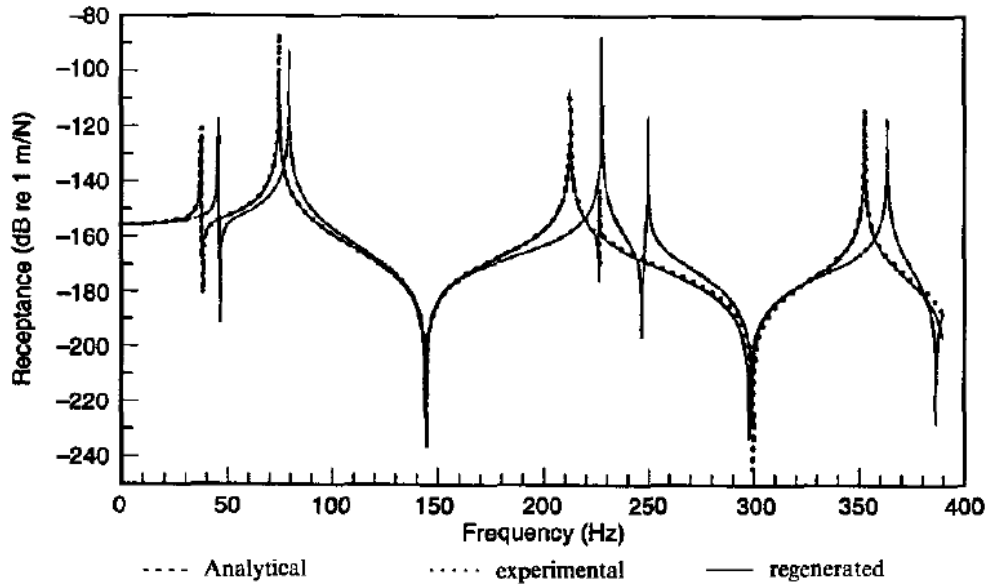


Figure 9. FRF comparison between experimental data and the analytical and regenerated results obtained from the standardized 216 DOF FEM [15]

Despite of the low complexity level of the analyzed objects, the study aim to a valid direction for the development of further correlation studies using more complex structures.

In [16] Pradhan S. and Modak S. V. proposes an updating method dedicated to improving the accuracy of structural damping matrix, while keeping the stiffness and mass matrices steady.

The method formulation is based on the comparison between normal and complex FRFs. For this, an experimental study is performed in a fixed-fixed beam structure (Figure 10) and the FRFs measured in a frequency range from 0Hz to 1000Hz, covering its first seven structural modes. In this case, as these FRFs are acquired from an experiment, their formulation is based in complex numbers.



Figure 10. Fixed-fixed beam structure [16]

Then, the numerical model is analyzed and the FRFs acquired. Like the experimental data, these analytical FRFs have their formulation based in complex numbers too. On the sequence, the numerical model is re-analyzed, but at this time without considering the structural damping by considering only the real part of the FRFs.

So, based on the analysis of the governing equations of motion of the numerical model without damping, an estimated structural damping is assessed, and a new set of updated FRFs calculated.

The comparison between the experimental data and the analytical results before and after updating is shown in Figure 11.

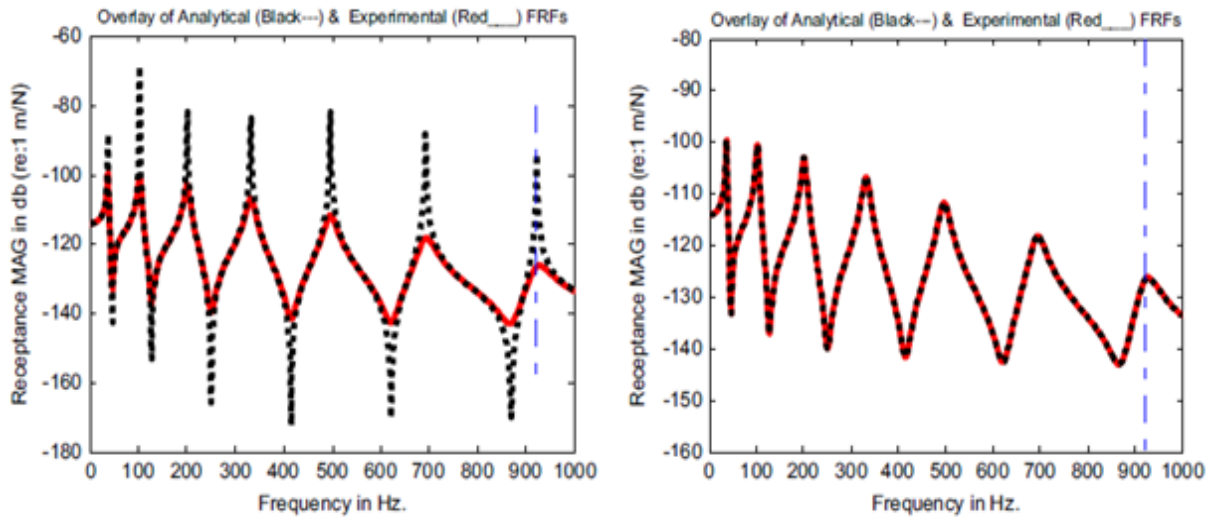


Figure 11. Comparison of FRFs overlay. Before updating (left) and after updating (right) [16]

Indeed, they show excellent results considering the studied structure and aims for the emphasizing the importance of using a separate treatment for damping phenomena when studying FRF update methods in more complex structures.

In addition to this, other works have advanced even further with the use of high complexity models for the evaluation of adjustments of multiple parameters such as mesh, mass, stiffness and damping.

The use of transfer functions as a parameter to evaluate the levels of correlation between simulation models and test results continue to be developed and used in different ways, as well as the evaluation of eigenmode shapes and MAC coefficients.

In [17], Seifi R. and Abbasi K. proposes an interference shaft-bush joint model to validate a method for updating the friction coefficient of a contact surface, with the aim of improving the precision of FE models stress distribution analyses.

For this, a refined FE model (Figure 12) is build considering high dimensional accuracy with the objective of having enough sensitivity to read the variation of stress on the contact surface.

Then, a bench test was performed with a set of shaft-bush joint species, in order to experimentally acquire the mechanical properties of the joint. Material data is also experimentally accessed.

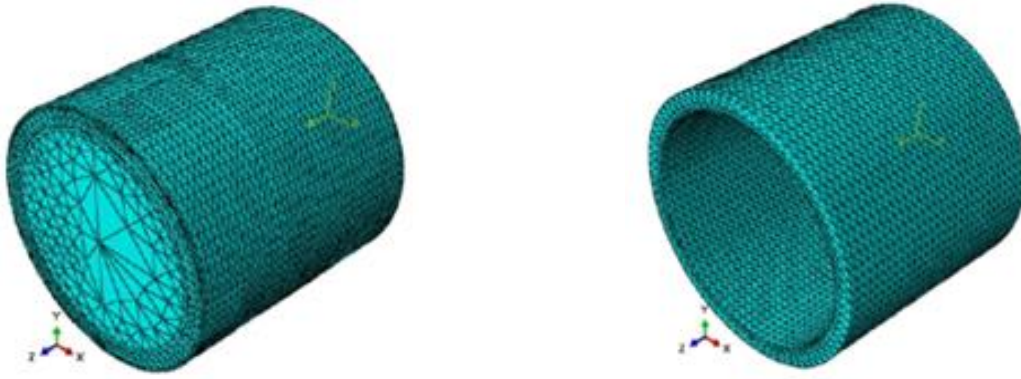


Figure 12. Mesh pattern of the shaft (left) and bush (right) in the interference modeling [17]

In that way, the data uncertain referring to the geometry of the parts and their construction materials are reduced to the minimum level, and the sources of error restricted to the variations in the coefficient of friction. So, the FE model is updated and analyzed using this data, and the updated stress distribution plotted considering both rough and smooth joints (Figure 13).

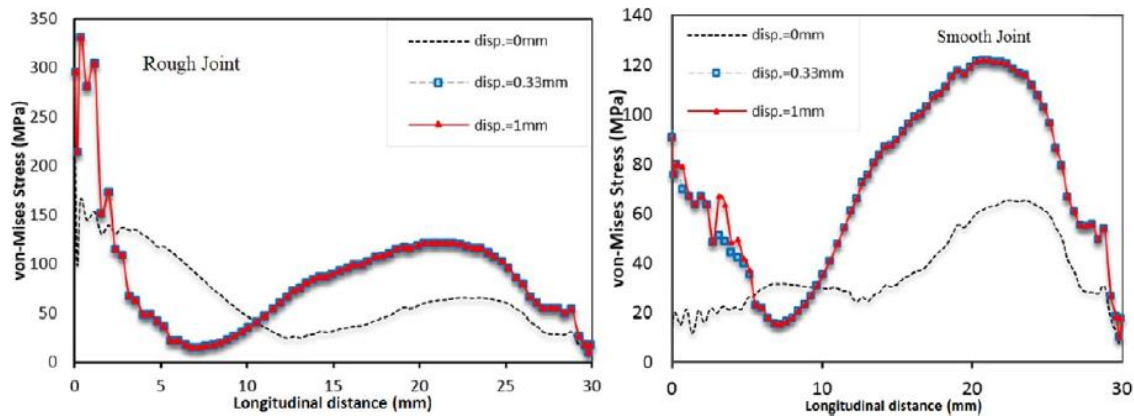


Figure 13. FE model results for von-Mises stress at rough (left) and smooth (right) joints [17]

As the extracted responses are not experimentally accessible, the contrast between the experimental and analytical data is not considered. Nevertheless, the ability of updating the FE model with experimental data leads to an improvement of the model reliability, and by consequence, to more accurate analytical results.

In [18] Min C-H, Hong S. and Park S-Y presents a FE updating method based on the sensitivity analysis of the modal behavior of a damped cantilever beam structure.

For this, an experimental test is first performed in order to acquire the modal parameters of the real structure. On the sequence a FE model of the structure (Figure 14) is build and analyzed, so the natural frequencies and the zero frequencies (the point minimum amplitude frequency point between two

sequential resonant frequencies) are calculated. A degree of freedom reduction method is also applied with the aim of enhance the FE analysis efficiency.

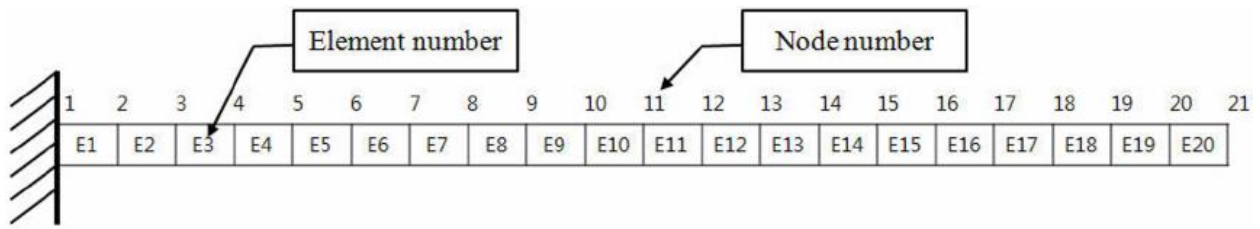


Figure 14. Numerical model of a cantilever beam [18]

So, the modal stiffness and mass matrices are numerically identified, and based on this, the viscous damping matrix is determined. Then the FE model is updated by minimizing the natural frequency and zero frequency components of the matrix with non-determined stiffness, already identified along the initial FE model sensitivity analysis.

The comparison between the natural frequency and zero frequency results of the initial and updated FE models shows the improvement of the accuracy of the FE model prediction. FRFs comparison confirms the good performance of the damping estimation updating, and the MAC analysis validates the mass, stiffness and damping matching between the experimental results and the updated FE model.

In [19] Kang Y., Qiu Z. and Zhang H. presents a FE updating method using a rotor-discs system supported by journal bearings for testing and validation.

For this, an experimental verification is performed in order to acquire the rotor-discs frequency responses. Then, a FE model of the dual rotor-discs (Figure 15) is build and analyzed, having its mass and stiffness matrices determined by an optimization process to correct that minimizes its object function.



Figure 15. Dual rotor-discs FE model [19]

On the sequence, the damping matrix is determined using the measured modal parameters from the updated matrices. So, the frequency responses from the initial and the updated models are predicted are compared with the measured data, having the stiffness and the damping experimental coefficients of the journal bearings analytically identified.

Then, the system eigenmodes of the initial and updated FE models are compared with the experimental data. The modal shapes and FRFs are plotted, and also the MAC coefficients (Figure 16) are calculated with this purpose.

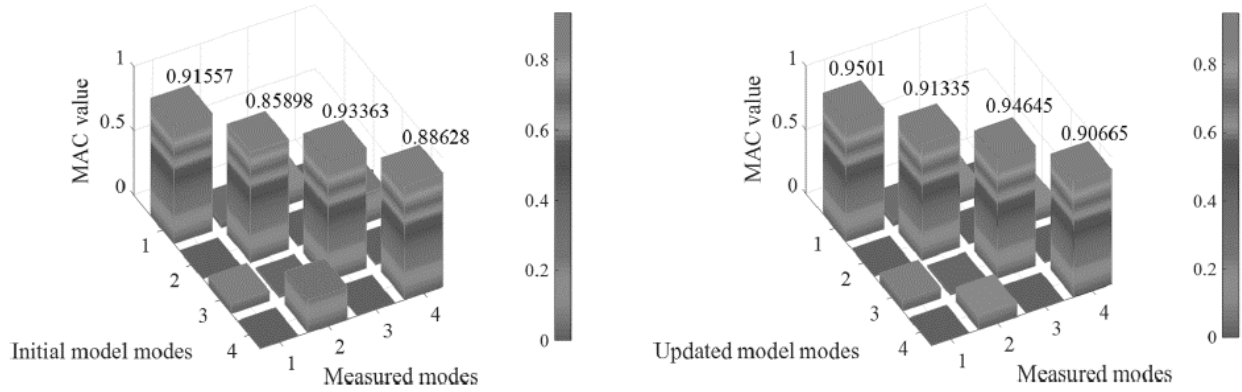


Figure 16. MAC results between the initial (left) and updated (right) FE models [19]

As a result, the updated FE model eigenmodes values and MAC coefficients are more accurate than the initial FE model when comparing with the experimental measurements.

Moreover, the more accurate FRF curves from the updated FE model respect to the initial FE model confirms the effectiveness of the method considering the proposed experiment.

In [20], O. S. David-west, J. Wang and R. Cooper presents a FE model updating method where the correlation coefficient between the test and the initial FE model is determined by performing a dynamic analysis of a thin wall structure (Figure 17) with the Lanczos method to extract the modes between 0 and 200 Hz, and to comparing it against a set of experimental test results.

Initially, the initial FE model was updated by changing the damping constant, material properties and discretization. So, the correlation coefficient was calculated again.

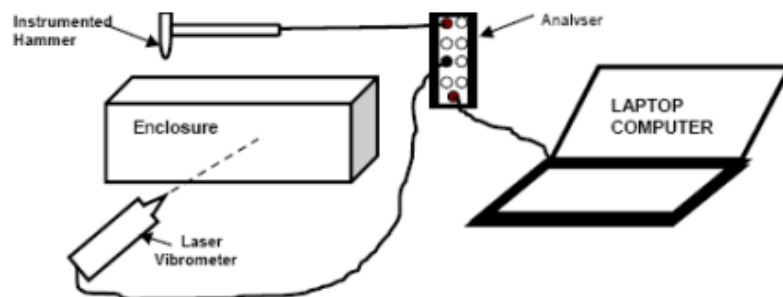


Figure 17. Schematic of the experimental set-up [20]

Next, the thin wall structure used for this study was a box made of steel measuring 900 x 600 x 750 [mm], having 1.5mm of wall thickness.

The response vibration measurement of the box under the impact load were obtained under the fixed base boundary condition of the structure.

As a result, the frequency response functions of the structure, as well as the sum frequency response function of every test, which intend to represent the dynamic properties of the structure, were calculated and plot (Figure 18).

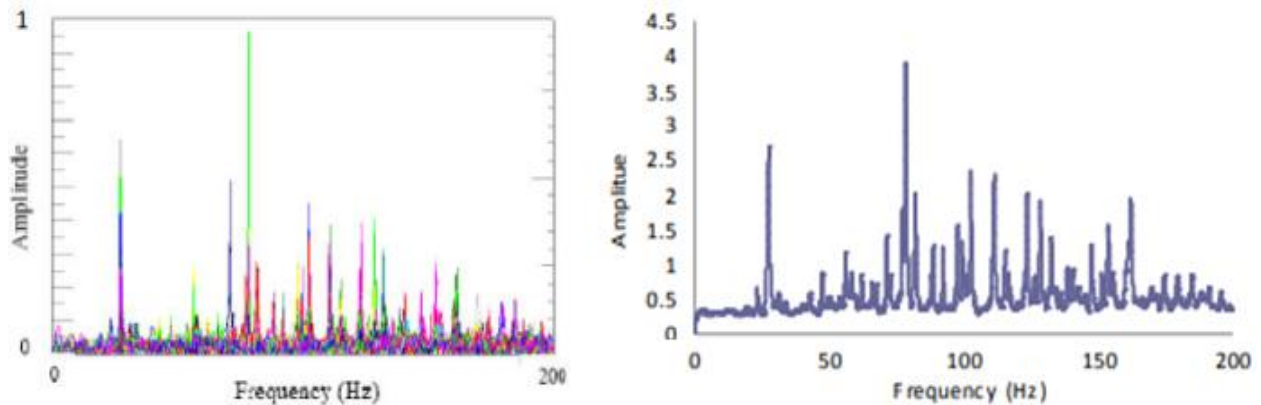


Figure 18. Overlay of frequency response functions (left) and sum frequency response function of a test (right) [20]

On the sequence, a sensitivity study was conducted to select the parameters for tuning the model. Based on this, the material density, the Young's modulus and other selected parameters were updated (Figure 19).

Element Type	Number of Elements	Shell thickness (mm)	Modulus of Elasticity (N/m ²)	Mass Density (kg/m ³)	Damping Constant	Material Type
Shell element with mid-side nodes	27600	1.6	2.1E+11	7850	0.02	Steel

Figure 19. Updated parameters at FE model [20]

Then, the FE updated model eigenfrequency results were calculated and compared with the results already obtained from the FE model before updating (Table 7).

Experimental Result (Hz)	Result before Updating (Hz)	Result after Updating (Hz)
16.99	15.83	17.27
19.45	17.5	19.1
23.47	21.36	23.31
26.07	23.37	25.5
27.48	26.29	28.69
31.63	31.01	33.83
33.97	32.0	34.91
40.25	37.01	40.38
42.55	40.17	43.83
46.34	42.51	46.38

Table 7. Experimental eigenfrequency results comparison against FE model results before and after parameters updating [20]

Also, the natural frequencies of the box obtained from the excitation response test and the finite element analysis results after the FE model updating were compared. The relation between them was plotted, and the R^2 correlation coefficient (a measure that describe the strength of the correlation between two quantities) was calculated (Figure 20).

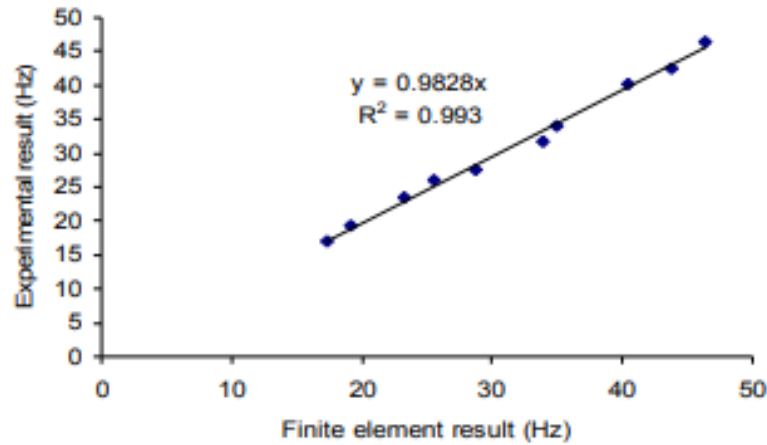


Figure 20. Correlation between experimental results and FE updated model [20]

As a result, the R^2 value of 0.99 implies a very good agreement between the updated numerical result and the experiment, meaning that a satisfactory correlation of approximately was obtained between the results from experiment and finite element updated simulation.

In [21], Rotondella V.; Merulla A.; Baldini A. and Mantovani S. presents a FE seam welding technique in which a modal correlation between the experimental data and the Finite Element (FE) modal analysis results from an automotive rear sub-frame is presented.

The prototype part (Figure 21) is all aluminum made, and have its different components jointed together through metal inert gas (MIG) welding, weighting in total 26.3 kg.



Figure 21. Aluminum rear sub-frame [21]

So, an Experimental Model Analysis (EMA) test was performed by doing an impact hammer test and acquiring the response in a set of accelerometers. so, the following frequency response functions (FRFs) (Figure 22, left) were calculated and plotted from the obtained data.

Then, the frequency poles were calculated, and the results presented in a so-called stabilization diagram (Figure 22, right).

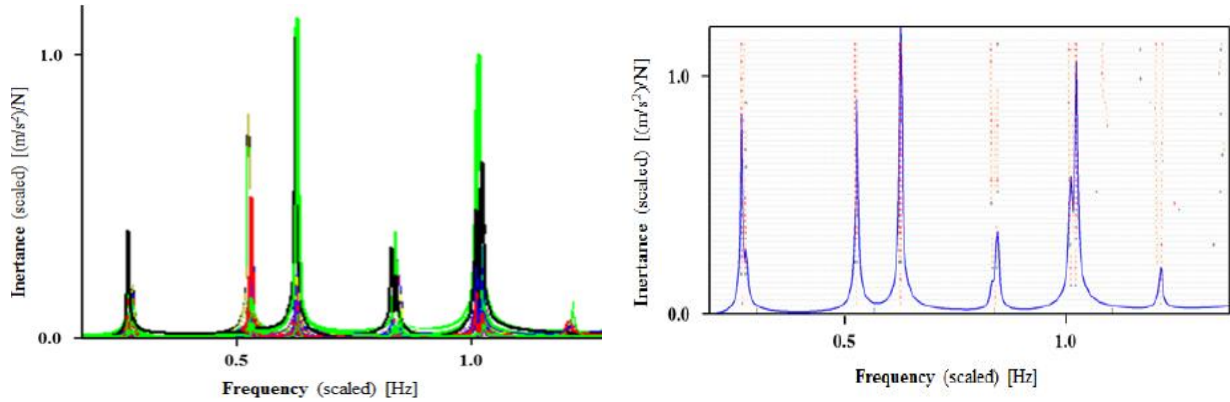


Figure 22. FRF plots from EMA test reference points (left) and EMA stabilization diagram (right) [21]

The interpretation of the stabilization diagram yields to a set of complex poles and participation factors, from which it was possible to retrieve the mode shape for each eigenfrequency. So, the natural frequencies of the sub-frame assembly (Table 8) were acquired from this.

Mode	Frequency (scaled) [Hz]	Damping Ratio (%)
1	0.244	0.20
2	0.254	0.35
3	0.519	0.04
4	0.623	0.08
5	0.834	0.08
6	0.849	0.03
7	1.023	0.09
8	1.037	0.07
9	1.236	0.13

Table 8. Natural frequencies of the sub-frame assembly EMA [21]

On the sequence, a Finite Element Analysis (FEA) model was built using a modified welding technique, which consists in extending the ACM2 welding nuggets along the welding length but keeping the same element properties previously used.

In this case, the connections elements followed a specific rule, having RBE2 elements attaching the welding seam to the sheet metal parts in the case there is a direct normal project, and RBE3 elements in the case there are only non-normal projections.

As a result, the following comparison (Table 9) was performed between the EMA and the FEA results in terms of percentual error, as well as considering the Modal Assurance Criterion correlation coefficient:

Mode Pair	EMA Scaled Freq. [Hz]	FEA Scaled Freq. [Hz]	Error (%)	MAC
1	0.244	0.244	0.060	0.90
2	0.254	0.252	1.055	0.89
3	0.519	0.503	3.076	0.84
4	0.623	0.612	1.677	0.96
5	0.834	0.817	2.016	0.55
6	0.849	0.833	1.934	0.76
7	1.023	1.000	2.221	0.73
8	1.037	1.019	1.709	0.73
9	1.236	1.234	0.134	0.82

Table 9. Summary of correlation between EMA and FEA results [21]

By this, it can be noticed that a good correlation was achieved. The percentage error was lower than 4 percent which evidences an agreement in terms of dynamic behavior prediction of the FE model.

Also, the values on the main diagonal of the MAC matrix confirms the good correlation in terms of modal shapes, once most values are greater than 0.7, which represents an acceptable correlation level.

3. MATERIALS AND METHODS

3.1. Experimental Test and Modeling Definition

The use of analytical simulation models to replace experimental tests is dependent on the ability that such models have to accurately represent the events to be measured and, therefore, calculate the results obtained from the analysis of these events.

So, the first condition for a correct assessment of the degree of correlation of an analytical model is the execution of experimental tests that can produce accurate results. For this reason, the experimental tests are performed under controlled conditions, in bench tests, and by experienced teams in the measurement and treatment of such parameters. In this way, the experimental test results are the reference data against which the simulation model are compared.

Then, the simulation models are build and analyzed through the use of an established technique, the finite element modeling analysis, in order to minimize the margins of error inherent to the process.

Furthermore, all results are compared using proven methods, whether they are mathematical, such as the Modal Assurance Criterion (MAC) and the Frequency Response Assurance Criterion (FRAC); analytical, such as the modal shapes and FRF curves comparison; or statistical, such as the Medium Absolute Error (MAE) and the Medium Squared Error (MSE).

The test results acquirement process, as well as the modeling build and analysis, and the results treatment are carefully discussed in the following sections.

3.2. Tests Performed on the Bench Test – Reference Data

3.2.1. Test Setup

Based on the vehicle manufacturer's experience, an experimental modal analysis of the physical structure of the vehicle body is performed on a bench modal (Figure 23, left) and FRF (Figure 23, right) test, as explained in the following paragraphs. From this test, a reference set of results is generated to support the correlation analysis of the FE models.

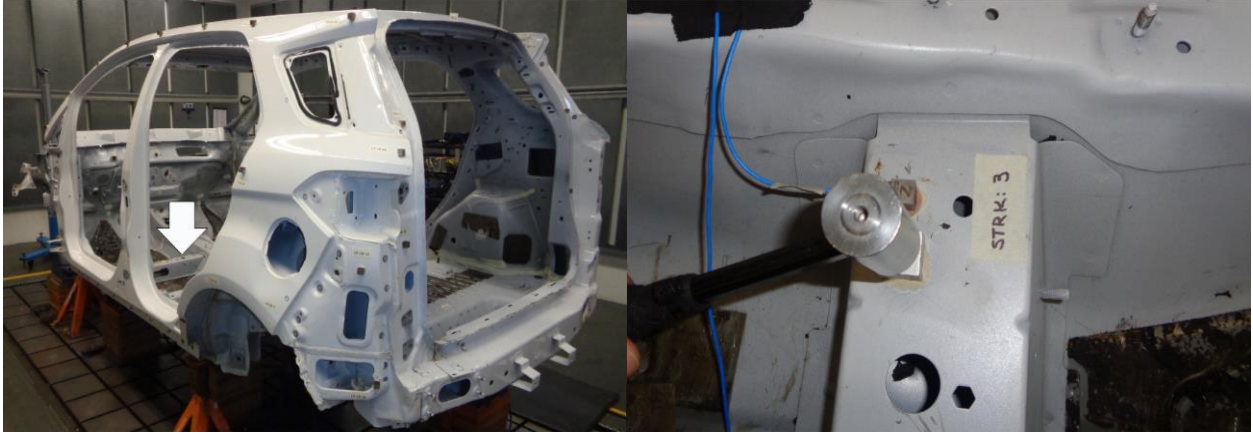


Figure 23. Experimental test body (left) and impact hammer load at front seat attachment (right).

Two electrodynamic shakers are used to excite the body structure with a time-domain burst random noise. In addition, the shape of the power spectral density (PSD) of the input signal is configured to concentrate the higher amount of energy within the frequency range from 10 to 60 Hz, where the main eigenmodes can usually be found.

So, the response of the vehicle body structure is acquired through a mesh of accelerometers (Figure 24) to identify the main body structure eigenmodes and gather their eigenfrequencies and damping ratios.

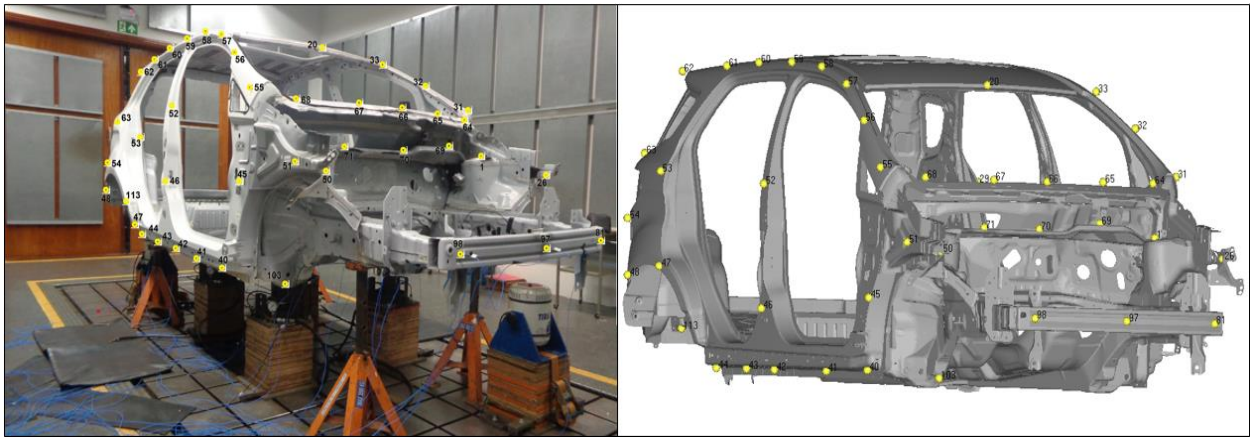


Figure 24. Experimental test body accelerometers (left) and FEM accelerometers representation (right).

Three sequential runs with fifty sample records per run are performed to cover all excitation and response points. In this way, the data analyses are more accurate and faster. Finally, this data is acquired to compute the eigenfrequencies and eigenmodes of the vehicle body structure.

An impact hammer is also used to perform an impact test by exciting, one at a time, the four shock tower attachment points of the vehicle body structure. The magnitude and the phase of the accelerations are measured at the driver seat attachment point (Figure 23, right) in the x, y and z-dir, by applying a 1.0 N pulse at each point on the x, y, and z directions (dir) separately (Figures 25 to 30).

An average of ten impacts per direction is performed to smooth the transfer functions and improve data accuracy even more. The fast Fourier transform (FFT) is computed to convert the acquired data from the time to frequency domains. In this way, the frequency response function (FRF) is obtained by exciting the four shock towers, one at a time, and acquiring the response for each point and direction.



Figure 25. Impact hammer x-dir load on front left and front right body structure shock tower.



Figure 26. Impact hammer y-dir load on front left and front right body structure shock tower.



Figure 27. Impact hammer z-dir load on front left and front right body structure shock tower.



Figure 28. Impact hammer x-dir load on rear left and rear right body structure shock tower.



Figure 29. Impact hammer y-dir load on rear left and rear right body structure shock tower.

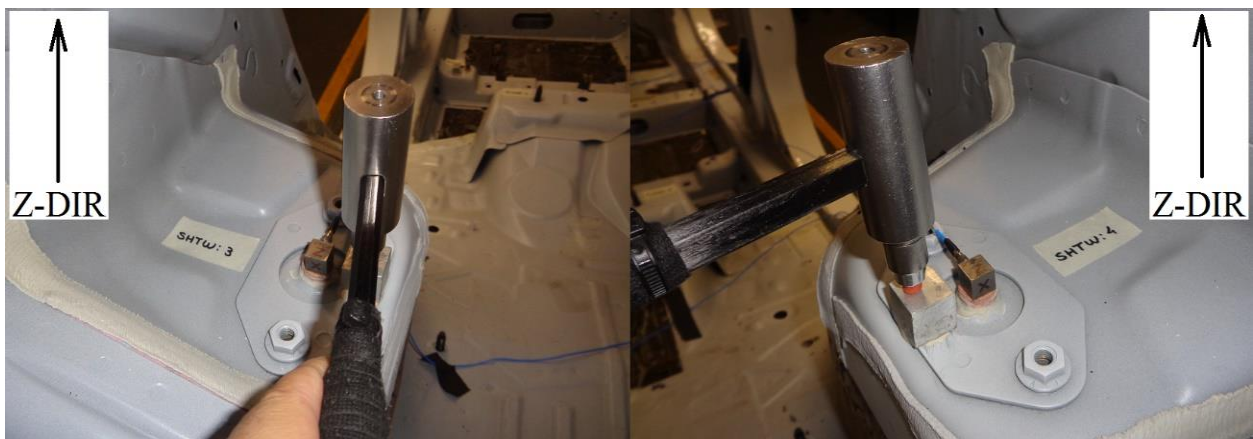


Figure 30. Impact hammer z-dir load on rear left and rear right body structure shock tower.

All equipment used for the experimental test measurement is described and shown in the Table 10. In the total, in addition to the responses in x, y and z directions already acquired at each point of the accelerometers meshing for the eigenfrequencies analysis, 72 FRF data sets (4 excitation points $\times$ 3 excitation

directions $\times$ 3 response point directions $\times$ 2 complex roots—magnitude and phase) are measured and acquired.

Equipment and specs	Qty	Figure
Measurement Arm Faro 8-Axis Edge • Measurement range: 3.7m • Accuracy: $\pm 25\mu\text{m}$ • Repeatability: $25\mu\text{m}$, 2σ • Stand-off: 115mm	1	
Modal Tuned Impulse Hammer PCB086C03 • Force range: 0 to 2224N • Sensitivity: 2.25 mV.N^{-1} ($\pm 15\%$) • Measurement Range: $\pm 2224\text{N}$ pk • Hammer Mass: 0.16kg	1	
Triaxial Accelerometers PCB J356B07 • Frequency range: 2Hz to 5kHz • Sensitivity: ($\pm 10\%$) 100mV.g^{-1} • Measurement Range: $\pm 50\text{g}$ pk • Electrical Connector: 1/4-28 4-Pin	152	
Air Springs Goodyear 169-XY • Minimum height: 95mm • Maximum height: 275mm • Diameter: 265mm • Maximum load: 150Kg	4	
Shakers TIRA S 5220-M • Rated peak force 1000/650N • Frequency range 5 - 5000Hz • Maximum velocity 1.5m.s^{-1} • Main resonance frequency $>3000\text{Hz}$	2	

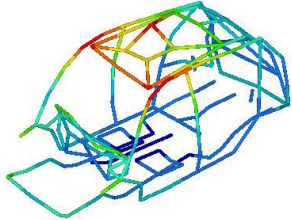
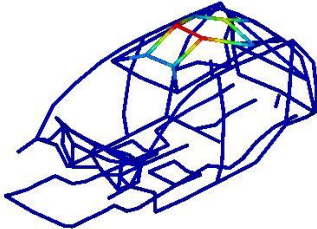
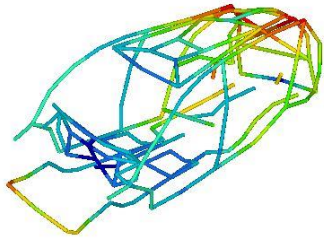
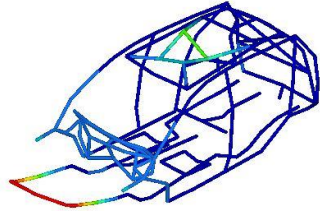
LMS Scadas Data Analysis Set (128 Channels)	
• Bandwidth/Resolution 128/0.25Hz • Modal FRF algorithm: PolyMAX • Acquisition Control Mode: Burst random • Uniform reference and response windows	1



Table 10. Experimental test measurement equipment specifications and figures.

3.2.2. Eigenfrequencies

Table 11 gathers the eigenfrequencies and damping ratios of the main eigenmodes of the body structure found in the bench test, from 10 to 60 Hz.

Mode	Frequency [Hz]	Damping c/co []	Mode Shape	Mode Picture
1	33.38	0.00513	Windshield Torsion	
2	33.38	0.00390	1st Moonroof Bending	
3	39.23	0.00567	Global Torsion	
4	44.71	0.00228	Global Lateral Bending	

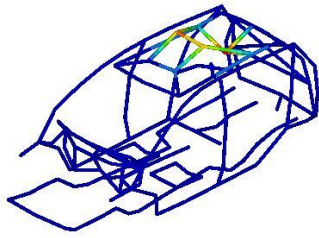
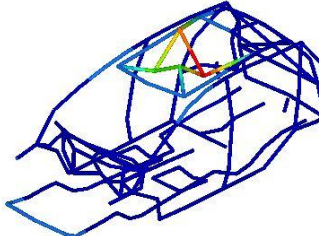
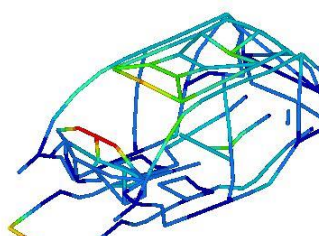
5	46.34	0.00642	2nd Moonroof Bending	
6	49.34	0.00344	Moonroof Torsion	
7	53.73	0.00453	Global Vertical Bending	

Table 11. Eigenfrequencies and damping ratios of the vehicle body structure from the bench test.

3.3. Reference FE Model of the Vehicle Body Structure

3.3.1. FE Model Modal Consistence Analysis

As a starting point, an initial FE model (named as V1 Linear FE) of the vehicle body structure (Figure 31) is build. The Hyperworks software pack is used to support the FE model building (Hypermesh), the FE analysis solving (Optistruct), the post-processing FRF graphics (Hypergraphics), visualization (Hyperview), MAC (Hyperview NVH analysis module and Compose) and FRAC (Compose) calculations. MS Excel is used to plot additional graphics other than FRFs.

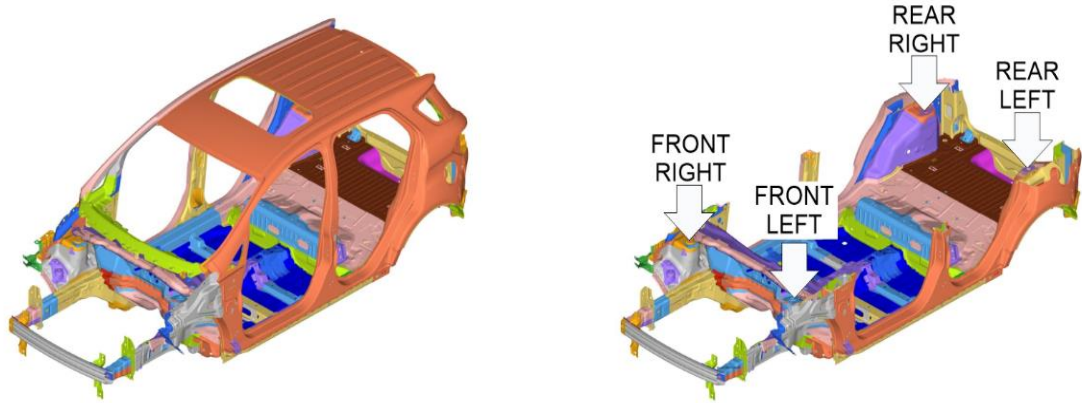


Figure 31. Linear FE model of vehicle body structure (left) and body excitation points (right).

The V1 Linear FE model is meshed based on the vehicle manufacturer's experience, as summarized below. A 5 mm meshing size is used with minimum size of 2 mm for refinement at complex design regions as the edges, flanges, beads, and attachment points. TRIA3 and QUAD4 2D shell elements are used to mesh stamped parts such as panels and structural beams; HEXA8 3D + RBE3 1D elements to represent spot welds and adhesive connections; RBE2 1D elements to represent bolts and nuts, and CBUSH 1D spring-damper elements to the adjusted model tuning.

Linear material properties are also used to represent the stamped parts, made of steel (Young's modulus = 210.0 GPa, Poisson's ratio = 0.30 and density = 7900 Kg/m³), as well as the polymeric adhesive material (Young's modulus = 600 GPa, Poisson's ratio = 0.42 and density = 1190 Kg/m³). Spot-welded points are represented by ACM weld elements.

A free-free modal analysis (Normal Modes solver solution) is performed with the V1 Linear FE model to get the same mode shapes [22] as those obtained in the modal bench test (Figure 23, left).

The hammer impact test (Frequency Response Modal solver solution) is simulated using a frequency response analysis to find the FRFs from the four shock tower attachments, considering the six degrees of freedom [23], like in the FRF bench test (Figure 23, right). The eigenvalues are extracted using the Lanczos' method, considering a structural critical damping ratio of 0.008 [24,25] from 0 to 100 Hz.

The Modal Assurance Criterion (MAC) metric is used to analyze the modal consistence of the V1 Linear FE model and the Frequency Response Assurance Criterion (FRAC) metric is used to measure the differences between the bench test and the FE model results.

The MAC [26–28] is a normalized single-value metric that estimates the consistency between eigenvectors from diverse sources. It is computed to assess the accuracy of the modal behavior of the V1 Linear FE

model. The eigenvectors found from this model are contrasted with the bench test eigenvectors, focusing on the frequency range of concern, from 10 to 60 Hz. The MAC is computed as follows [28]:

$$MAC_{(a,x)} = \frac{\left| \sum_{j=1}^{N_f} \{\varphi_a\}_j \{\varphi_x\}_j \right|}{\left(\sum_{j=1}^{N_f} \{\varphi_a\}_j^2 \right) \left(\sum_{j=1}^{N_f} \{\varphi_x\}_j^2 \right)} \quad (1)$$

where the eigenvectors $\{\varphi_a\}$, extracted from the V1 Linear FE model analysis, are compared with the reference eigenvectors $\{\varphi_x\}$, extracted from the bench test data. N_f variable refers to the mode number, in ascending order.

The following Optistruct calculation deck (Table 12) is used to extract the eigenvectors from the V1 Linear FE model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
SOL 103	Free-free modal solution
CEND	Designates the end of the Executive Control Section
ECHO = NONE	Print bulk data on response option
METHOD (STRUCTURE) = 1	Selects the real eigenvalue extraction parameters for the structural analysis
DISPLACEMENT (PUNCH, PRINT) = 1	Requests the form and type of displacement response
BEGIN BULK	Designates the start of the Bulk Data Section
PARAM, AUTOSPC, YES	Requests that stiffness singularities and near singularities automatically constrains via single or multipoint constraints
PARAM, GRDPNT, 0	Defines the origin of the basic coordinate system as the weight reference grid point
PARAM, PRGPST, NO	Suppresses the automatic printing of the flutter summary in flutter analysis
EIGRL, 1, , 100.0 , , ,	Defines data needed to perform real eigenvalue (vibration or buckling) analysis with the Lanczos method, in this case, from 0Hz to 100Hz
ENDDATA	Designates the end of the Bulk Data Section

Table 12. Optistruct deck used to extract the eigenvectors from the V1 Linear FE model

Then, a Compose script is built to calculate the MAC values. It is detailed in the Appendix I.

The (FRAC) [27–30] is a frequency-dependent normalized single-value metric that estimates the correlation between two FRFs with the same excitation and response points. It is computed to assess the accuracy of the transfer functions found with the V1 Linear FE model. The FRFs found from this model are contrasted with those found in the bench test, restricting to the frequency range of concern, from 25 to 50 Hz, where the FRF peaks are most representative. The FRAC is computed as follows [27]:

$$FRAC_{(a,x)} = \frac{\left| \sum_{j=1}^{N_f} (\mathbf{H}_a(\omega_j)^H \cdot \mathbf{H}_x(\omega_j)) \right|^2}{\left[\sum_{j=1}^{N_f} (\mathbf{H}_a(\omega_j)^H \cdot \mathbf{H}_a(\omega_j)) \right] \left[\sum_{j=1}^{N_f} (\mathbf{H}_x(\omega_j)^H \cdot \mathbf{H}_x(\omega_j)) \right]} \quad (2)$$

where the FRFs $\mathbf{H}_a$, extracted from the V1 Linear FE model, is compared with the reference FRFs $\mathbf{H}_x$, extracted from the bench test data. Both $\mathbf{H}_a$ and $\mathbf{H}_x$ are complex functions. The superscript $\mathbf{H}$ refers to the Hermitian, which is the transpose of the complex conjugate. N_f and ω refer to the mode number and to the frequency value in ascending order.

The following Optistruct calculation deck (Table 13) is used to calculate the FRF curves of the V1 Linear FE model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
SOL 111	Frequency response function (FRF) solution
CEND	Designates the end of the Executive Control Section
ECHO = NONE	Print bulk data on response option
METHOD (STRUCTURE) = 1	Selects the real eigenvalue extraction parameters for the structural analysis
SDAMP (STRUCTURE) = 101	Selects the damping parameters for the structural analysis
FREQ = 3	Defines the set of frequencies to be used in the frequency response problem solution
SUBCASE 1	Delimits and identifies the first subcase
LABEL = 1 X	Defines the first subcase label, on this case exciting at the hard point #1 in X direction
DLOAD = 201	Defines the subcase DLOAD parameter
SET 1 = 100	Defines the subcase set of analysis
VELOCITY (SORT2, PUNCH, PHASE) = 1	Defines the subcase output result parameter reference

:	Subcases will continue to use this same format from the first until the last subcase
SUBCASE 12	Delimits and identifies the last subcase
LABEL = 4 Z	Defines the first subcase label, on this case exciting at the hard point #4 in Z direction
DLOAD = 212	Defines the subcase DLOAD parameter
SET 12 = 100	Defines the subcase set of analysis
VELOCITY (SORT2, PUNCH, PHASE)=12	Defines the subcase output result parameter reference
BEGIN BULK	Designates the start of the Bulk Data Section
DAREA 1 1 1 1.0	Defines scale (area) factors for dynamic loads, in this case an 1N pulse applied to the hard point #1 in the X direction
RLOAD2 201 1 0 0 4	Defines a frequency-dependent dynamic excitation of the form
:	The excitations will continue to use this same format from the first until the last subcase
DAREA 12 4 3 1.0	Defines scale (area) factors for dynamic loads, in this case an 1N pulse applied to the hard point #4 in the Z direction
RLOAD2 212 12 0 0 4	Defines a frequency-dependent dynamic excitation of the form
PARAM, AUTOSPC, YES	Requests that stiffness singularities and near singularities be automatically constrained via single or multipoint constraints
PARAM, GRDPNT, 0	Defines the origin of the basic coordinate system as the weight reference grid point
PARAM, PRGPST, NO	Suppresses the automatic printing of the flutter summary in flutter analysis
TABLED1 4 LINEAR LINEAR +TABLE +TABLE 0.0 1.0 150.0 1.0 ENDT	Defines a tabular function for use in generating the frequency-dependent dynamic loads and its sequential continuation line
TABDMP1 101 G +DAMP1 +DAMP1 0.0 0.016 150.0 0.016 ENDT	Defines modal damping as a tabular function of natural frequency and its sequential continuation line
FREQ1 32.0 1.0 48	Defines a set of frequencies to be used in the solution of frequency response problems by specification of a starting frequency, frequency increment, and the number of

								increments desired, in this case, from 2Hz to 50Hz at each 1Hz
								Defines data needed to perform real eigenvalue (vibration or buckling) analysis with the Lanczos method, in this case, from 0Hz to 100Hz
								Specifies the degrees-of-freedom where unit loads are to be applied to obtain static solutions for use in residual vector computations
								Designates the end of the Bulk Data Section

Table 13. Optistruct deck used to the FRF curves of the V1 Linear FE model

Then, a Compose script is built to calculate the FRAC values. It is detailed in the Appendix II.

3.3.2. Initial Model V1 MAC Matrix

Table 14 gathers the MAC between the V1 Linear FE model of the vehicle body structure and the experimental test modes, the first obtained from the modal analysis simulation of the V1 Linear FE model and the second from the hammer impact bench test.

		BENCH TEST EIGENFREQUENCIES [Hz]						
		33.4	33.4	39.2	44.7	46.3	49.3	53.7
FE MODEL V1 EIGENFREQUENCIES [Hz]	31.3	0.02	0.99	0.00	0.00	0.10	0.00	0.00
	36.2	0.96	0.00	0.01	0.01	0.00	0.00	0.00
	37.5	0.01	0.00	0.96	0.01	0.00	0.00	0.00
	43.2	0.00	0.07	0.00	0.00	0.98	0.01	0.00
	47.3	0.00	0.00	0.01	0.67	0.00	0.00	0.00
	50.2	0.00	0.01	0.00	0.00	0.01	0.84	0.00
	53.0	0.00	0.00	0.00	0.00	0.00	0.16	0.60

Table 14. MAC results for experimental test vs FE linear model V1.

These results show that the main diagonal terms of the MAC matrix are lower than 0.9. It means the modal results of the V1 Linear FE model are not consistent with the experimental results [28].

The experimental test and FE linear model Seat Track curves, shown at Appendix III, are also calculated and plotted in order to check if some unusual FRF behavior raised from the model update, but the FRAC calculation is not performed as the modal convergence is not good enough to consider the linear model consistent with the test results.

3.3.3. FE Model Modal Consistence Improvement

3.3.3.1. V2 Improvement Model

Considering that the V1 Linear FE model do not achieve the minimum modal consistence required to perform a FRAC analysis, a refinement process is necessary to improve the FE modeling representation and, with this, increase the FRF results accuracy.

The acoustic deadeners representation is removed from the V1 Linear FE model as the prototype evaluated do not have these parts along experimental test execution, creating the V2 Linear FE model (Figure 32). They represent an important amount of mass at the structure panels, as well as high impact over the panels damping.

Also, a set of 12 spot welds that are misrepresented (represented as simple node-to-node rigid elements) in the V1 Linear FE model are add to the V2 Linear FE model at the cowl top joint area (Figure 32). In this case, despite of the small number of spot welds misrepresented, it is important to assure its correct representation as this joint is usually very sensitive.

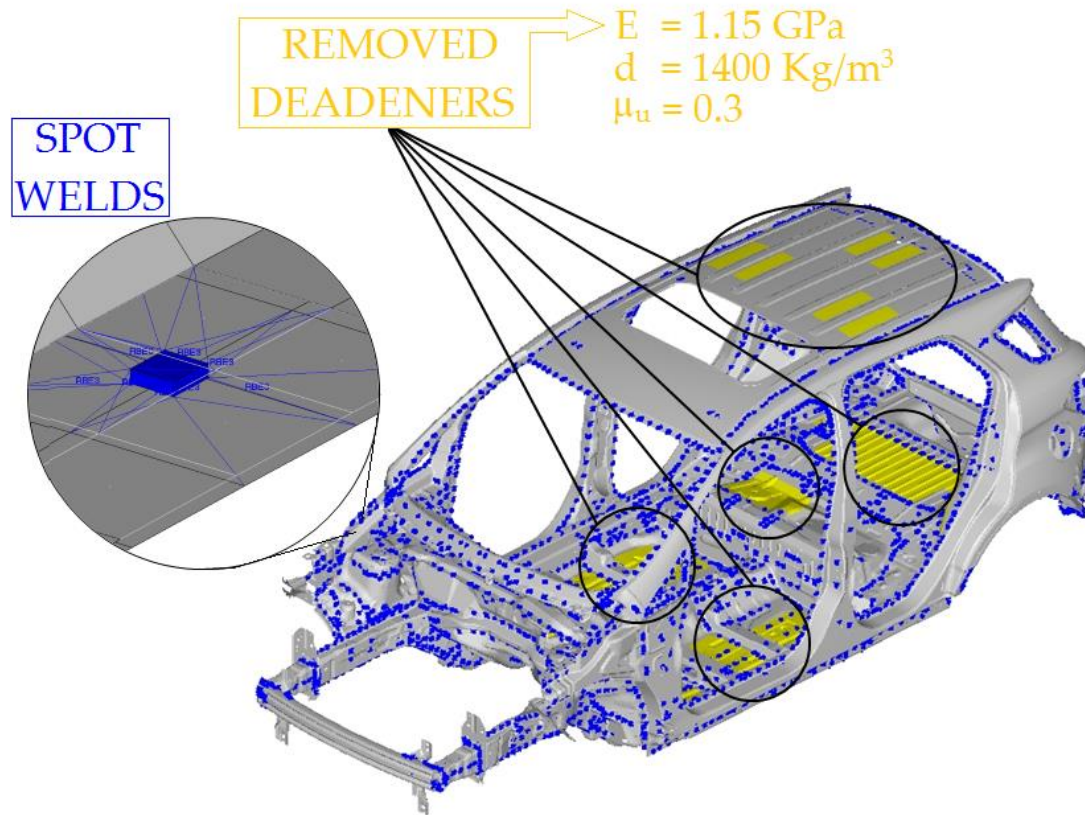


Figure 32. Linear FE model V2 of the vehicle body structure with spot-welds and deadeners.

So, a new set of analysis considering the Point Mobility and Seat track FRFs are performed with the updated Linear FE model V2.

As a result, some FRF changes are noticed when visually comparing the V1 and V2 Linear FE model FRF curves against the experimental test. So, the most relevant changes are carefully analyzed in order to have a clear understanding about them.

For this, it is necessary to identify whether such differences are found at most of the points in the analysis (global effect) or if they are isolated events (local effect). In the case that a global effect is identified, it is still necessary to identify how it behaves considering the entire structure as a single body.

Furthermore, in both cases, it is important to understand the relation between the root cause and the effect given between the refinements proposed in the simulation model and the results obtained.

This verification is performed for each step of the simulation model development process in order to ensure that all proposed improvements to the FE model of the vehicle structure are consistent and widely understood.

Initially, the experimental test and the FE models FRFs are analyzed. Comparing the V1 and V2 Linear FE model FRF curves with the experimental test, a 2Hz frequency improvement is noticed (Figure 33) at 31Hz in x-dir response (left) and at 30Hz in z-dir response (right) when exciting the front left shock tower in the x-dir. The amplitude of the V2 FRF peaks are also closer to the experimental test amplitude level then V1.

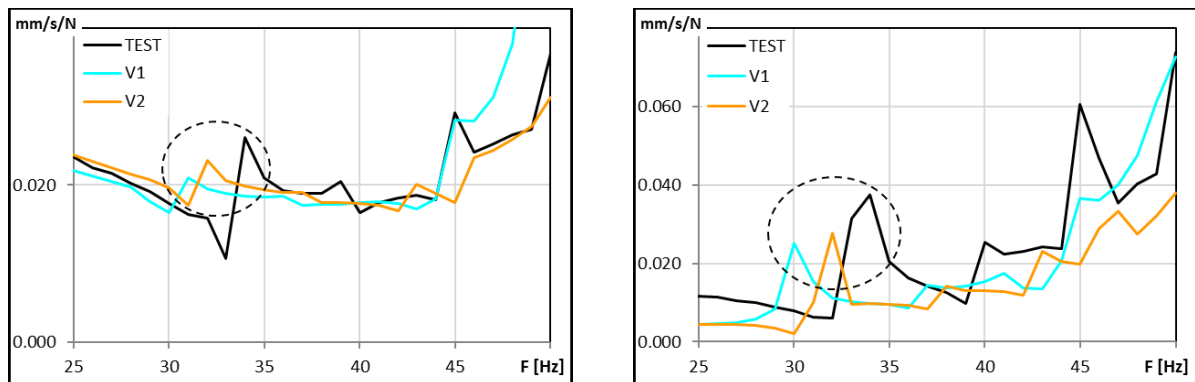


Figure 33. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary x-dir excitation at front left shock tower).

A similar frequency improvement is found at x-dir and z-dir responses when exciting the front left shock tower in y-dir (Figure 34) and in z-dir (Figure 35). The amplitude level is also closer to the experimental test in both cases.

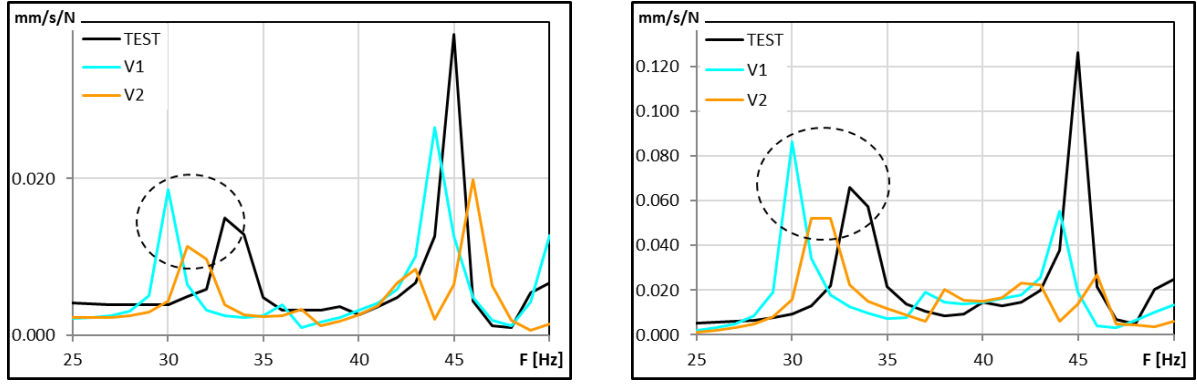


Figure 34. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary y-dir excitation at front left shock tower).

Furthermore, a small improvement is also noticed at 36Hz in the z-dir case (Figure 35), considering both frequency and amplitude levels.

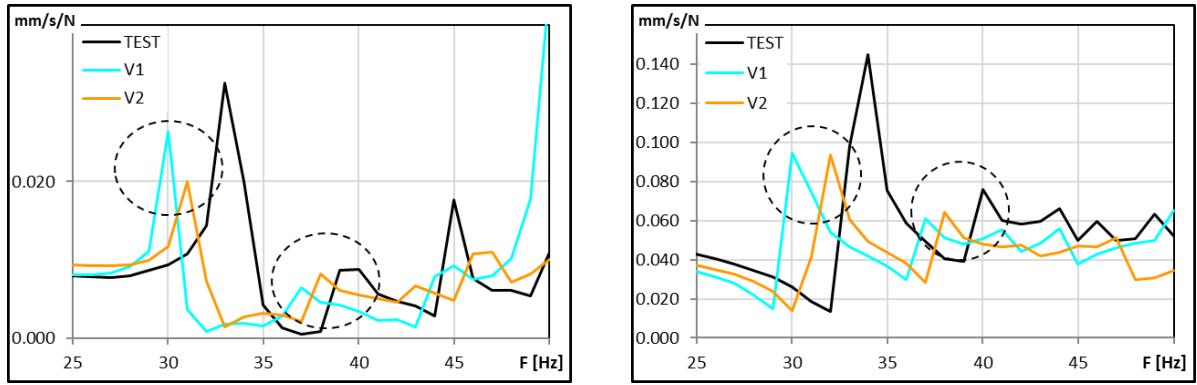


Figure 35. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary z-dir excitation at front left shock tower).

The analysis of the FRFs from the front right shock tower (Figures 36 to 38) shows similar improvements when compared with the front left shock tower (Figures 33 to 35), reflecting the lateral (y-dir) symmetry of the structure at this region.

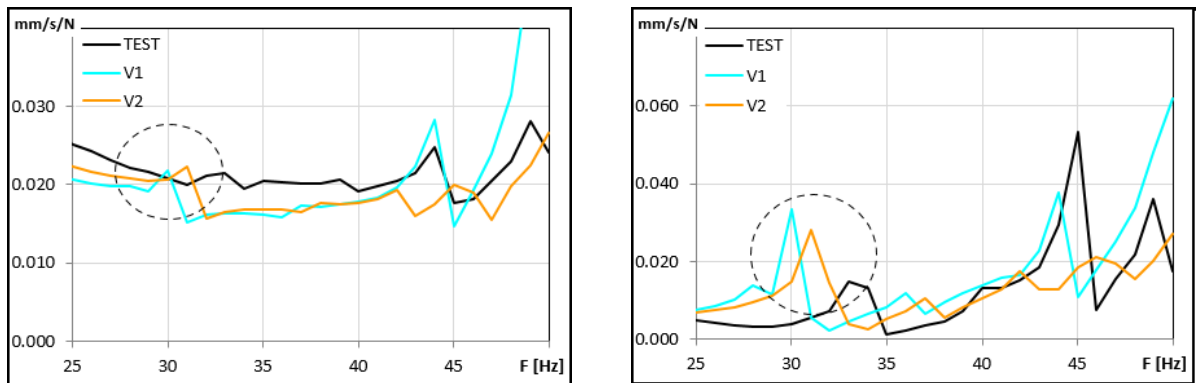


Figure 36. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary x-dir excitation at front right shock tower).

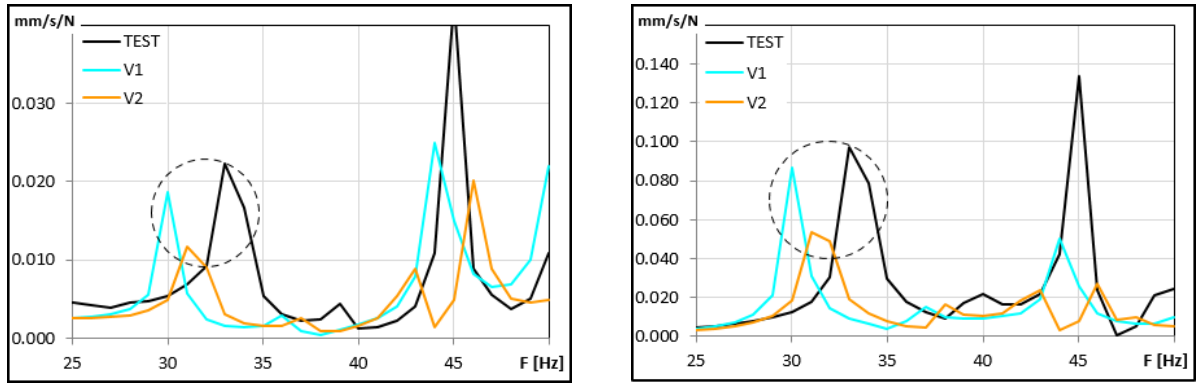


Figure 37. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary y-dir excitation at front right shock tower).

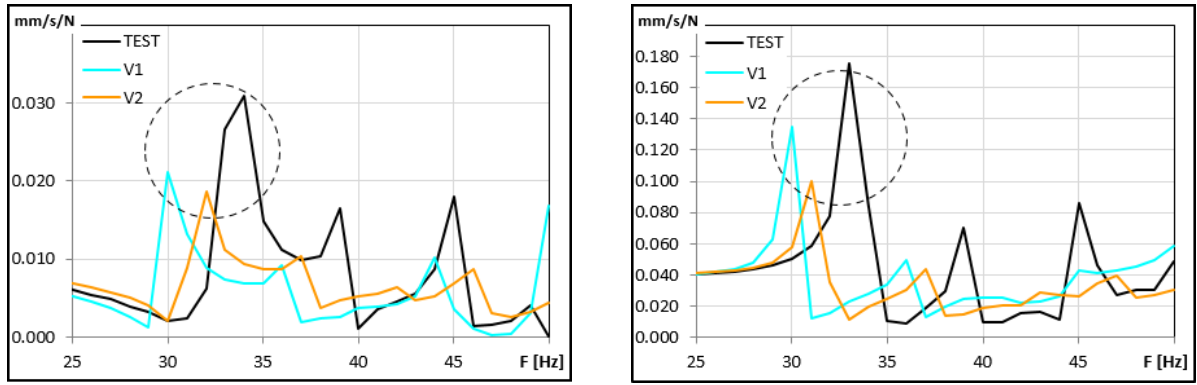


Figure 38. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary z-dir excitation at front right shock tower).

In contrast, the analysis of the FRFs from the rear left and right shock towers shows similar improvements as noticed from the front shock towers in x-dir (Figures 39 and 40).

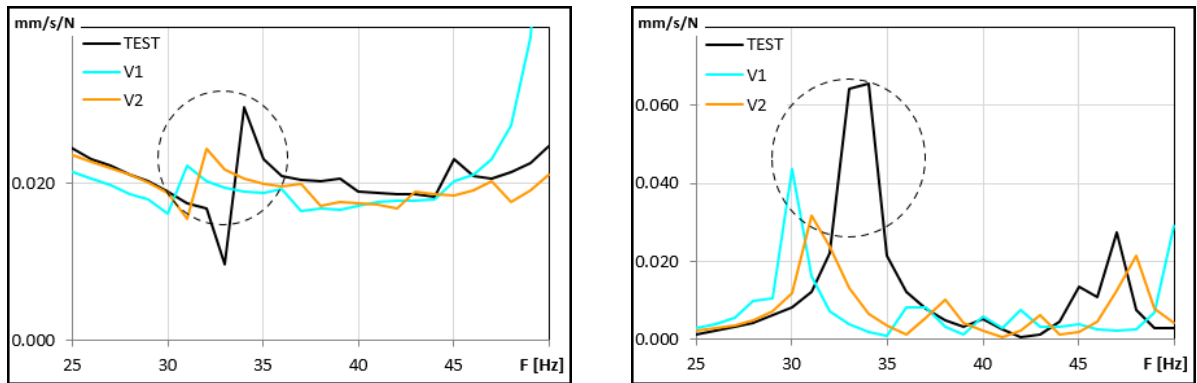


Figure 39. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary x-dir excitation at rear left shock tower).

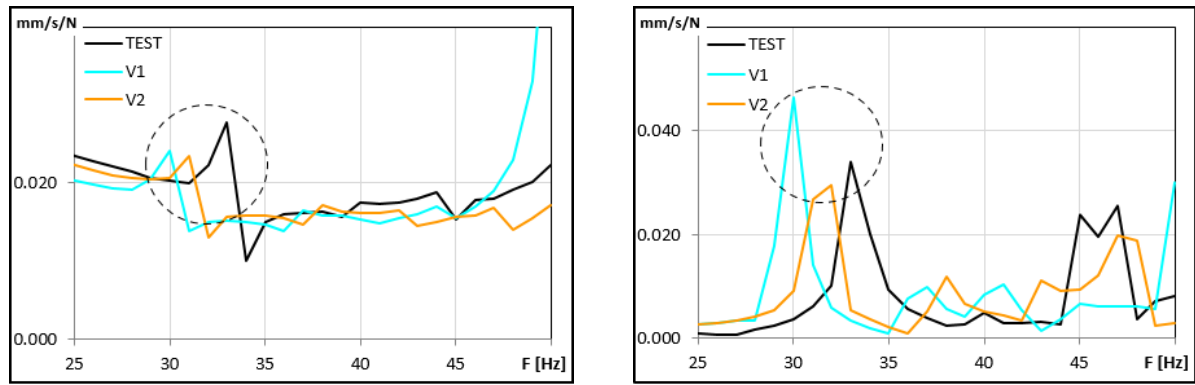


Figure 40. Experimental test *vs* FE linear model V1 *vs* V2 FRFs (x-dir (left) and z-dir (right) responses to unitary x-dir excitation at rear right shock tower).

In this case, the behavior of the first mode (windshield torsion mode), concentrated in the front part of the vehicle structure, explains the reason why the high similarity level between the front shock tower FRFs cannot be found with the same strength level when comparing it against the FRFs acquired from the rear shock towers.

In addition to this, the point mobility curves acquired from the front shock towers (Figure 41) and rear shock towers (Figure 42) of the structure, also confirm the correlation improvements already noticed around 32Hz on the transfer functions at both sides of the front shock towers and also around 36Hz at the rear shock towers.

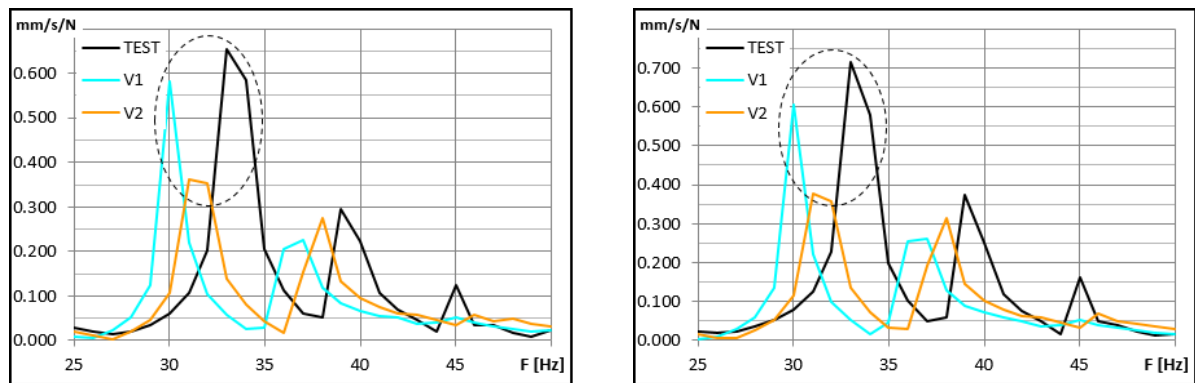


Figure 41. Experimental test *vs* FE linear model V1 *vs* V2 point mobility z-dir at front left shock tower (left) and z-dir at front right shock tower (right).

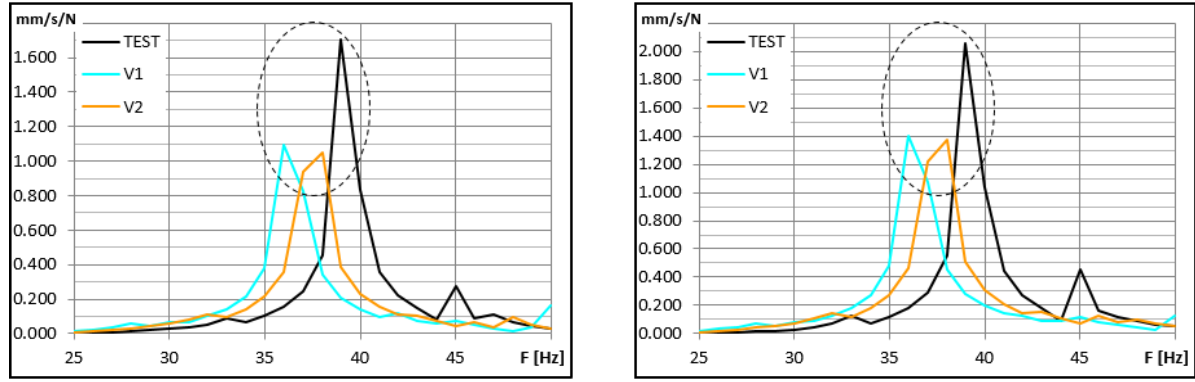


Figure 42. Experimental test *vs* FE linear model V1 *vs* V2 point mobility z-dir at rear left shock tower (left) and z-dir at rear right shock tower (right).

The comparison between the animation of the experimental test first natural mode (Figure 43, left) against V1 (Figure 43, center) and V2 (Figure 43, right) FEM analyses confirms that the windshield torsion mode is the main contributor mode at both models.

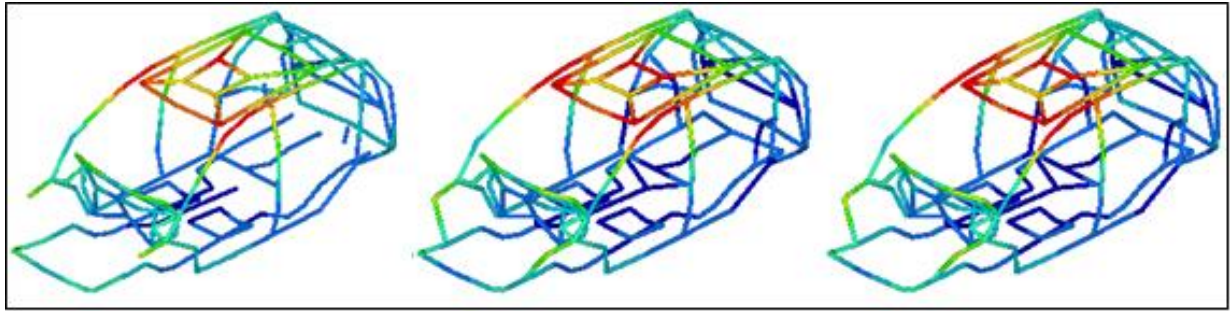


Figure 43. Modal shape of the first natural mode at the experimental test (left), FE linear model V1 (center) and V2 (right).

At this point, as the FRF curves have been sufficiently analyzed, the MAC coefficients have to be also evaluated. The comparison between the MAC tables from V1 and V2 models (Table 15) highlights how the first and second structural modes are swapped in V1 model, not following the main diagonal alignment. On the other hand, these structural modes are now properly aligned in the V2 model, following the main diagonal alignment.

		BENCH TEST EIGENFREQUENCIES [Hz]							BENCH TEST EIGENFREQUENCIES [Hz]								
		33.4	33.4	39.2	44.7	46.3	49.3	53.7	33.4	33.4	39.2	44.7	46.3	49.3	53.7		
FE MODEL V1 EIGENFREQUENCIES [Hz]	31.3	0.02	0.99	0.00	0.00	0.10	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	FE MODEL V2 EIGENFREQUENCIES [Hz]	31.3
	36.2	0.96	0.00	0.01	0.01	0.00	0.00	0.00	0.02	0.98	0.00	0.00	0.14	0.00	0.00		36.2
	37.5	0.01	0.00	0.96	0.01	0.00	0.00	0.00	0.00	0.00	0.97	0.00	0.00	0.00	0.00		37.5
	43.2	0.00	0.07	0.00	0.00	0.98	0.01	0.00	0.00	0.00	0.02	0.77	0.00	0.18	0.00		43.2
	47.3	0.00	0.00	0.01	0.67	0.00	0.00	0.00	0.01	0.00	0.00	0.22	0.01	0.77	0.00		47.3
	50.2	0.00	0.01	0.00	0.00	0.01	0.84	0.00	0.00	0.02	0.00	0.00	0.94	0.02	0.00		50.2
	53.0	0.00	0.00	0.00	0.00	0.00	0.16	0.60	0.00	0.00	0.00	0.00	0.01	0.00	0.62		53.0

Table 15. First and second modes highlighted at MAC tables from V1 (left) and V2 (right) models

This modal correlation improvement shown at the MAC analysis corroborates the improvements already noticed at the FRFs analysis between the models V1 and V2 around the first natural mode frequency (31.3Hz), moving the FRF peaks 2Hz closer to the experimental test when exciting the front left and right shock towers in x, y and z-dir.

Furthermore, it justifies the FRF improvement that is noticed around the second natural mode frequency (36.2Hz) when exciting the left side and also the right side front shock towers in z-dir (Figure 35).

The plots regarding all the experimental test excitations and responses of V1 model are shown in Appendix III.

So, as the main diagonal terms of the MAC matrix (Table 15) are still lower than 0.9, the FRAC calculation should not be processed, once the modal convergence is still not good enough to consider the linear model consistent.

However, considering that an important improvement in the 4th mode is noticed comparing the V1 x V2 MAC tables, the V2 Linear FE model is taken as the base from which the modeling refinement should be continued.

3.3.3.2. V3 Improvement Model

As a next step of refinement, the bolts and nuts representations, as well as their masses, are add to the model, creating the V3 Linear FE model (Figure 44).

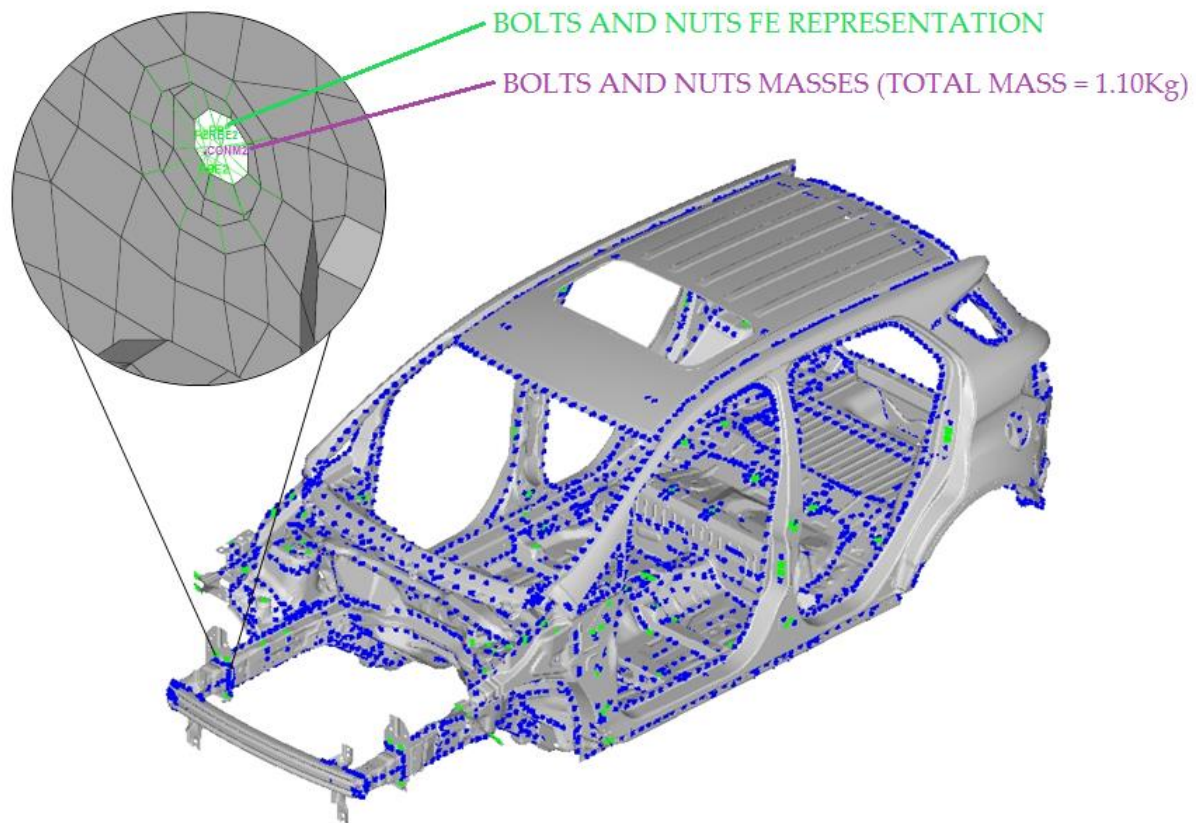


Figure 44. Linear FE model V3 of the vehicle body structure with bolts and nuts.

So, a new set of analysis considering the Point Mobility and Seat track FRFs are performed with the updated Linear FE model V3.

The FRFs analysis shows that, in this case, only a few changes are noticed between the V2 and V3 models (Figures 45 and 46), considering exciting the body structure at the front and rear shock tower hard points, and at both left and right sides.

This small effect over the FRFs is justified by the purpose of the bolted connections. The spot welds, already present in the model, are the responsible for keeping the structural integrity of the whole sheet metal body structure.

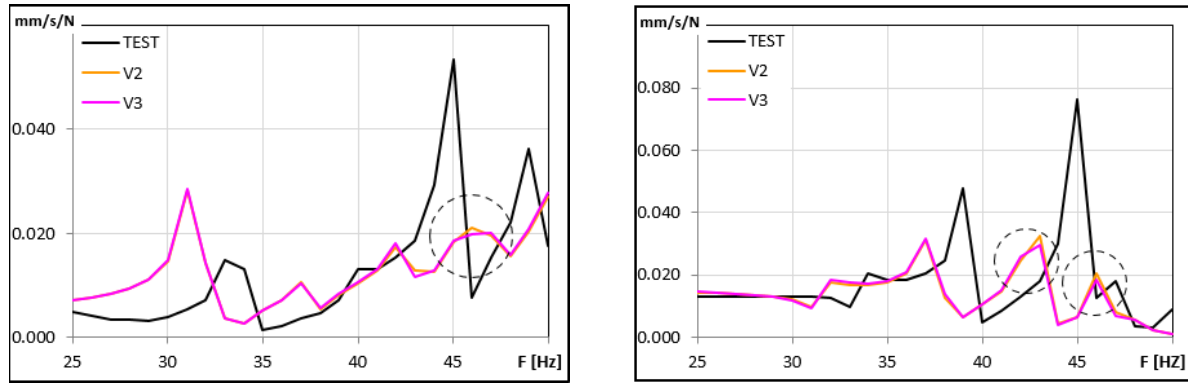


Figure 45. Experimental test *vs* FE linear model V2 *vs* V3 FRFs (z-dir response to unitary x-dir excitation at front right shock tower (left) and y-dir response to unitary y-dir excitation at rear left shock tower (right)).

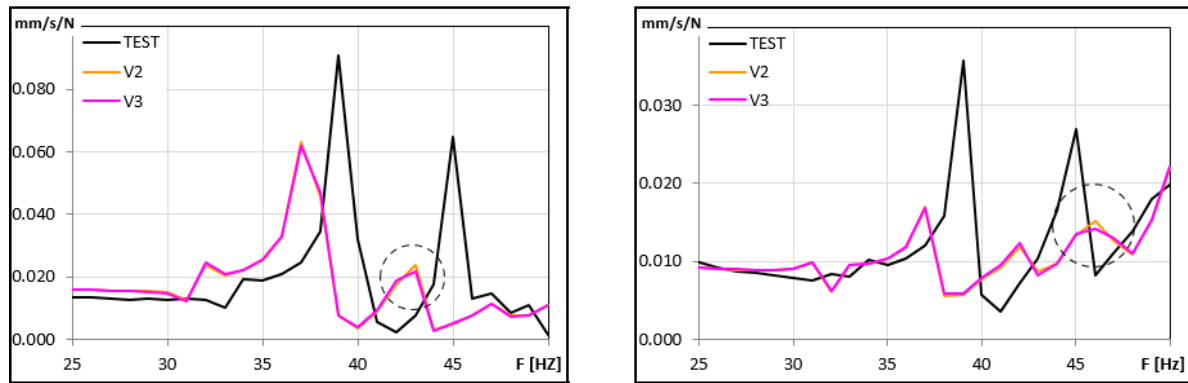


Figure 46. Experimental test *vs* FE linear model V2 *vs* V3 FRFs (y-dir response to unitary y-dir excitation at rear right shock tower (left) and x-dir response to unitary z-dir excitation at rear right shock tower (right)).

On the other hand, the bolted connections function is to keep body sub-structures (closures, subframes and body brackets), chassis components (reservoirs, suspension, and electric modules) and powertrain (engine and transmission) components properly attached to the body structure.

So, as any sub-structure or component is attached to the body structure, despite the bolts are present, they do not add considerable global stiffness to the body structure. The bolts also slightly affect the mass distribution of the body structure once they are not concentrated at any specific area.

The point mobility curves shows that the stiffness at the four shock towers are slightly affected alto. Only small improvements (Figure 47) around 46Hz at the front left shock tower and around 39Hz at the rear left shock tower can be noticed.

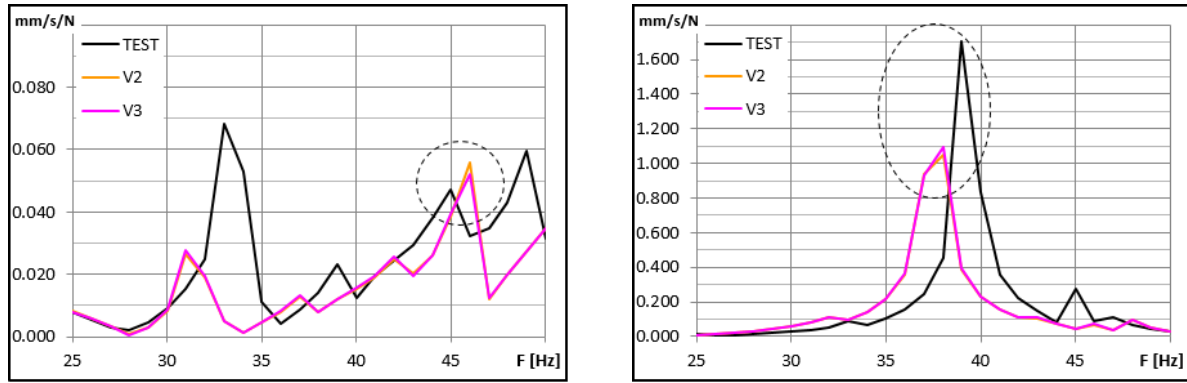


Figure 47. Experimental test *vs* FE linear model V2 *vs* V3 point mobility x-dir at front left shock tower (left) and x-dir at rear left shock tower (right).

The comparison between the animation of the experimental test global lateral bending mode (Figure 48, left) with V2 (Figure 48, center) and V3 (Figure 48, right) FEM results shows that, despite of the improvement found, V2 and V3 models shows similar modal behavior. The same conclusion can be extended to the other modes affected by V3 updates.

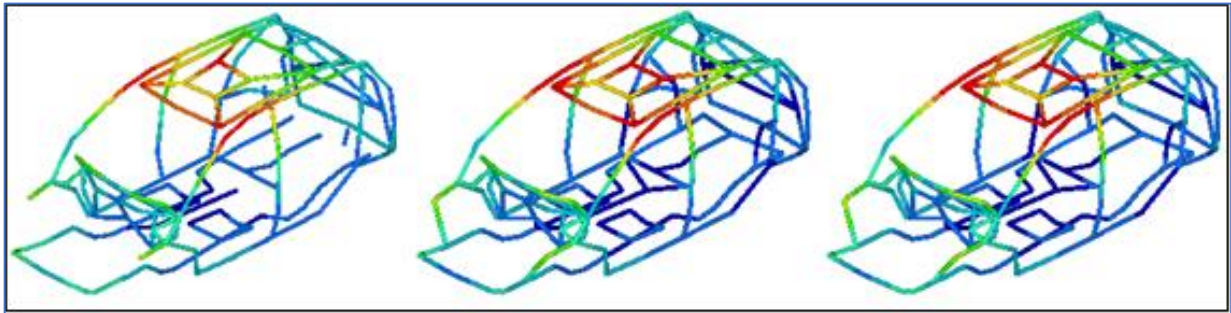


Figure 48. Modal shape of the first natural mode at the experimental test (left), FE linear model V2 (left) and V3 (right).

As a result, the modal behavior of the body structure is only slightly affected. A comparison between V2 and V3 MAC tables (Table 16) shows the small improvements in the modes between 39Hz and 50Hz.

		BENCH TEST EIGENFREQUENCIES [Hz]							BENCH TEST EIGENFREQUENCIES [Hz]								
		33.4	33.4	39.2	44.7	46.3	49.3	53.7	33.4	33.4	39.2	44.7	46.3	49.3	53.7		
FE MODEL V2 EIGENFREQUENCIES [Hz]	31.3	0.97	0.00	0.00	0.00	0.00	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	31.3	FE MODEL V3 EIGENFREQUENCIES [Hz]
	36.2	0.02	0.98	0.00	0.00	0.14	0.00	0.00	0.02	0.98	0.00	0.00	0.13	0.00	0.00	36.2	
	37.5	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	0.98	0.00	0.00	0.00	0.00	37.5	
	43.2	0.00	0.00	0.02	0.77	0.00	0.18	0.00	0.00	0.00	0.02	0.79	0.00	0.17	0.00	43.2	
	47.3	0.01	0.00	0.00	0.22	0.01	0.77	0.00	0.01	0.00	0.00	0.20	0.01	0.79	0.00	47.3	
	50.2	0.00	0.02	0.00	0.00	0.94	0.02	0.00	0.00	0.02	0.00	0.00	0.94	0.02	0.00	50.2	
	53.0	0.00	0.00	0.00	0.00	0.01	0.00	0.62	0.00	0.00	0.00	0.00	0.01	0.00	0.61	53.0	

Table 16. Third, fourth, sixth and seventh modes highlighted at MAC tables from V2 (left) and V3 (right) models.

This small modal correlation improvement shown at the MAC analysis is confirmed by the improvements noticed at the FRFs analysis between the models V2 and V3, at the third, fourth, and fifth modes (between 39Hz and 50Hz).

The plots regarding all the experimental test excitations and responses of V2 model are shown in Appendix IV.

So, as the main diagonal terms of the MAC matrix (Table 16) are still lower than 0.9, the FRAC calculation should not be processed, once the modal convergence is still not good enough to consider the linear model consistent.

However, considering that improvements in the third, fourth, and fifth modes are noticed comparing the V2 x V3 MAC tables, the V3 Linear FE model is taken as the base from which the modeling refinement should be continued.

3.3.3.3. V4 Improvement Model

As a next step of refinement, the accelerometers and impact masses representation, as well as their masses, are add to the model, creating the V4 Linear FE model (Figure 49).

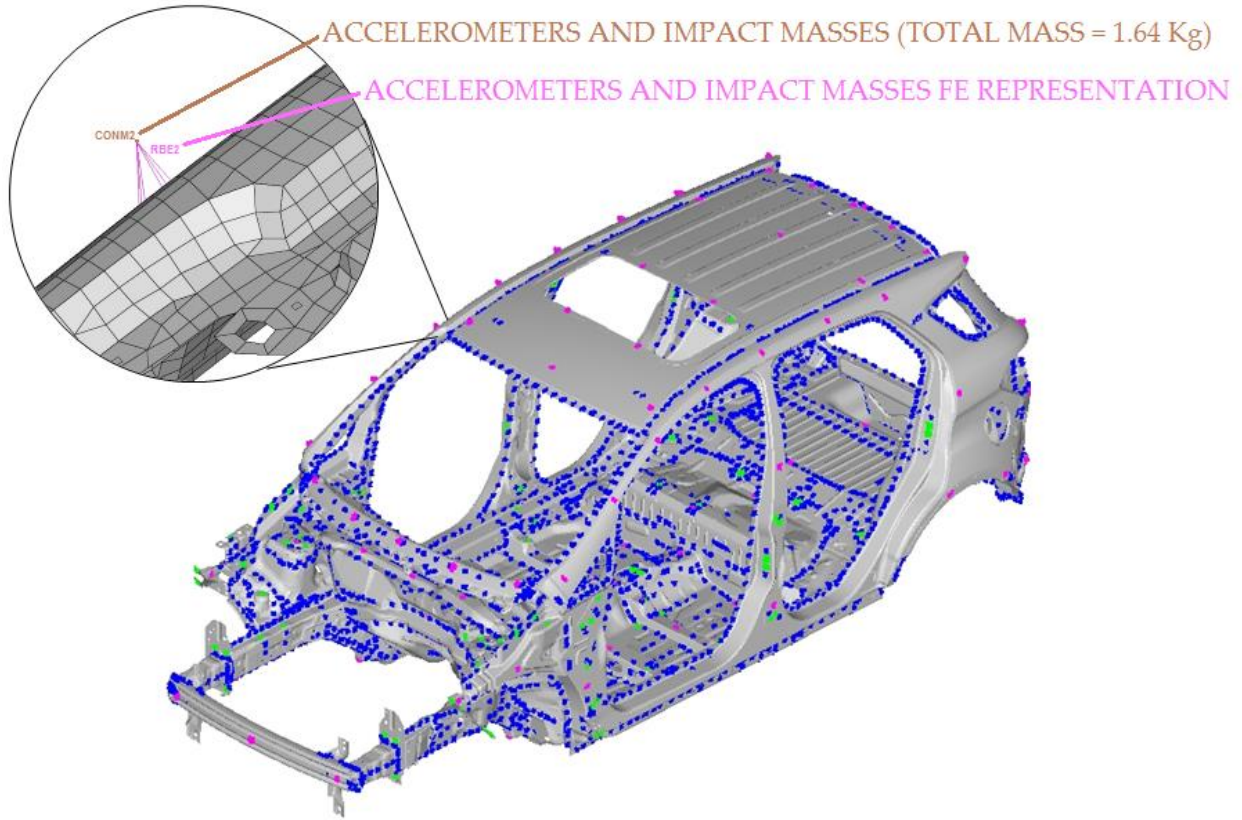


Figure 49. Linear FE model V4 of the vehicle body structure with accelerometers and impact masses.

So, a new set of analysis considering the Point Mobility and Seat track FRFs are performed with the updated Linear FE model V4.

The FRF analysis of this model shows that, in this case, the most significant changes take place between 30 Hz and 32 Hz. It can be noticed at both left and right front shock towers, mainly when exciting the body structure in x and y directions (Figures 50 and 51) but considering the responses in all directions.

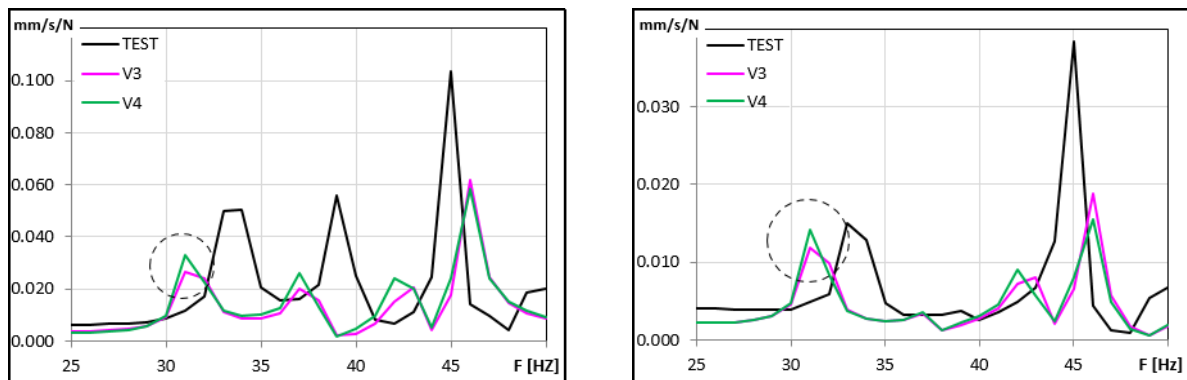


Figure 50. Experimental test *vs* FE linear model V3 *vs* V4 FRFs (y-dir response to unitary x-dir excitation at front left shock tower (left) and x-dir response to unitary y-dir excitation at front left shock tower (right)).

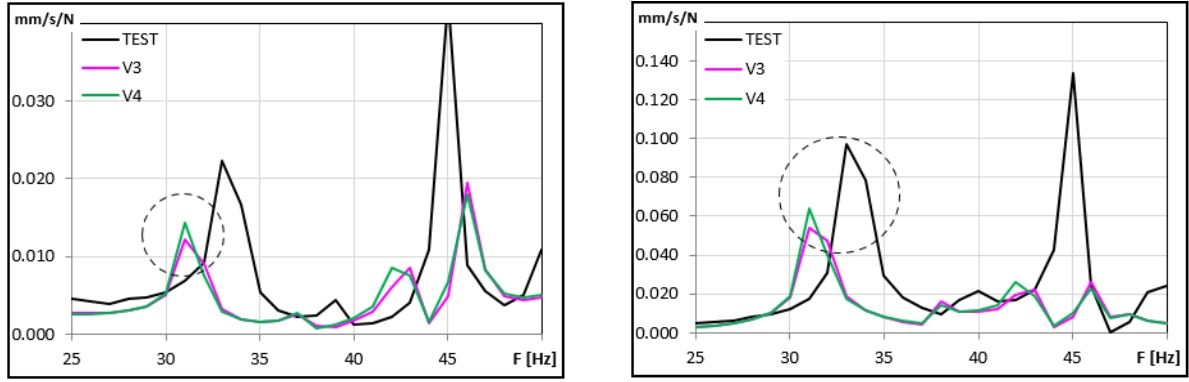


Figure 51. Experimental test *vs* FE linear model V3 *vs* V4 FRFs (x-dir response to unitary y-dir excitation at front right shock tower (left) and y-dir response to unitary z-dir excitation at front right shock tower (right)).

As well as already find at the front part of the body structure, a similar behavior can be also noticed at the rear part of the body structure. Also, the FRFs shows a consistent improvement between 30 Hz and 32 Hz (Figures 52 and 53) at both left and right shock towers.

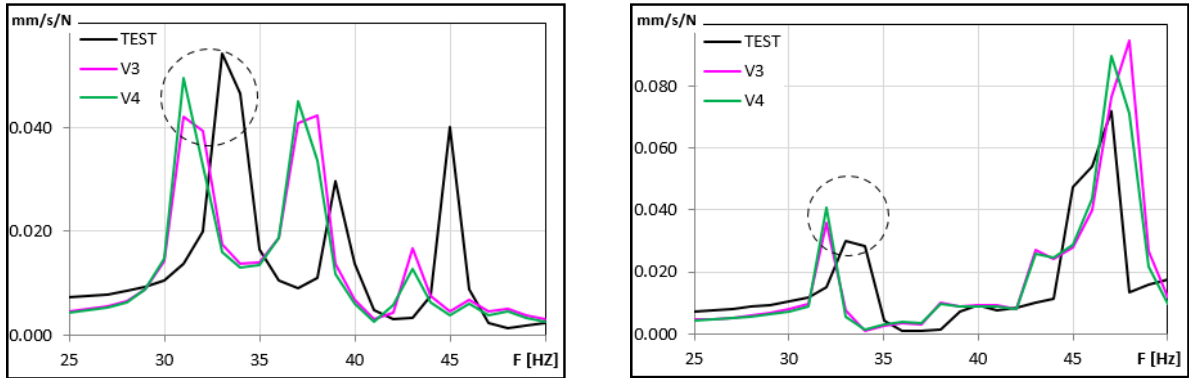


Figure 52. Experimental test *vs* FE linear model V3 *vs* V4 FRFs (y-dir response to unitary x-dir excitation at rear left shock tower (left) and x-dir response to unitary y-dir excitation at rear left shock tower (right)).

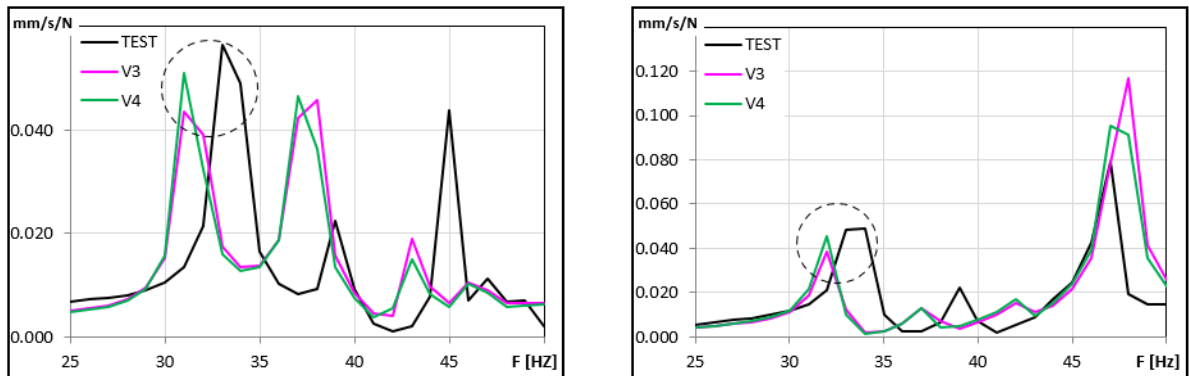


Figure 53. Experimental test *vs* FE linear model V3 *vs* V4 FRFs (x-dir response to unitary y-dir excitation at rear right shock tower (left) and y-dir response to unitary z-dir excitation at rear right shock tower (right)).

So, in order to verify whether this improvement between 30 Hz and 32 Hz consist in a global structure effect or in a local response, a comparison between both V3 and V4 MAC tables (Table 17) have to be performed.

		BENCH TEST EIGENFREQUENCIES [Hz]							BENCH TEST EIGENFREQUENCIES [Hz]							
		33.4	33.4	39.2	44.7	46.3	49.3	53.7	33.4	33.4	39.2	44.7	46.3	49.3	53.7	
FE MODEL V3 EIGENFREQUENCIES [Hz]	31.3	0.97	0.00	0.00	0.00	0.00	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	FE MODEL V4 EIGENFREQUENCIES [Hz]
	36.2	0.02	0.98	0.00	0.00	0.13	0.00	0.00	0.02	0.98	0.00	0.00	0.14	0.00	0.00	
	37.5	0.00	0.00	0.98	0.00	0.00	0.00	0.00	0.00	0.00	0.98	0.00	0.00	0.00	0.00	
	43.2	0.00	0.00	0.02	0.79	0.00	0.17	0.00	0.00	0.00	0.01	0.79	0.00	0.17	0.00	
	47.3	0.01	0.00	0.00	0.20	0.01	0.79	0.00	0.01	0.00	0.00	0.19	0.01	0.80	0.00	
	50.2	0.00	0.02	0.00	0.00	0.94	0.02	0.00	0.00	0.02	0.00	0.00	0.94	0.02	0.00	
	53.0	0.00	0.00	0.00	0.00	0.01	0.00	0.61	0.00	0.00	0.00	0.00	0.01	0.00	0.59	

Table 17. Fifth and seventh modes highlighted at MAC tables from V3 (left) and V4 (right) models

As a result, it can be noticed that the modal behavior of the body structure is only slightly affected. The comparison between V3 and V4 MAC tables (Table 17) shows that that eigenmodes small difference.

In addition to this, a comparison between the results of the experimental test animation (Figure 54 and 55, left) against the V3 (Figures 54 and 55, center) and V4 models (Figures 54 and 55, right) is performed. The results shows that the moonroof torsion mode and the global vertical bending mode are not significantly affected. Despite the front bumper structure of both V3 and V4 models do not follow the experimental test behavior, no differences can be noticed between them considering this animation. So, a global structure improvement cannot be considered the root cause of the FRF improvement in this case.

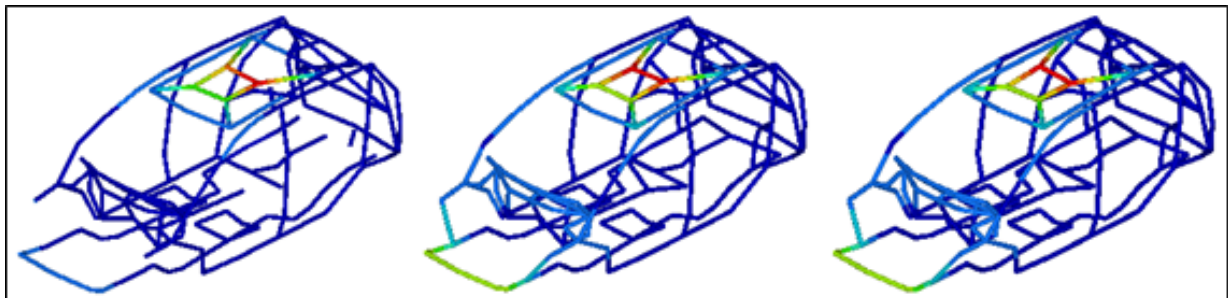


Figure 54. Modal shape of moonroof torsion natural mode at the experimental test (left), FE linear model V3 (left) and V4 (right).

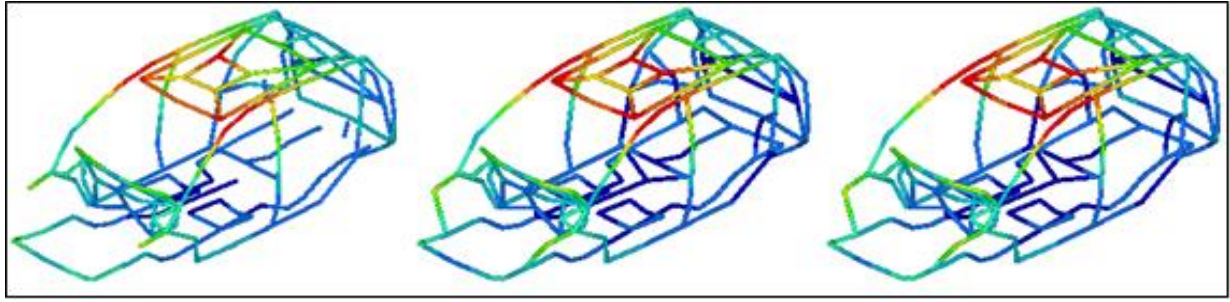


Figure 55. Modal shape of the global vertical bending mode at the experimental test (left), FE linear model V3 (left) and V4 (right).

On the other hand, it can be noticed at the point mobility curves the similar improvements between 30Hz and 32 Hz, already noticed at the FRF curves in the front shock towers (Figure 56) and rear shock towers (Figure 57) in all directions.

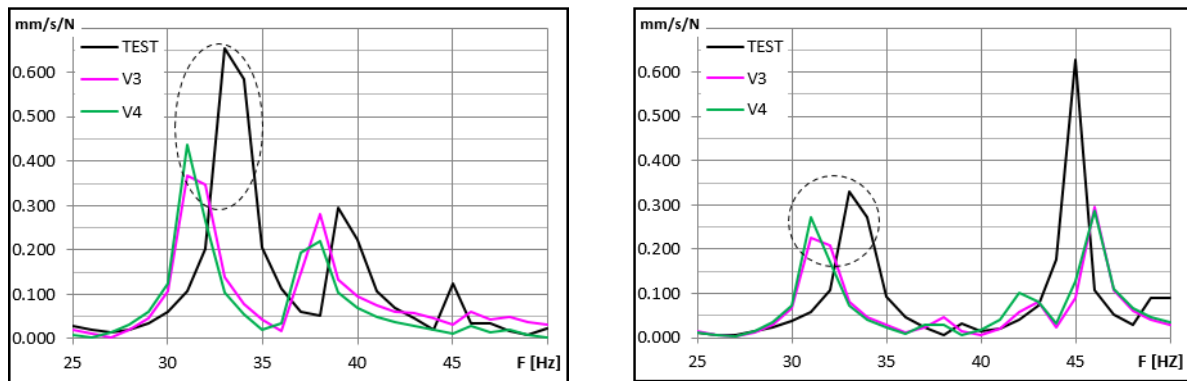


Figure 56. Experimental test *vs* FE linear model V3 *vs* V4 point mobility z-dir at front left shock tower (left) and y-dir at front right shock tower (right).

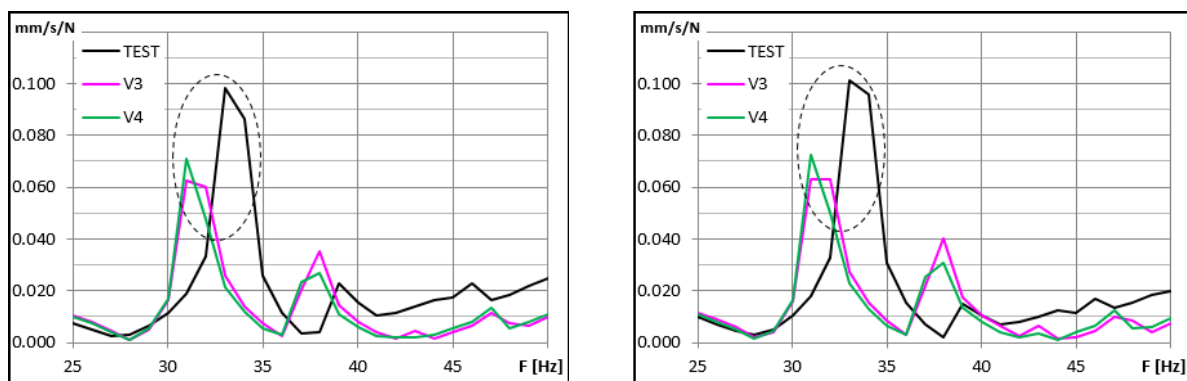


Figure 57. Experimental test *vs* FE linear model V3 *vs* V4 point mobility x-dir at rear left shock tower (left) and x-dir at rear right shock tower (right).

This comparison shows that the simulation results are consistent, considering that the model update is given by the addition of the accelerometers and impact masses at the hard point locations, which is

mainly affected the results close to the hard point locations, instead of affecting the global modes of the structure.

The plots regarding all the experimental test excitations and responses of V3 model are shown in Appendix V.

So, as the main diagonal terms of the MAC matrix (Table 17) are still lower than 0.9, the FRAC calculation should not be processed, once the modal convergence is still not good enough to consider the linear model consistent.

On the other hand, considering that an improvement in the 6th mode is noticed comparing the V3 x V4 MAC tables, the V4 Linear FE model is taken as the base from which the modeling refinement should be continued.

3.3.3.4. V5 Improvement Model

As a next step of refinement, the sealers and adhesive material properties representation are updated in the model based in the product supplier specification, as well as the addition of the cowl top sealer, creating the V5 Linear FE model (Figure 58).

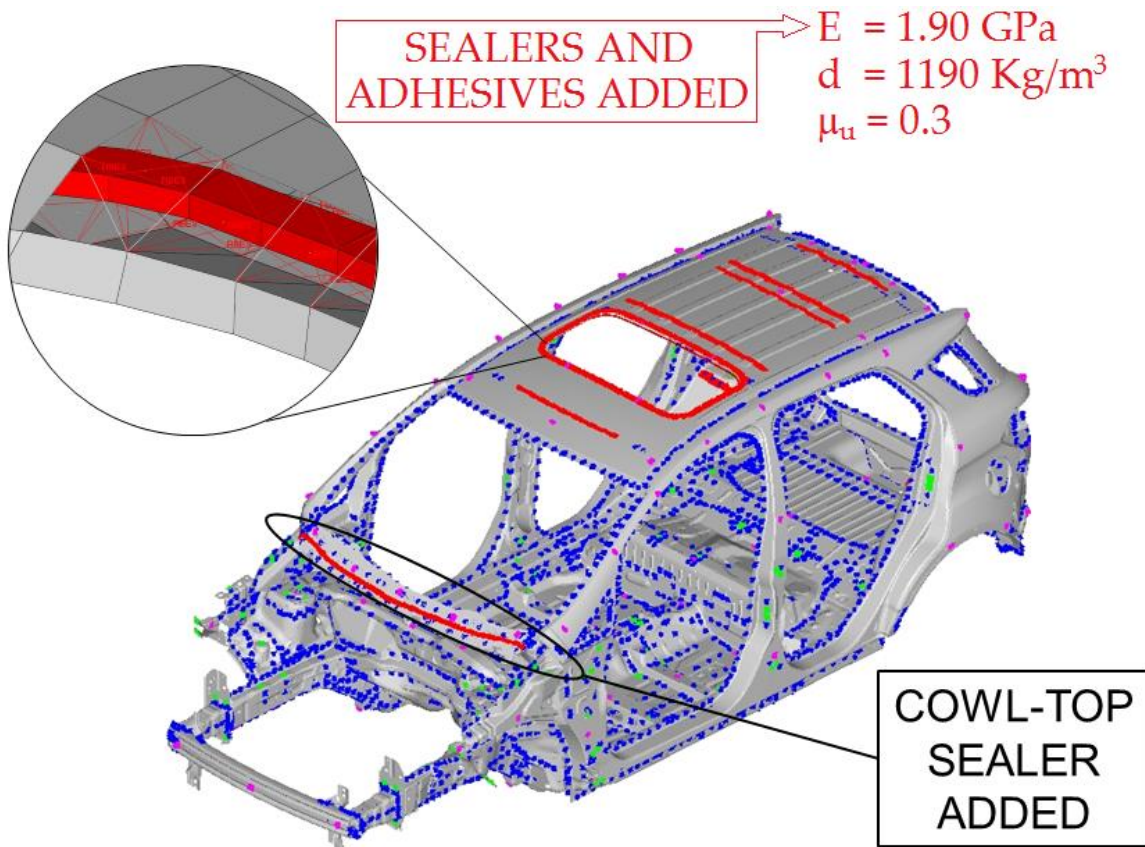


Figure 58. Linear FE model V5 of the vehicle body structure with sealers and adhesive properties.

So, a new set of analysis considering the Point Mobility and Seat track FRFs are performed with the updated Linear FE model V5.

The analysis shows that the most significant FRF improvements can be found between 44 Hz and 47 Hz. As a result, analyzing the front left shock tower FRFs, it can be noticed that the modal correspondent peak found in V4 curve at 46 Hz have moved to 44 Hz in V5 curve.

This FRF improvement can be seen exciting the front left shock tower in X-dir and getting the response in Y-dir as well as exciting the front shock tower in Y-dir and getting the response in X-dir (Figure 59).

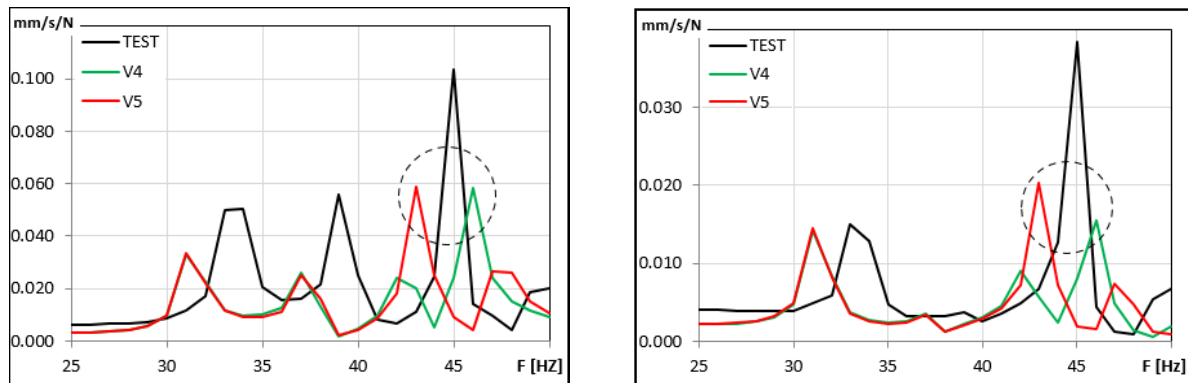


Figure 59. Experimental test *vs* FE linear model V4 *vs* V5 FRFs (y-dir response to unitary x-dir excitation at front left shock tower (left) and x-dir response to unitary y-dir excitation at front left shock tower (right)).

The same improvements are found when exciting the front right shock tower (Figure 60) considering both X-dir and Y-dir.

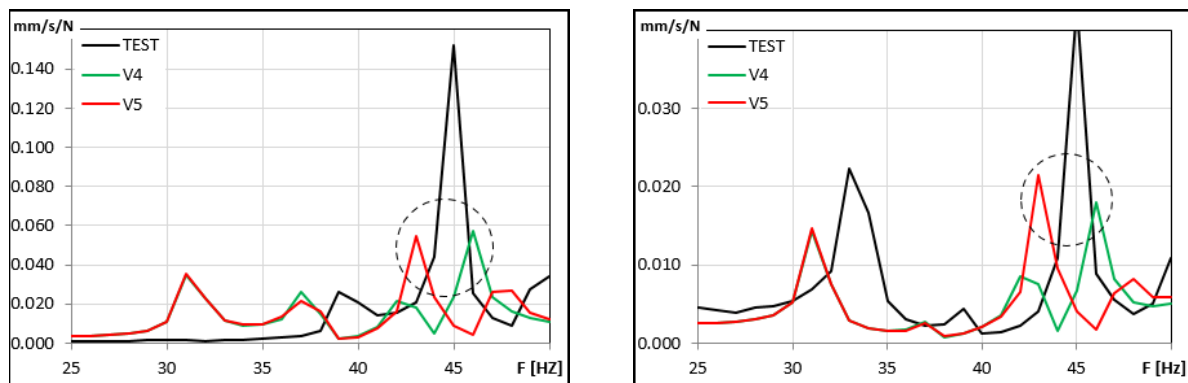


Figure 60. Experimental test *vs* FE linear model V4 *vs* V5 FRFs (y-dir response to unitary x-dir excitation at front right shock tower (left) and x-dir response to unitary y-dir excitation at front right shock tower (right)).

The FRF curves from the rear part of the structure (Figures 61 and 62) also shows similar behavior when comparing with the front structure FRFs, but this time affecting not only Y-dir FRFs, but Z-dir FRFs too.

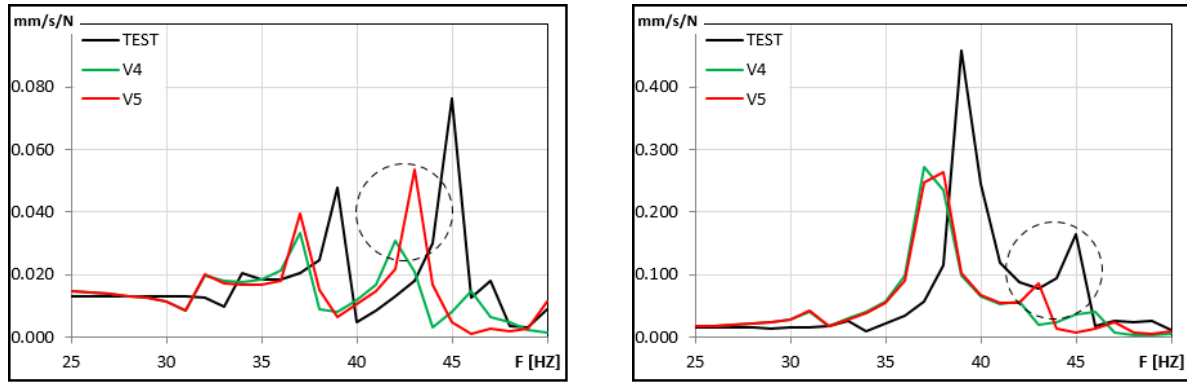


Figure 61. Experimental test *vs* FE linear model V4 *vs* V5 FRFs (y-dir response to unitary y-dir excitation at rear left shock tower (left) and y-dir response to unitary z-dir excitation at rear left shock tower (right)).

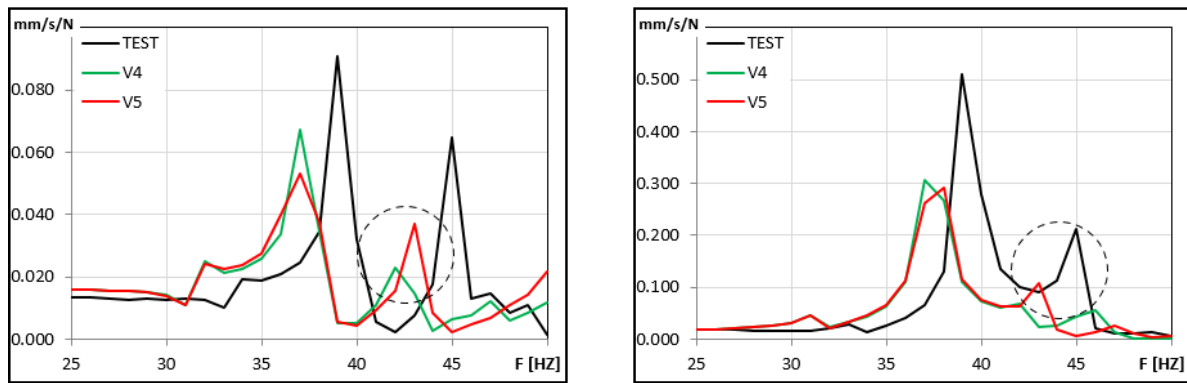


Figure 62. Experimental test *vs* FE linear model V4 *vs* V5 FRFs (y-dir response to unitary y-dir excitation at rear right shock tower (left) and y-dir response to unitary z-dir excitation at rear right shock tower (right)).

So, in order to verify the effect of the model updates over the global structure modes, a comparison between both V4 and V5 MAC tables (Table 18) have to be performed.

		BENCH TEST EIGENFREQUENCIES [Hz]							BENCH TEST EIGENFREQUENCIES [Hz]								
		33.4	33.4	39.2	44.7	46.3	49.3	53.7	33.4	33.4	39.2	44.7	46.3	49.3	53.7		
FE MODEL V4 EIGENFREQUENCIES [Hz]	31.3	0.97	0.00	0.00	0.00	0.00	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	31.3	FE MODEL V5 EIGENFREQUENCIES [Hz]
	36.2	0.02	0.98	0.00	0.00	0.14	0.00	0.00	0.02	0.98	0.00	0.00	0.07	0.00	0.00	36.2	
	37.5	0.00	0.00	0.98	0.00	0.00	0.00	0.00	0.00	0.01	0.98	0.00	0.00	0.00	0.00	37.5	
	43.2	0.00	0.00	0.01	0.79	0.00	0.17	0.00	0.00	0.00	0.01	0.96	0.00	0.01	0.00	43.2	
	47.3	0.01	0.00	0.00	0.19	0.01	0.80	0.00	0.01	0.00	0.00	0.04	0.01	0.91	0.00	47.3	
	50.2	0.00	0.02	0.00	0.00	0.94	0.02	0.00	0.00	0.09	0.00	0.00	0.95	0.02	0.00	50.2	
	53.0	0.00	0.00	0.00	0.00	0.01	0.00	0.59	0.00	0.00	0.00	0.00	0.06	0.00	0.93	53.0	

Table 18. Sixth and seventh modes highlighted at MAC tables from V4 (left) and V5 (right) models

As a result, it can be noticed at the MAC table that the lateral bending global mode (44.7 Hz) and the vertical bending global mode (53.7 Hz) of the structure are significantly improved by the model updates performed at the V5 model. Moonroof modes (46.3 Hz and 49.3 Hz) are also affected, despite having local effect only.

These improvements can also be noticed comparing the animation of the experimental test lateral and vertical global modes (Figures 63 and 64, left) with the V4 (Figures 63 and 64, center) and the V5 (Figures 63 and 64, right) models.

Considering the global lateral bending mode, the front bumper and the moonroof areas of V5 model follows the experimental test better than V4. This same improvement can be seen for the global vertical bending mode at the front bumper, moon roof, and at the cowl top area.

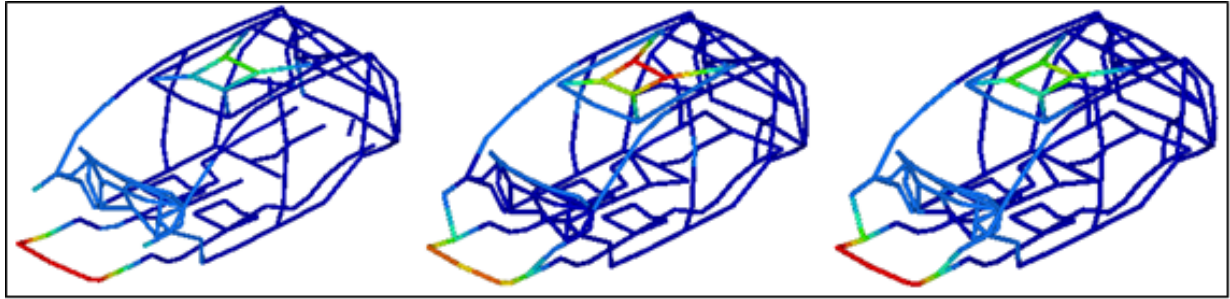


Figure 63. Modal shape of global lateral bending natural mode at the experimental test (left), FE linear model V4 (left) and V5 (right).

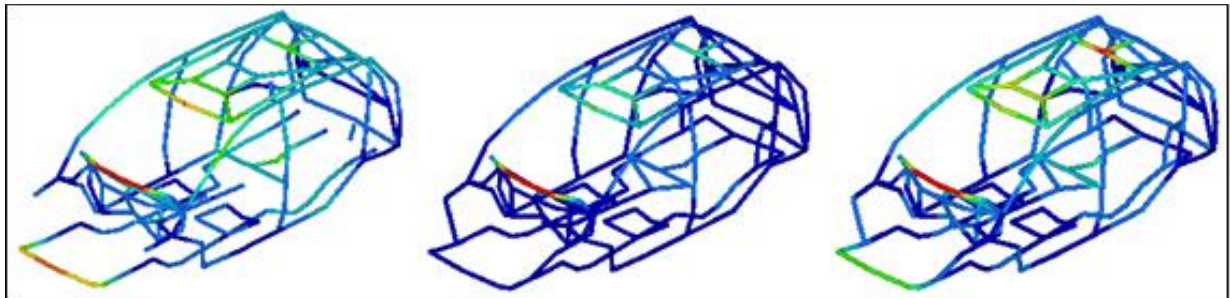


Figure 64. Modal shape of the global vertical bending mode at the experimental test (left), FE linear model V4 (left) and V5 (right).

In addition to this, the point mobility curves at the left and right front shock towers in Y-dir (Figure 65) have moved the peak in V4 curve at 46 Hz to 44 Hz in V5 curve, indicating that part of this MAC improvement is related to a local adjustment of the front shock towers stiffness in the Y-dir too.

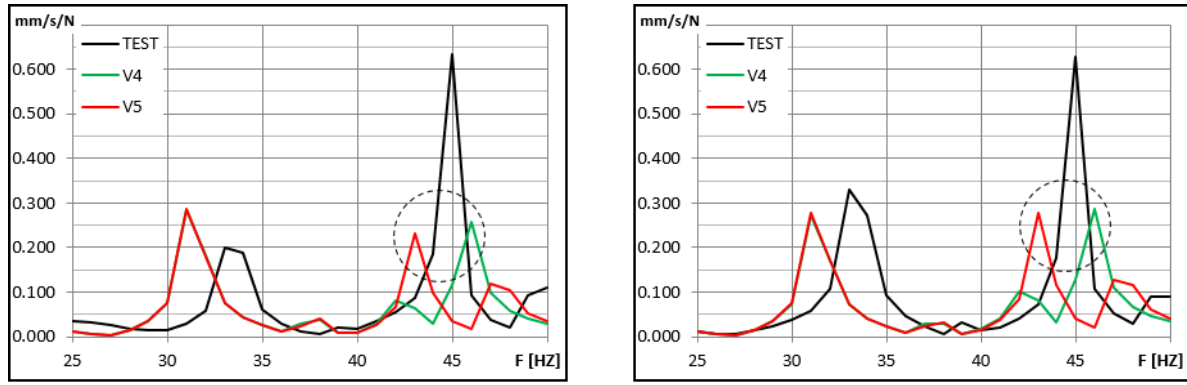


Figure 65. Experimental test *vs* FE linear model V4 *vs* V5 point mobility y-dir at front left shock tower (left) and y-dir at front right shock tower (right).

3.3.3.5. Reference FE Model of the Vehicle Body Structure

This comparison shows that the simulation results are consistent, considering that the model update is given by the revision of the sealers properties at the cowl top area, close to front shock towers joints, and at the moonroof connections. As a result, the bending global modes and the moonroof local modes of the structure are the main affected.

The plots regarding all the experimental test excitations and responses of V4 and V5 models are shown in Appendix VI and VII.

Considering that improvements in the FRF curves and point mobility curves are consistent when comparing the V4 x V5 graphics and MAC tables, the V5 Linear FE model is taken as the base from which the analysis should be continued.

It is also noticed that the fifth and sixth modes are switched. This switch is considered to be an undesirable effect of the small number of accelerometers used in the bench test to acquire the local modes in the moonroof opening region.

Even so, the main diagonal terms of the MAC matrix (Table 18) have higher coefficients than 0.9 for all global modes. In any case, the effect of these switched modes affects only the moonroof area. Thus, it is concluded that, despite this permutation of modes, the results are still consistent.

So, in this case, as the results shows that all the main diagonal terms of the MAC matrix (Table 18) are higher than 0.9, the FRAC calculation can be processed, once the modal convergence is good enough to consider the linear model consistent.

From this point on the FE linear model V5 is assumed as the reference FE model of the vehicle body structure model for the further analysis calculation.

The table 19 shows the results of the FRAC analysis between the reference FE model and the experimental test of the vehicle body structure:

Impact Point/Dir		FRAC	Impact Point/Dir		FRAC	Impact Point/Dir		FRAC
REFERENCE FE MODEL RESPONSE POINT: X-DIR	front left/x	0.9560	REFERENCE FE MODEL RESPONSE POINT: Y-DIR	front left/x	0.2315	REFERENCE FE MODEL RESPONSE POINT: Z-DIR	front left/x	0.8497
	front left/y	0.2270		front left/y	0.1654		front left/y	0.1428
	front left/z	0.4164		front left/z	0.2343		front left/z	0.6700
	front right/x	0.9320		front right/x	0.1547		front right/x	0.5928
	front right/y	0.1732		front right/y	0.1720		front right/y	0.1246
	front right/z	0.3994		front right/z	0.2403		front right/z	0.4808
	rear left/x	0.9647		rear left/x	0.2690		rear left/x	0.2206
	rear left/y	0.8365		rear left/y	0.2645		rear left/y	0.4020
	rear left/z	0.7581		rear left/z	0.1026		rear left/z	0.4220
	rear right/x	0.9682		rear right/x	0.2488		rear right/x	0.3059
	rear right/y	0.8887		rear right/y	0.2379		rear right/y	0.3238
	rear right/z	0.6374		rear right/z	0.1110		rear right/z	0.5542
Average x-dir		0.6798	Average y-dir		0.2027	Average z-dir		0.4241
-	-	-	-	-	-	Overall average		0.4355

Table 19. FRAC results for the reference FE model of the vehicle body structure.

3.4. FE Test Models

3.4.1. Model Definition

From the MAC and FRAC results for the reference FE model is noticed a gap between the results with the bench experimental test and the reference FE model of the vehicle body structure. Since the reference FE model already includes all possible linear structural components and connections, it is not possible to increase even more its accuracy just by considering the linear boundary conditions.

Therefore, the FE model should also include other nonlinear elements. Since the contacts between the welded flanges of the vehicle body parts are present in the whole structure, it can be also included in the FE model of the body structure.

However, the change from linear to nonlinear boundary conditions, given by the addition of nonlinear contact elements on the whole structure, exponentially increases the required calculation efforts. Therefore, it is unfeasible to analyze this nonlinear FE model using the current computational resources available.

Taking into account this limitation, a linear FE test model (Figure 66, left) is build, replicating at a small scale the design of the flanges of the vehicle body structure, according to the following parameters:

- Plate size [mm]: $250 \times 50 \times 1.0$;
- Gap between upper and lower plates: 1.0 mm;
- Spot welds: 5 spots located every 50.0 mm; and
- System support condition: simply supported.

A nonlinear FE test model (Figure 66, center) is built by adding nonlinear contact elements to the linear FE test model in the surroundings of each spot weld. A contact static friction coefficient $\{\phi_s\} = 0.25$ [31–33] is applied to this.

So, a third FE test model with linear contact elements (Figure 66, right) is also build, adjusting their stiffness and damping coefficients by comparing the results of the other two FE test models.

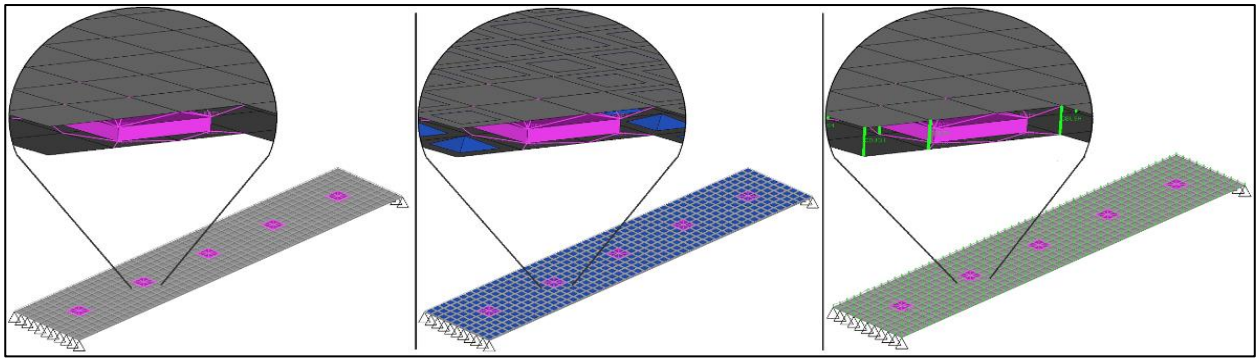


Figure 66. FE test models: without contact elements (left), with nonlinear contact elements (center) and with linear adjusted contact elements (right).

3.4.2. Comparison of the First Eigenfrequencies

Initially, a first set of analyses is performed to assess the differences in the first eigenfrequency between the linear and nonlinear FE test models. To do this, a load of 0.01 N is applied to all nodes of the upper plate, exciting the system with a time-dependent sinusoidal enforced motion during 1 s, with a frequency ranging from 10 to 60 Hz, with a 1 Hz step.

Since the plate deflection shows a bending shape, the point mobility FFT of the deflection of the central node of the upper plate is plotted. From the FFT curves, the first eigenfrequency of each model is identified.

The following calculation deck (Table 20) is used to extract the point mobility curve under the time-dependent sinusoidal enforced motion from the linear FE test model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
CONTF (H3D, ALL) = ALL	Request contact results at all contact elements in H3D format
DISPLACEMENT (H3D) = ALL	Request displacement results at all nodes in H3D format
SPCFORCE (H3D) = ALL	Request force results at all nodes in H3D format
SUBCASE 1	Delimits and identifies the first subcase
LABEL transient_linear	Defines the analysis label
ANALYSIS DTRAN	Identify the solution used to perform the analysis, in this case, a Direct Transient Response Analysis
SPC = 2	Defines a single-point constraints set for the subcase analysis
DLOAD = 7	Defines the subcase DLOAD parameter
TSTEP (TIME) = 9	Defines the transient response analysis integration, in this case, a time step integration in the time domain
BEGIN BULK	Designates the start of the Bulk Data Section
LOADCOL 10"TABLE_10Hz"	Defines a data file where the frequency points of the enforced motion excitation to be applied are available, in this case, at 10Hz
TLOAD1 7 3 10	Defines the type of load to be applied for the enforced motion, in this case, force
TSTEP 91000 0.001	Defines the total time and the time step to be used at the analysis
FORCE 3 1	Request structural element force output
SET1 1	Defines the element set for the structural element force output
ENDDATA	Designates the end of the Bulk Data Section

Table 20. Optistruct deck used to extract the point mobility curve under the time-dependent sinusoidal enforced motion from the linear FE test model

So, the following calculation deck (Table 21) is used to extract the point mobility curve under the time-dependent sinusoidal enforced motion from the non-linear FE test model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
CONTF (H3D, ALL) = ALL	Request contact results at all contact elements in H3D format
DISPLACEMENT (H3D) = ALL	Request displacement results at all nodes in H3D format
SPCFORCE (H3D) = ALL	Request force results at all nodes in H3D format
SUBCASE 1	Delimits and identifies the first subcase
LABEL transient_linear	Defines the analysis label
ANALYSIS DTRAN	Identify the solution used to perform the analysis, in this case, a Direct Transient Response Analysis
SPC = 2	Defines a single-point constraints set for the subcase analysis
DLOAD = 7	Defines the subcase DLOAD parameter
NLPARM (LGDISP) = 4	Activate nonlinear solution method for the subcase and select the parameters used for Nonlinear Direct Transient Analysis (when placed at Subcase section)
TSTEP (TIME) = 9	Defines the transient response analysis integration, in this case, a time step integration in the time domain
NLADAPT 101	Select the parameters used for Nonlinear Transient Analysis
NLOUT = 5	Select the incremental result output parameters used for Nonlinear Transient Analysis
BEGIN BULK	Designates the start of the Bulk Data Section
LOADCOL 10 "TABLE_10Hz"	Defines a data file where the frequency points of the enforced motion excitation to be applied are available, in this case, at 10Hz
TLOAD1 7 3 10	Defines the type of load to be applied for the enforced motion, in this case, force
NLPARM 4 20 0.1	Defines parameters for Nonlinear Direct Transient Analysis (when placed at Bulk Data section). In this case, the total number of increments and the initial load
TSTEP 91000 0.001	Defines the total time and the time step to be used at the analysis

NLADAPT 101DTMAX 5.0-5	Defines parameters for time-stepping and convergence criteria in Nonlinear Analysis
NLOUT 5NINT 1000SVNONCNV YES	Select the incremental result output parameters used for Nonlinear Transient Analysis
FORCE 3 1	Request structural element force output
SET1 1	Defines the element set for the structural element force output
ENDDATA	Designates the end of the Bulk Data Section

Table 21. Optistruct deck used to extract the point mobility curve under the time-dependent sinusoidal enforced motion from the non-linear FE test model

For the FE test model with only linear boundary conditions (Transient Direct solver solution), without contact representation, the first eigenfrequency is found at 66.4 Hz, with no variation of the results (< 0.001 Hz) over the excitation frequency range. An example at 10 Hz is shown in Figure 67 left.

From the FE test model with nonlinear boundary conditions (Transient Non-Linear solver solution) and including nonlinear contacts, the first eigenfrequency is found at 70.3 Hz, with no other relevant results variation (< 0.001 Hz) over the excitation frequency range. An example at 10 Hz is shown in Figure 67 right.

The difference between the first results of the two first FE test models (3.9 Hz) is used to adjust the stiffness of the spring-damper contact elements included in the third linear FE test model.

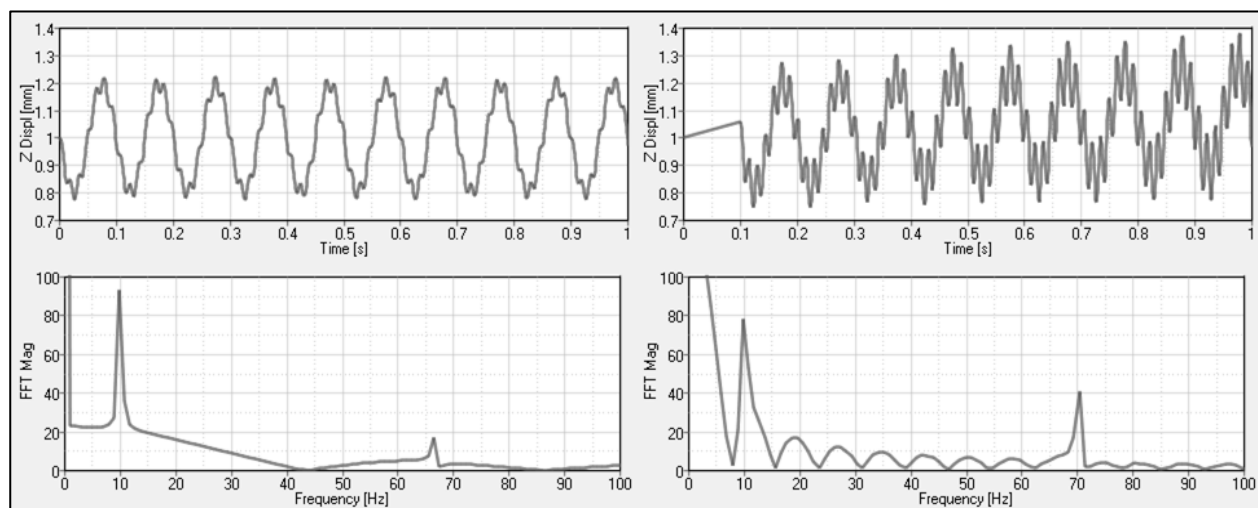


Figure 67. linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 10Hz.

Considering the results for the first eigenfrequency remain constant over the frequency range of interest (from 10 to 60 Hz), a linear interpolation approach is used to estimate an adjusted stiffness value. All other frequency graphics are shown in Appendix VIII.

The first eigenfrequency of the linear FE reduced model, $F_a = 66.4 \text{ Hz}$, is taken as the starting point. Since there are no included spring-damper elements in this model, it is assumed $K_a = 0.0 \text{ N/mm}$ as its correspondent stiffness.

Next, a set of spring-damper elements is added to the linear FE test model using an arbitrary stiffness value of $K_b = 1000 \text{ N/mm}$. After this, the same transient direct analyses already performed are run as before and the first eigenfrequency, F_b , is find at 71.3 Hz , with no results variation over the same excitation frequency range.

In the sequence, the first eigenfrequency of the nonlinear FE test model (70.3 Hz) is interpolated between the points a and b (Figure 68) and the corresponding adjusted stiffness value $K_{adj} = 801.0 \text{ N/mm}$ is find.

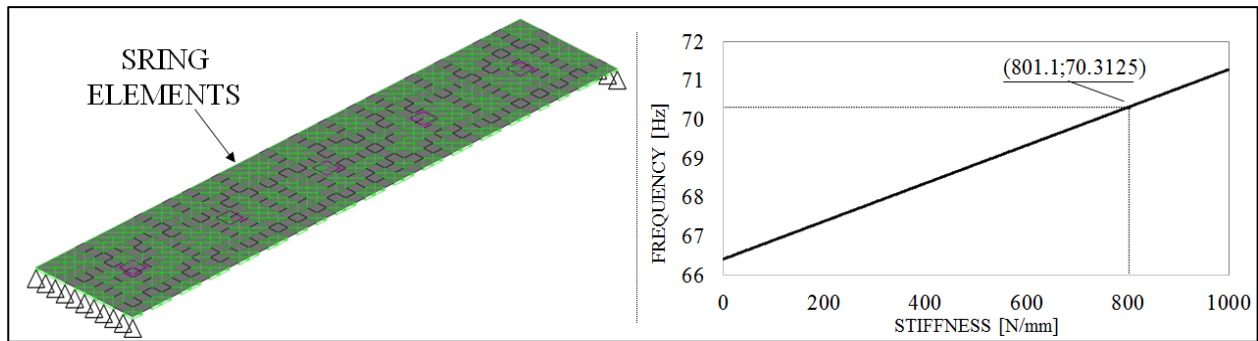


Figure 68. FE test models: without contact elements (left), with nonlinear contact elements (center) and with linear adjusted contact elements (right).

The adjusted linear spring-damper elements ($K_{adj} = 801.0 \text{ N/mm}$) are also validated by the analysis of a model build from the linear FE test model with the addition of such elements. In this way, the first eigenfrequency is find at 71.3 Hz , which is the same value found with the nonlinear FE test model.

3.4.3. Comparison of the Damping Decay Curves

A second set of analyses is performed to assess the differences in the damping decay curves between the linear and nonlinear FE test models. To do this, a pulse of 1 N is applied on the central node of the upper plate for 0.01 s (Transient Modal solver solution).

The following calculation deck (Table 22) is used to extract the point mobility curve under the 1 N pulse excitation from the linear FE test model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
OUTPUT, HGTRANS, ALL	Request results formatted for advanced graphic analysis
DISPLACEMENT (BOTH) = 1	Request displacement results at all nodes of SET 1 in H3D and in advanced graphics format
SUBCASE 1000	Delimits and identifies the first subcase
LABEL PULSE_LINEAR	Defines the analysis label
ANALYSIS MTRAN	Identify the solution used to perform the analysis, in this case, a Modal Transient Response Analysis
SPC = 2	Defines a single-point constraints set for the subcase analysis
METHOD (STRUCTURE) = 1004	Selects the real eigenvalue extraction parameters set for the structural analysis
SDAMP (STRUCTURE) = 1003	Selects the damping parameters set for the structural analysis
DLOAD = 1005	Defines the subcase DLOAD parameter
TSTEP (TIME) = 1001	Defines the transient response analysis integration, in this case, a time step integration in the time domain
SET1 = 1	Defines SET1 nodes
BEGIN BULK	Designates the start of the Bulk Data Section
EIGRL 1004 0.0 100.0 MASS	Defines data needed to perform real eigenvalue (vibration or buckling) analysis with the Lanczos method, in this case, from 0Hz to 100Hz
DAREA 1002 4714 3 -1.0	Defines scale (area) factors for dynamic load excitations, in this case an 1N negative pulse, applied to the hard point #4714 in the Z direction
TABLED1 1000 LINEAR LINEAR + 0.0 1.0 0.001 1.0 + 0.002 0.0 1.0 0.0	Defines a tabular function for use in generating the frequency-dependent dynamic loads and its sequential continuation line
TABDMP1 100 3 G + 0.0 0.016 0 0.0 0.016 ENDT	Defines modal damping as a tabular function of natural frequency and its sequential continuation line
TLOAD1 1005 1002 LOAD 1000	Defines the type of load to be applied for the enforced motion, in this case, the excitation force

TSTEP 1001 1000 0.001	Defines the total time and the time step to be used at the analysis
ENDDATA	Designates the end of the Bulk Data Section

Table 22. Optistruct deck used to extract the point mobility curve under the 1 N pulse excitation from the linear FE test model

The following calculation deck (Table 23) is used to extract the point mobility curve under the 1 N pulse excitation from the non-linear FE test model. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
OUTPUT, H3D, ALL	Request results at nodes and elements in H3D format
OUTPUT, HGTRANS, ALL	Request results formatted for graphic analysis
DISPLACEMENT (BOTH) = 1	Request displacement results at all nodes of SET 1 considering both H3D and graphics format
SUBCASE 1000	Delimits and identifies the first subcase
LABEL PULSE_LINEAR	Defines the analysis label
ANALYSIS MTRAN	Identify the solution used to perform the analysis, in this case, a Modal Transient Response Analysis
SPC = 2	Defines a single-point constraints set for the subcase analysis
METHOD (STRUCTURE) = 1004	Selects the real eigenvalue extraction parameters set for the structural analysis
SDAMP (STRUCTURE) = 1003	Selects the damping parameters set for the structural analysis
NLPARM (LGDISP) = 4	Activate nonlinear solution method for the subcase. This command also select the parameters used for Nonlinear Direct Transient Analysis when placed at Subcase section
NLOUT = 5	Select the incremental result output parameters used for Nonlinear Transient Analysis
DLOAD = 1005	Defines the subcase DLOAD parameter

TSTEP (TIME) = 1001	Defines the transient response analysis integration, in this case, a time step integration in the time domain
SET1 = 1	Defines SET1 nodes
BEGIN BULK	Designates the start of the Bulk Data Section
EIGRL 1004 0.0 100.0 MASS	Defines data needed to perform real eigenvalue (vibration or buckling) analysis with the Lanczos method, in this case, from 0Hz to 100Hz
DAREA 1002 4714 3 -1.0	Defines scale (area) factors for dynamic loads, in this case an 1N negative pulse, applied to the hard point #4714 in the Z direction
TABLED1 1000 LINEAR LINEAR + 0.0 1.0 0.001 1.0 + 0.002 0.0 1.0 0.0	Defines a tabular function for use in generating the frequency-dependent dynamic loads and its sequential continuation line
TABDMP1 100 3 G + 0.0 0.016 0 0.0 0.016 ENDT	Defines modal damping as a tabular function of natural frequency and its sequential continuation line
TLOAD1 1005 1002 LOAD 1000	Defines the type of load to be applied for the enforced motion, in this case, the excitation force
TSTEP 10011000 0.001	Defines the total time and the time step to be used at the analysis
NLADAPT 101DTMAX 5.0 -5	Defines parameters for time-stepping and convergence criteria for the Nonlinear Analysis
ENDDATA	Designates the end of the Bulk Data Section

Table 23. Optistruct deck used to extract the point mobility curve under the 1 N pulse excitation from the non-linear FE test model

The time response of the node displacement over a 1 s span (Figure 69, left) is taken and its envelope (Figure 69, right) represented. In this way, the exponential damping decay curves are plotted for both the linear and nonlinear FE test models. Also, both linear and nonlinear FE test model curve equation are calculated and shown.

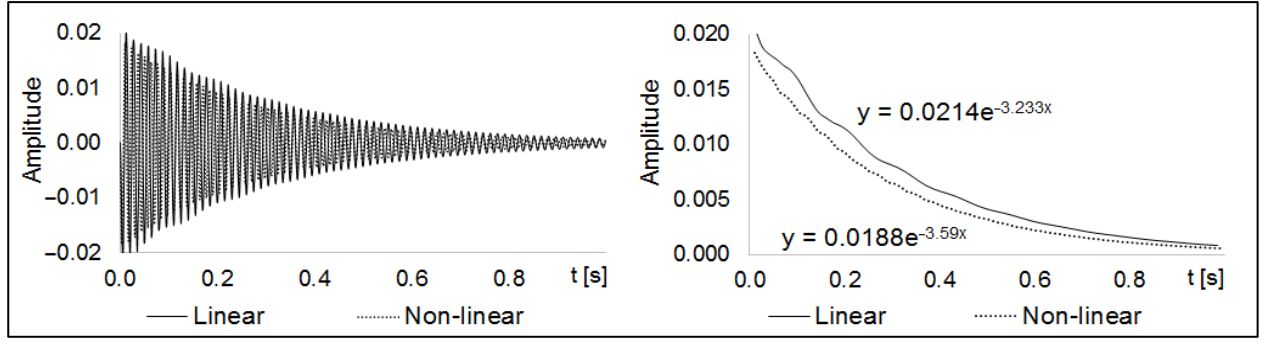


Figure 69. Linear and nonlinear z-dir displacement (left) and decaying rate (right) curves.

The exponents of the equations of these curves are their decaying rates, σ . This variable is also used to calculate the damping ratio ζ for an undamped system [34] on Equation (3), given by:

$$\zeta = \frac{1}{\sqrt{1 + \left(\frac{2\pi}{\sigma}\right)^2}} \quad (3)$$

being $\sigma \triangleq \ln \frac{y_1}{y_2}$ where y_1 and y_2 are the respective vibration amplitudes at two successive peaks of a decaying vibration, from which the decay rates $\sigma_{lin} = -3.233$ and $\sigma_{non-lin} = -3.590$ are calculated. It is also known that:

$$\sigma = \frac{B}{2m} \quad (4)$$

The relation $B_{non-linear} = 1.1104 \cdot B_{linear}$ is found from Equation (4), assuming that both linear and nonlinear systems have the same mass.

Then, this relationship is used to adjust the damping of the linear FE spring-damper elements, to simulate the effect of the nonlinear contacts.

The adjusted linear spring-damper elements are also validated by adding them to the linear FE test model. Figure 70 shows the decay curves and equations for the adjusted linear FE test model and the nonlinear FE test model.

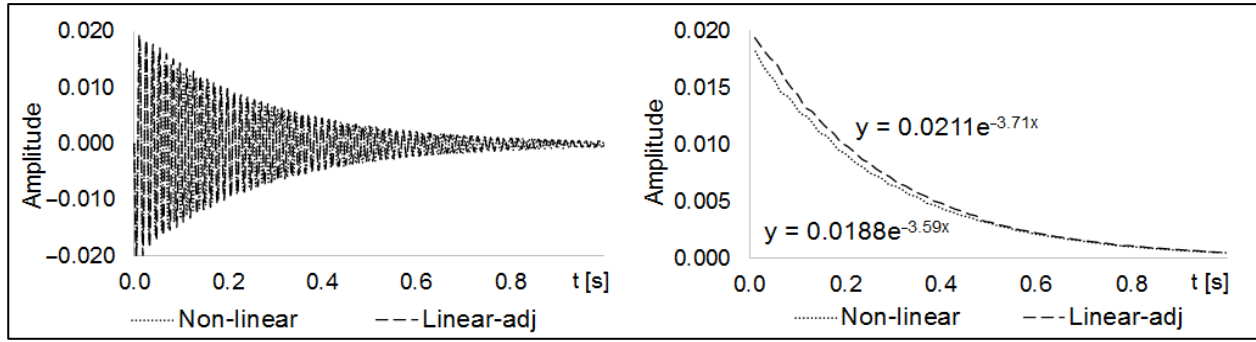


Figure 70. Nonlinear and linear adjusted z-dir displacement (left) and decaying rate (right) curves.

The comparison between the decay rate of the non-adjusted ($\sigma_{lin} = -3.233$) and adjusted ($\sigma_{adj} = -3.715$) linear FE test models with the decay rate of the nonlinear FE test model ($\sigma_{non-lin} = -3.590$), shows that the difference between the decay rates of the linear and nonlinear models decrease from 11.04% to 3.34% when using the adjustment. It can also be visually confirmed by comparing the curves and decay rates shown in Figure 70 details.

4. RESULTS

4.1. Adjusted FE Model of the Vehicle Body Structure

4.1.1. Model Definition

So, an adjusted linear FE model of the vehicle body structure (Figure 71) is build. To this end, having the to the reference FE model of the vehicle body structure as the starting point, a set of adjusted spring-damper elements is add along all the welded flanges of the vehicle.

This set is adjusted by using the stiffness and damping values, $K_{adj} = 801.0 \text{ N/mm}$ and $B_{adj} = 1.1104 \text{ N.s/mm}$, previously calculated from the FE test models.

From this adjusted FE model of the vehicle body structure, the same mode shapes and FRFs [22] as for the bench test and the reference FE model of the vehicle body structure are obtained.

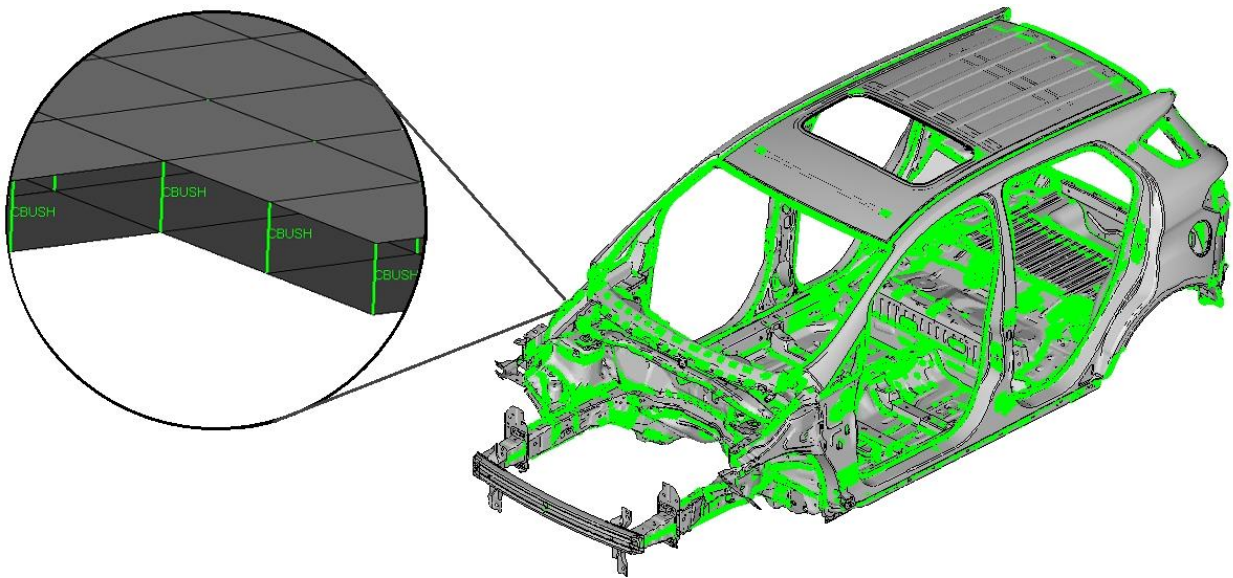


Figure 71. Adjusted linear FE model showing the added spring-damper elements.

Like for the bench test and the reference FE model of the vehicle body structure, the impact hammer test from 0 to 100 Hz is also simulated. To do so, a frequency response analysis to capture the FRFs is performed.

The real part of the eigenvalues is extracted using the Lanczos' method, considering the average structural damping ratio of 0.0044815, the same structural damping value of the bench test performed on the vehicle body structure.

So, in order to verify the effect of the model updates over the global structure modes, a comparison between both V5 and Vadj eigenmodes (Table 24) and MAC tables (Table 25) have to be performed.

Table 24 gathers the eigenfrequencies of the main eigenmodes of the body structure found in the bench test, as well as on the reference and adjusted FE models. The variation between the eigenfrequencies of the V5 and Vadj FE models with reference to the bench test results remains at similar levels, although showing some differences between the higher and lower frequency ranges.

Mode	Modal Shape	Bench Test [Hz]	Reference FE Model [Hz]	Adjusted FE Model [Hz]
1	Windshield Torsion	33.38	31.33	33.40
2	1st Moonroof Bending	33.38	36.22	37.10
3	Global Torsion	39.23	37.52	40.01
4	Global Lateral Bending	44.71	43.19	44.87
5	2nd Moonroof Bending	46.34	47.32	49.65
6	Moonroof Torsion	49.34	50.17	52.43
7	Global Vertical Bending	53.73	53.02	55.89

Table 24. Eigenfrequencies of the vehicle body structure from the reference and adjusted FE models and bench test.

Table 25 gathers the MAC of the adjusted FE model of the vehicle body structure contrasted with the bench test. Despite the switch mentioned above between the fifth and sixth eigenmodes, the MAC coefficients are still higher than 0.9.

Thus, the modal results also have consistent correspondence with the bench test data. The main diagonal terms on the MAC matrix of the adjusted FE model of the vehicle body structure are generally better (higher) than for the reference FE model, thus indicating an improved accuracy of the structure modal behavior for the adjusted model.

So, the FRAC calculation can be processed, once the modal convergence is still good enough to consider the linear model consistent.

4.2. Reference vs Adjusted FE Model Results Comparison

		BENCH TEST EIGENFREQUENCIES [Hz]							BENCH TEST EIGENFREQUENCIES [Hz]							
		33.4	33.4	39.2	44.7	46.3	49.3	53.7	33.4	33.4	39.2	44.7	46.3	49.3	53.7	
FE MODEL V3 EIGENFREQUENCIES [Hz]	31.3	0.97	0.00	0.00	0.00	0.00	0.00	0.00	0.97	0.00	0.00	0.00	0.00	0.00	0.00	33.4
	36.2	0.02	0.98	0.00	0.00	0.07	0.00	0.00	0.02	0.99	0.00	0.00	0.10	0.00	0.00	37.1
	37.5	0.00	0.01	0.98	0.00	0.00	0.00	0.00	0.00	0.00	0.99	0.00	0.00	0.00	0.00	40.0
	43.2	0.00	0.00	0.01	0.96	0.00	0.01	0.00	0.00	0.00	0.00	0.98	0.00	0.00	0.00	44.9
	47.3	0.01	0.00	0.00	0.04	0.01	0.91	0.00	0.01	0.00	0.00	0.03	0.01	0.93	0.00	49.7
	50.2	0.00	0.09	0.00	0.00	0.95	0.02	0.00	0.00	0.07	0.00	0.00	0.97	0.02	0.00	52.4
	53.0	0.00	0.00	0.00	0.00	0.06	0.00	0.93	0.00	0.00	0.00	0.00	0.02	0.00	0.96	55.9
																FE MODEL V _{adj} EIGENFREQUENCIES [Hz]

Table 25. MAC results of the adjusted FE model of the vehicle body structure.

4.2. Reference vs Adjusted FE Model results comparison

Having the reference and adjusted FE model results available, the modal correlation studies are performed with the aim to compare them against the bench test measurement. Three complementary methodologies are used for this.

First, the FRAC coefficients of the reference and adjusted FE models are calculated and compared, taking into account the whole frequency range defined for the transfer functions correlation analysis.

Second, the peaks of both reference and adjusted FE models transfer functions are compared with the peaks extracted from the bench test measurements, focusing the most critical ranges of the transfer functions.

And finally, a statistical analysis is performed to compare the set of results from both reference and adjusted FE models against the bench test measurements, highlighting the points of maximum divergence and convergence in the correlation analysis.

4.2.1. FRAC Coefficients Comparison Analysis

The FRAC coefficients of the FE models are contrasted against the bench test results. These coefficients can be here understood as an analytical method to evaluate the correlation based on a general behavior of the models, by comparing their vibrational transfer functions at all points of the curve.

So, a higher FRAC coefficient indicates more accurate simulation transfer functions, resulting in higher correlation levels.

Impact Point/Dir		FRAC	Impact Point/Dir		FRAC	Impact Point/Dir		FRAC
REFERENCE FE MODEL RESPONSE POINT: X-DIR	front left/x	0.9666	REFERENCE FE MODEL RESPONSE POINT: Y-DIR	front left/x	0.1839	REFERENCE FE MODEL RESPONSE POINT: Z-DIR	front left/x	0.8323
	front left/y	0.1859		front left/y	0.1795		front left/y	0.1909
	front left/z	0.4848		front left/z	0.3990		front left/z	0.7250
	front right/x	0.9846		front right/x	0.1181		front right/x	0.5154
	front right/y	0.1774		front right/y	0.1946		front right/y	0.2316
	front right/z	0.4986		front right/z	0.3854		front right/z	0.4522
	rear left/x	0.9647		rear left/x	0.3333		rear left/x	0.2151
	rear left/y	0.8627		rear left/y	0.3187		rear left/y	0.3266
	rear left/z	0.8263		rear left/z	0.3838		rear left/z	0.3873
	rear right/x	0.9657		rear right/x	0.2916		rear right/x	0.3961
	rear right/y	0.9432		rear right/y	0.3388		rear right/y	0.2362
	rear right/z	0.6384		rear right/z	0.3813		rear right/z	0.5864
Average x-dir		0.7082	Average y-dir		0.2923	Average z-dir		0.4246
-	-	-	-	-	-	Overall average		0.4751

Table 26. FRAC results for the adjusted FE model of the vehicle body structure.

The overall average FRAC for the adjusted FE model of the vehicle body structure, $FRAC_{adj} = 0.4751$, is computed from the FRAC coefficients shown in Table 26. It can be noticed that it is 9.1% higher than the overall average FRAC for the reference FE model of the vehicle body structure, $FRAC_{ref} = 0.4355$ (Table 19).

This comparison confirms the correlation improvement given by the proposed adjustment, also noticed by the contrasting of the MAC.

4.2.2. Frequency Response Functions peaks Comparison Analysis

4.2.2.1. Modal Shapes Comparison

The peaks of both reference and adjusted FE models transfer functions are compared with the peaks extracted from the bench test measurements. In this case, using a different strategy than that used previously to calculate the FRAC coefficient, only these specific points of each transfer function (peaks) are considered. The high criticality of these points when compared to the rest of the curve justify the special attention devoted to their analysis.

For this, the reference and adjusted FE models are excited by a unitary load at the same hard point and direction where each peak is found, also considering the same frequency. The nodal displacement response

is acquired in the whole model, and the modal shape animation of both reference and adjusted FE models are plotted. Then, the bench test modal plots are analyzed and the modal shape which represent the higher contributor mode at each peak taken. In fact, this is the bench test modal shape that will be compared with the reference and adjusted FE models.

A similarity scale is used to classify each transfer function peak modal shape. A higher score on the similarity scale indicates a higher correlation level between a FE model transfer function response and its correspondent bench test result. Based on the delta of amplitude between the FE models and the bench test results, the 30 most relevant peaks are selected.

The modal correlation of the FE models is analyzed at each selected peak. They are shown from the right left front shock tower to the right rear shock tower, following the x-y-z direction order, and also from the lowest to the highest frequency. On the sequence, the related modal shape of the reference and adjusted FE models, as well as the correspondent bench test mode are shown, followed by a similarity score that assigns 0 points when no similarity is noticed until to 3 when full similarity exists.

In addition to this, the point mobility curves in each shock tower are compared for qualitative purposes only, without assigning a score to them, since in this case the isolated analysis of the peaks would not be indicated. So, considering the first excitation, at the body structure at the front left shock tower in the x-direction, three relevant response peaks are noticed at the seat attachment transfer function acquired from the bench test at 34Hz , 39Hz, and 45Hz (Figure 72).

In the first case (peak A, at 34Hz) the modal shapes comparison clearly shows how the windshield and the front roof areas of the adjusted FE model follows the bench test modal shape better than the reference FE model. This behavior is also noticed at the transfer function curves, where the adjusted FE model FRF copies the bench test curve closely than the reference FE model. The high similarity score assigned to the adjusted FE model (three points) reflects this consistent correlation level improvement over the reference FE model (one point).

The analysis of the peak B (39Hz) shows that the reference and adjusted FE modal shapes have some similarities. The 1st moonroof bending mode, isolated from the rest of the structure on the bench test, is affected by the global torsion mode at the FE models. However, a closer analysis of the modal shapes makes it clear that while this discrepancy affects the entire rear, front and roof structure of the reference FE model, in the adjusted FE model the influence on these same areas has a significantly smaller reach.

The transfer function comparison confirms the modal shape analysis as the adjusted FE model peak at 39Hz is closer to the bench test results considering the y-dir response. So, a similarity score of one point is assigned to the reference FE model, while two points are assigned to the adjusted FE model.

At the third peak analyzed (peak C, at 45Hz), a strong contrast can be noticed when comparing the modal shapes of the FE models. While the reference model shows no similarity with the test bench results, the adjusted model accurately follow its modal shape along the whole structure. A similar behavior can be seen considering the transfer functions. It can be noticed that all responses of the adjusted model have the same bench test results frequency. Based on these results the reference FE model received the minimum score while the adjusted FE model received the maximum.

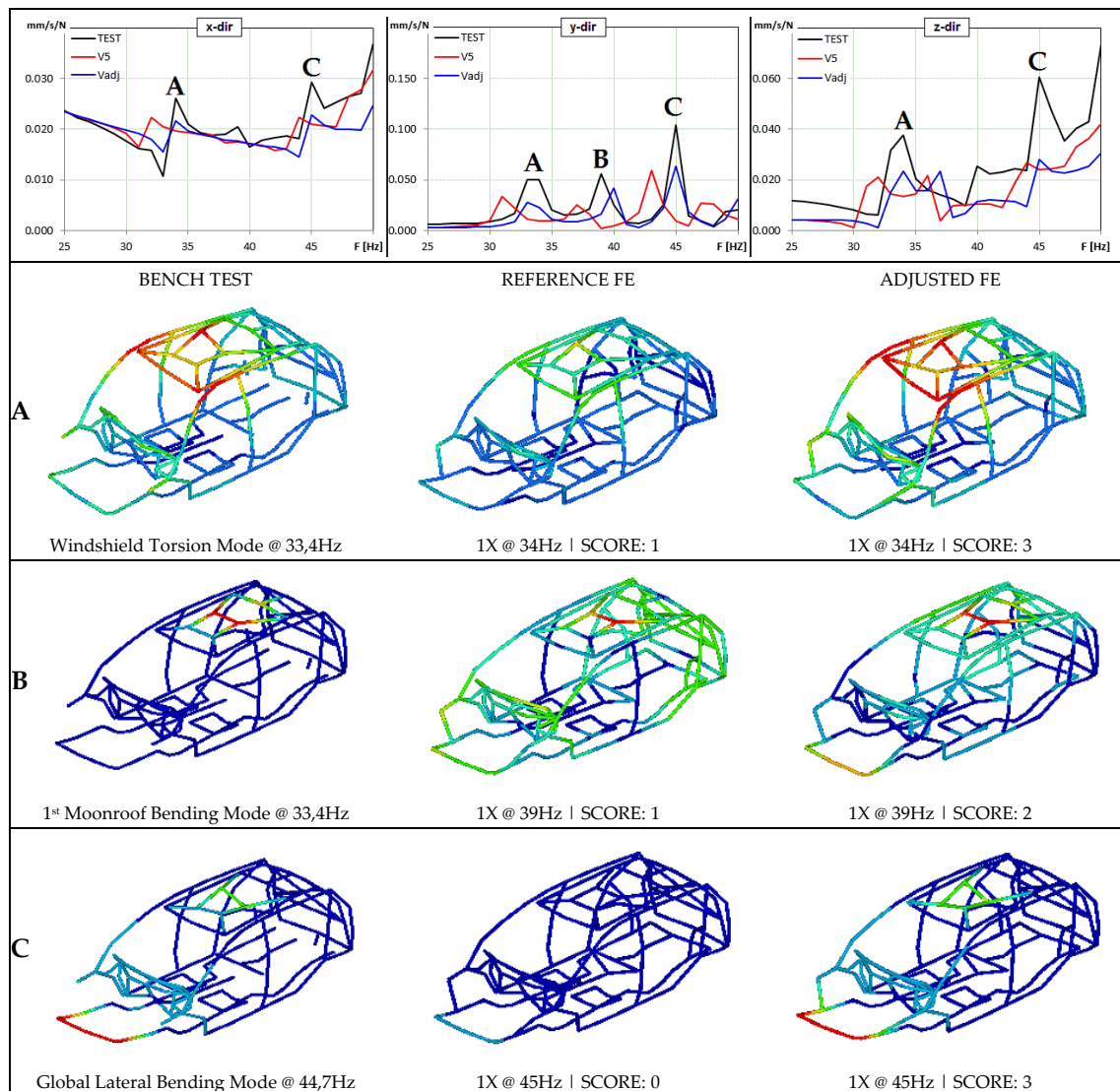


Figure 72. Experimental test *vs* reference *vs* adjusted FE model front left shock tower x-dir FRFs, modal shape responses and similarity scores.

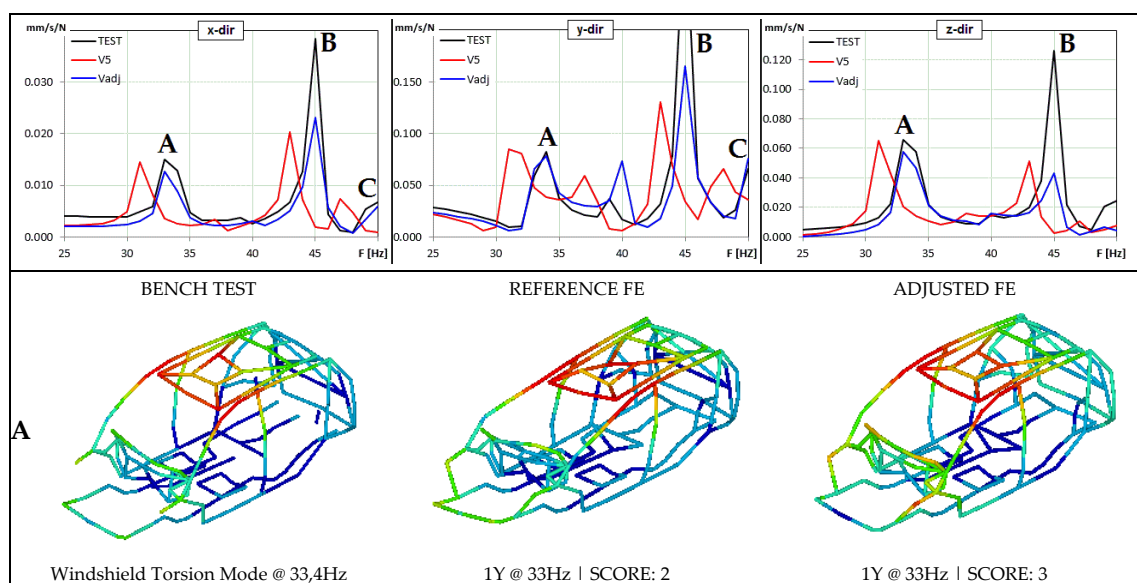
4.2. Reference vs Adjusted FE Model Results Comparison

Considering the case where the body structure is excited at the front left shock tower in the y-direction, three relevant response peaks are noticed at 33Hz , 39Hz, and 50Hz (Figure 73).

The modal shape analysis at the peak A (33 Hz) shows that the rear structure of the reference model have good correlation with the bench test results while the windshield and roof areas have lower correlation. In contrast, the adjusted model shows high correlation with the bench test modal shape along the whole body structure. It is also noticed at the transfer function comparison that the adjusted model follows much closely the bench test curve than the reference model. So a score 2 is given to the reference FE model and a score 3 to the adjusted.

At peak B (45 Hz), the adjusted FE model follows the bench test result, despite having lower amplitude, as noticed by the transfer function comparison. Also, the front bumper structure of the adjusted model, although moving in the opposite phase, follows the modal shape of the bench test showing clearly the lateral bending mode. This lateral bending mode, as well as the roof structure behavior, are not properly represented by the modal shape of the reference model. So, the reference model have a score 0 while a score 1 is assigned to the adjusted FE model.

The analysis of the peak C (50Hz) shows that the adjusted FE modal shape follows the moonroof torsion mode of the bench test result as noticed by the transfer function comparison. On the other hand, the reference model modal shape is affected by other modes (lateral bending), which also influence the transfer function accuracy. So, a score 1 is assigned to the reference FE model, while a score 2 is assigned to the adjusted.



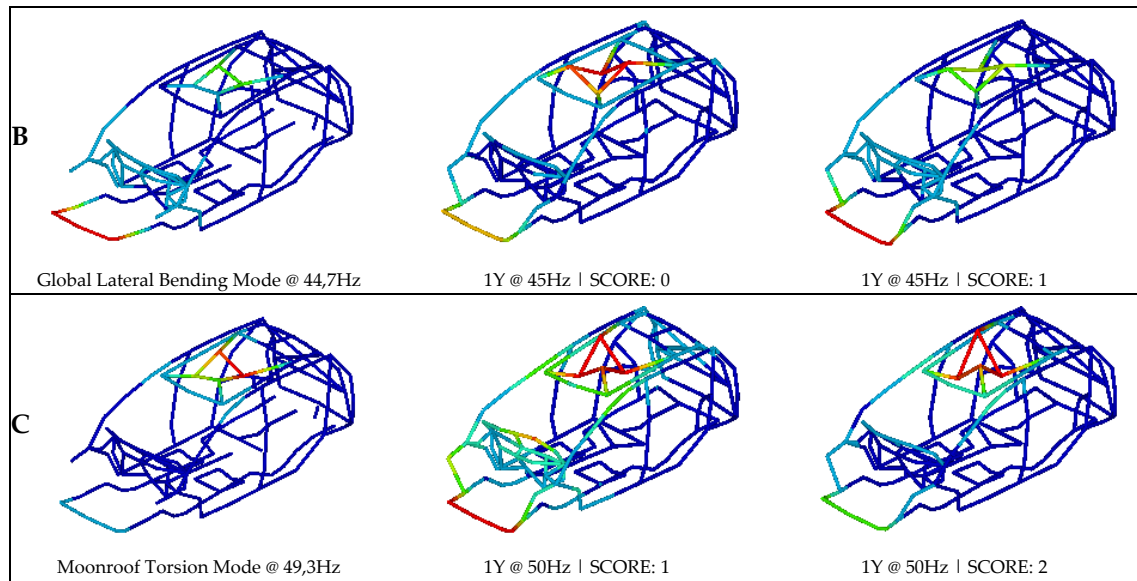


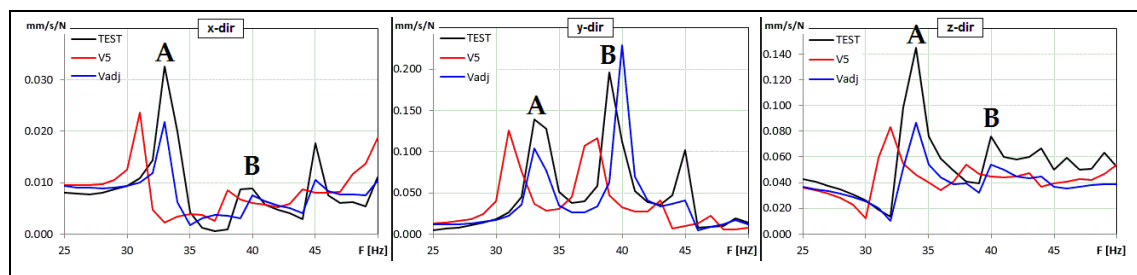
Figure 73. Experimental test *vs* reference *vs* adjusted FE model front left shock tower y-dir FRFs, modal shape responses and similarity scores.

Next, when the body structure is excited at the front left shock tower in the z-direction, two relevant response peaks are noticed at 34Hz and 40Hz (Figure 74).

The analysis of the mode shapes at peak A (34Hz) shows that the reference and the adjusted FE model plot correlates well with the bench test results, with exception of the front structure of the reference model, which do not follow the bench test dynamic behavior.

In addition to this, it can be noticed that the transfer function of the adjusted model have the same bench test results frequency at this peak, while the reference is 2Hz lower. Scores 1 to the reference and 2 to the adjusted FE model are assigned.

In the second case (peak B, 40Hz) the modal shapes of the reference and adjusted FE models are compared. A difference in the behavior of the upper joints of the rear frame can be seen. This part of the structure moves alternately in the bench test and in the adjusted FE model, characterizing the global torsion mode, while it remains stopped in the reference model. A score 2 is given to the reference FE model, while a score 3 is given to the adjusted.



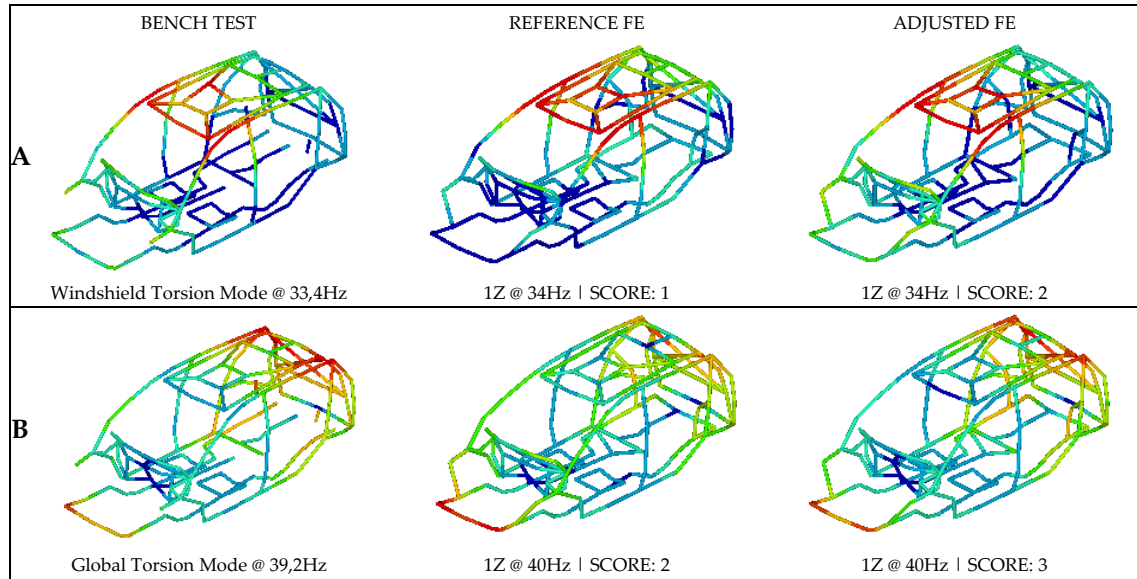


Figure 74. Experimental test *vs* reference *vs* adjusted FE model front left shock tower z-dir FRFs, modal shape responses and similarity scores.

In addition to this, the point mobility responses at the front left shock tower of the bench test results, the reference, and the adjusted FE models (Figure 75) are compared. In this case, as the point mobility represent the modal behavior at the loading points, not only the peaks are important, but all points along the whole frequency range too.

So, analyzing the full range of frequency it can be seen that the adjusted FE model curves follow the bench test results more accurate than the reference FE model, even considering the differences of amplitude shown by the adjusted curves at the peaks A, B and C. This behavior is also noticed at the frequency ranges D and E, where despite no peaks are seen, adjusted FE model point mobility curve shows better correlation than the reference model result.

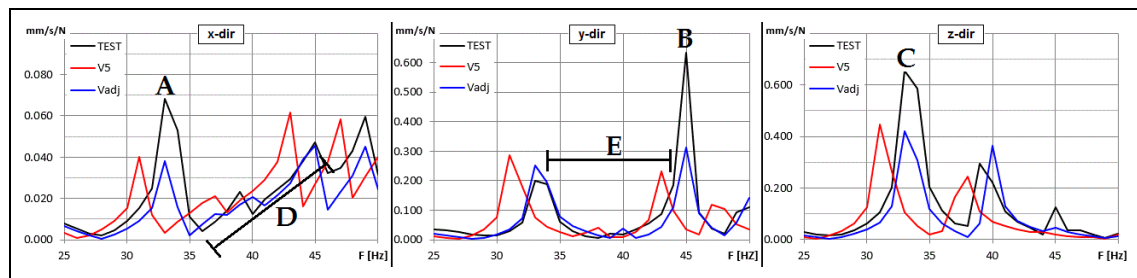


Figure 75. Experimental test *vs* reference *vs* adjusted FE model front left shock tower x, y and z-dir point mobility FRFs.

Next, three relevant response peaks are noticed at 34Hz , 39Hz, and 45Hz (Figure 76) in the case where the body structure is excited at the front right shock tower in the x-direction.

The analysis of peak A (34 Hz) shows that, although the modal shapes of the reference and adjusted FE model have similarities, the upper part of the reference model structure does not follow the bench test as

the adjusted model. The transfer function of the reference FE model is also close to the bench test result, but still not following as closely as the adjusted. A score 2 is given to the reference FE model and a 3 to the adjusted.

Analyzing the second peak (B at 39Hz), although the adjusted model does not perfectly correlate with the bench test results, its modal shape has more similarity than the reference model. This pattern is also seen in the transfer function analysis, where the adjusted FE model also follows the bench test more closely. So, scores 1 and 2 are assigned to the reference and adjusted FE models respectively.

At peak C (45Hz) there is a clear difference between the behavior of the reference FE model and the adjusted. While the reference model does not correlate well with the lateral bending mode of the bench test, the adjusted model follows it accurately. This same behavior can be seen at the transfer functions despite the amplitude delta. Scores 1 and 3 are given to the reference and adjusted models.

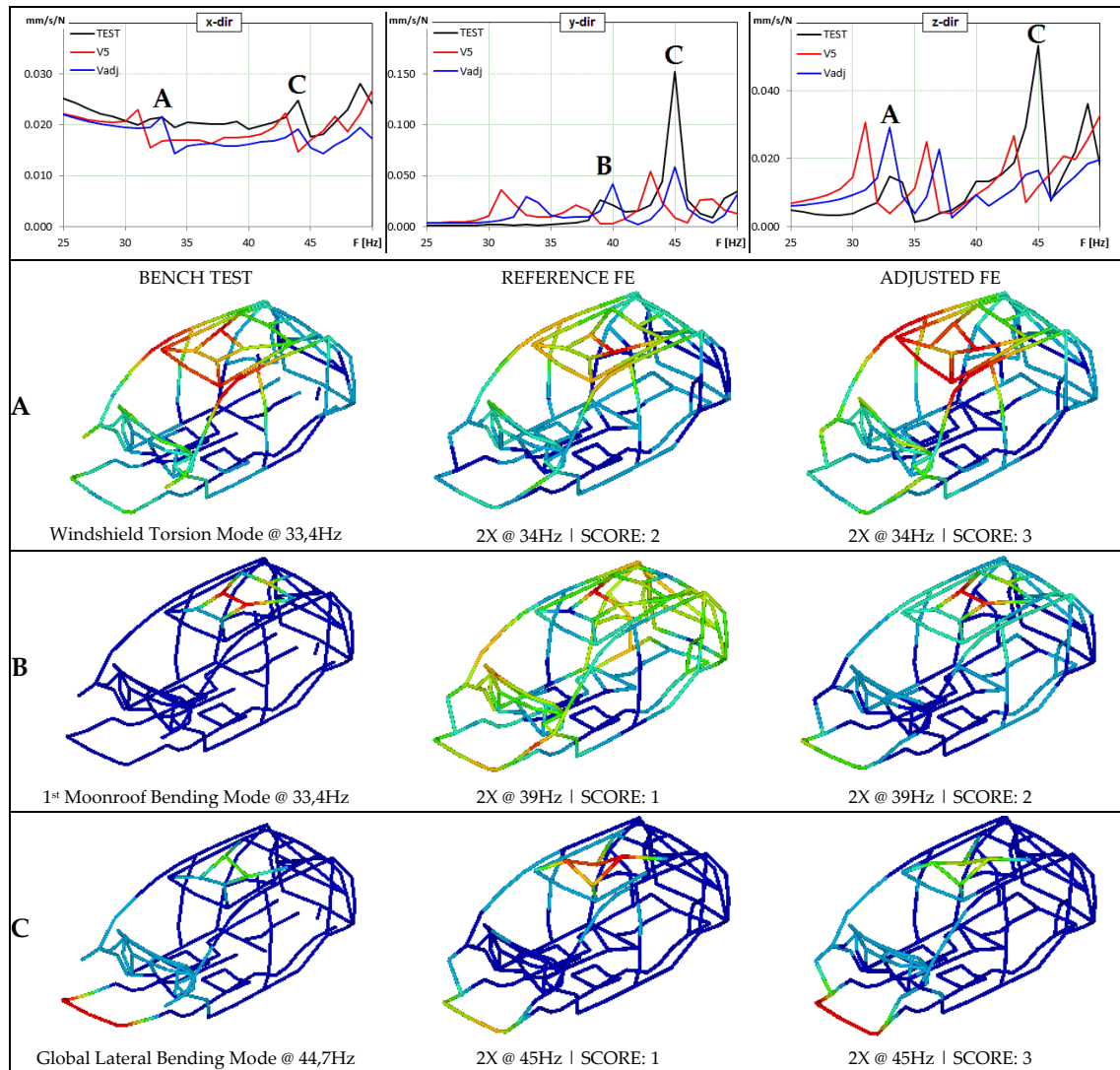
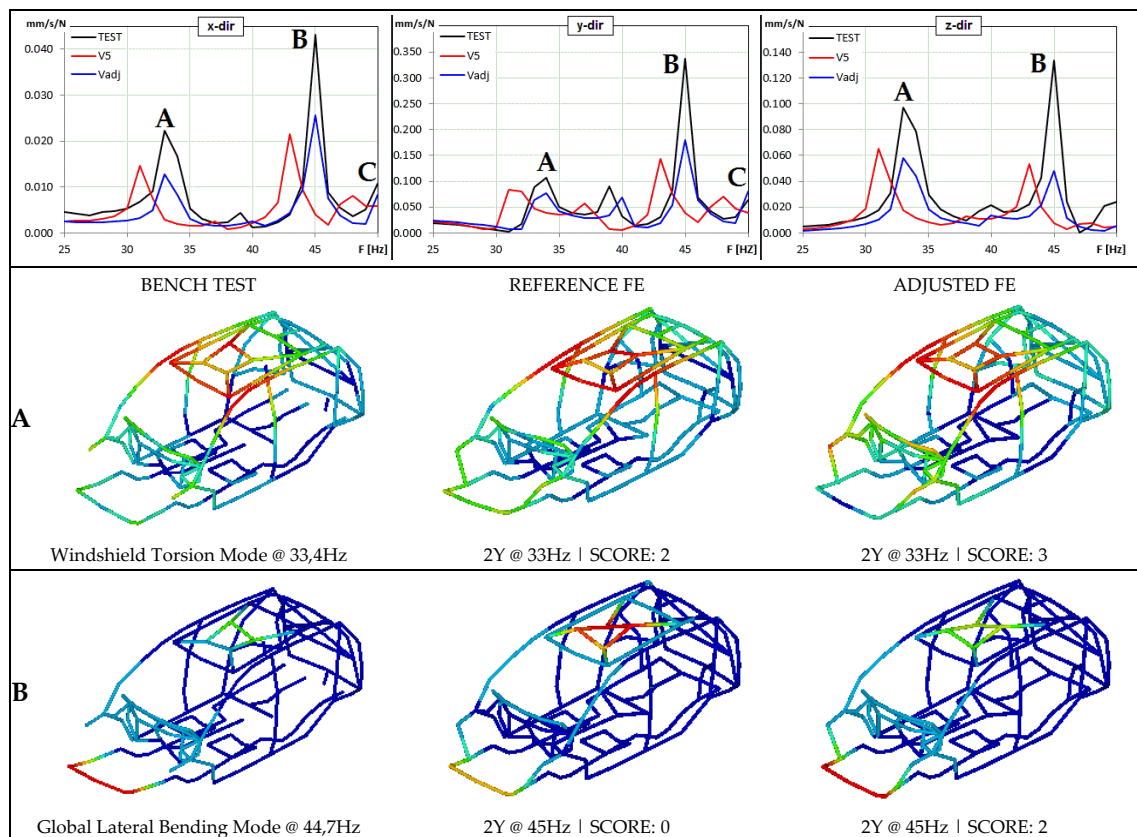


Figure 76. Experimental test *vs* reference *vs* adjusted FE model front right shock tower x-dir FRFs, modal shape responses and similarity scores.

Then, in the case where the body structure is excited at the front right shock tower in the y-direction, three relevant response peaks are noticed at 33Hz , 45Hz, and 50Hz (Figure 77). At peak A (33 Hz) the adjusted FE model closely follows the bench test results. The modal shape of the reference FE model also shows good correlation in part of the structure, but not following with accuracy the modal shape of the bench test at the roof and windshield structure areas. The reference FE model receives a score of 2 while the adjusted model receives a score of 3.

At the peak B (45 Hz) the adjusted FE model, despite of the amplitude, have the same frequency and similar modal shape as the bench test results. On the other hand, the reference model does not have a good correlation with the modal shape of the bench test in most of its structure, such as the front bumper, the roof and also the windshield structure areas. For these reasons, the reference FE model receives a score of 0 while the adjusted model receives a score of 2.

The peak C analysis (50Hz) shows that, while the reference FE model do not correlate with the bench test transfer functions, the adjusted model curves follow them closely at x and y-dir. Also, the adjusted model modal shapes are more similar to the bench tests than the reference, mainly at front bumper structure. Scores 1 and 2 are assigned to the reference and adjusted FE models.



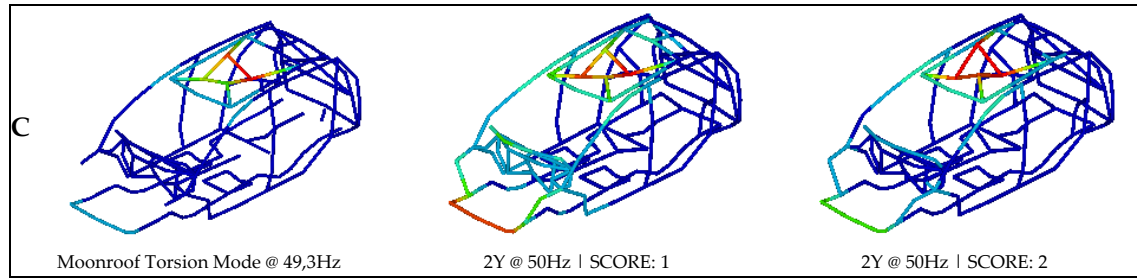


Figure 77. Experimental test *vs* reference *vs* adjusted FE model front right shock tower y-dir FRFs, modal shape responses and similarity scores.

When the body structure is excited at the front right shock tower in the z-direction, two relevant response peaks are noticed at 34Hz and 40Hz (Figure 78).

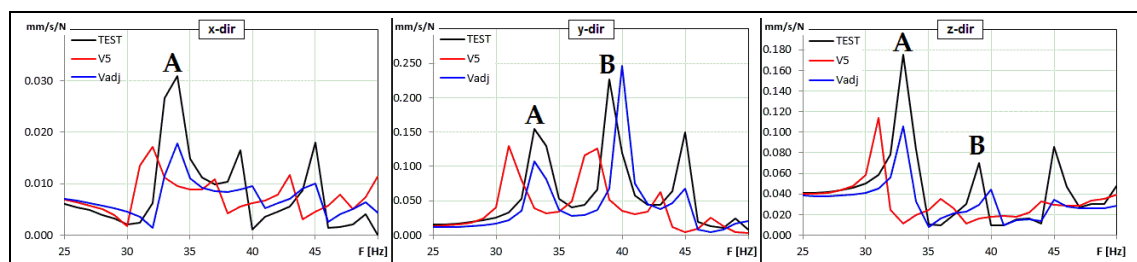
Analyzing the modal shapes of the first peak (A at 34Hz), it is noticed that the structures of the front bumper and the lower part of the windshield of the reference FE model do not move, diverging from the modal behavior of the bench test.

In contrast, the modal shape of the adjusted FE model follows with good correlation level the modal shape of the bench test considering the whole body structure.

Also, the transfer functions of the adjusted FE model at this peak have their frequency aligned with the bench test results, while the transfer functions of the reference FE model are between two and three hertz lagged. So, scores 2 and 3 are respectively assigned to the reference and adjusted FE models.

Next, the analysis of the peak B (40Hz) shows that the modal shapes of the reference and adjusted FE models are very similar to the modal shape of the bench test. However, it can be seen that the rear structure and the front bumper areas of the adjusted FE model follow slightly better the modal shape of the bench test than the reference model, making noticeable its higher level of correlation.

In addition to this, considering the peaks at this frequency, it can be noticed that the transfer functions of the adjusted FE model follow the bench test curves accurately than the reference model, specially considering y and z-dir. As a result, a score 2 is assigned to the reference FE model while a score 3 is assigned to the adjusted.



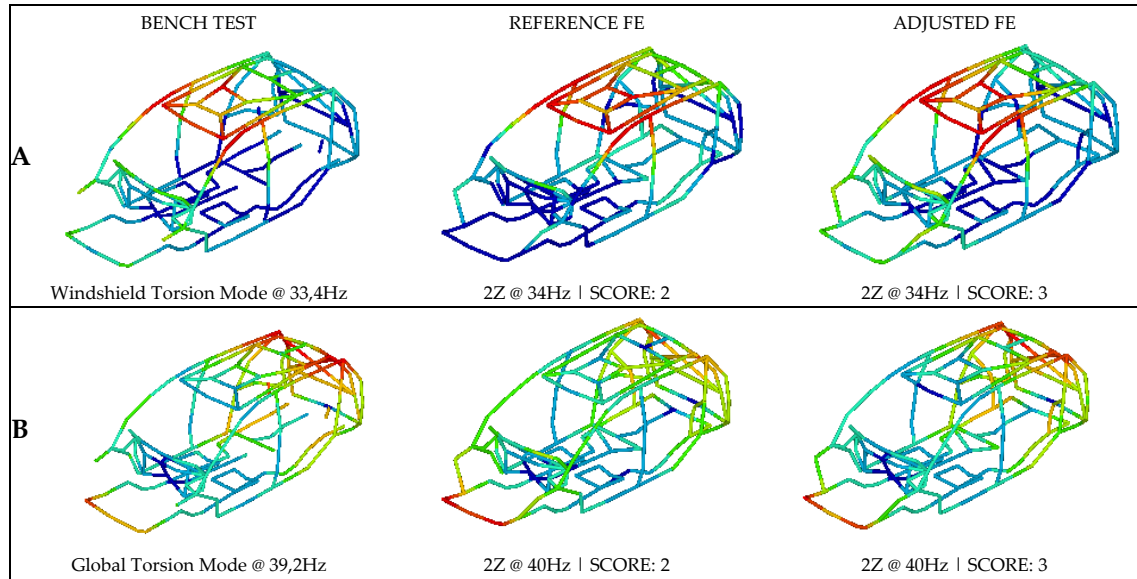


Figure 78. Experimental test *vs* reference *vs* adjusted FE model front right shock tower z-dir FRFs, modal shape responses and similarity scores.

Now, the point mobility responses at the front right shock tower of the bench test results, the reference, and the adjusted FE models (Figure 79) are compared. Like seen at the left shock tower curve, also here the adjusted FE model correlates better with the bench test results than the reference FE model. Some differences of amplitude are noticed at the adjusted curves at peaks A, B, C and D too. A better correlation is also noticed along the frequency range E, despite of no peaks are seen there.

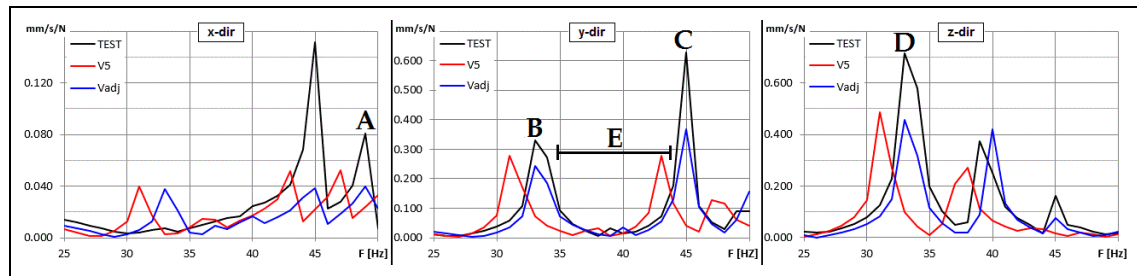


Figure 79. Experimental test *vs* reference *vs* adjusted FE model front left shock tower x, y and z-dir point mobility FRFs.

Next, when the body structure is excited at the rear left shock tower in the x-direction, three relevant response peaks are noticed at 33Hz, 39Hz and 45Hz (Figure 80).

At the peak A (33Hz), the transfer function of the reference FE model is around 2Hz lower than the bench test results while the adjusted curves follow the test frequency accurately. A high correlation level is noticed when comparing the adjusted FE model with the bench test. However, this correlation is slightly lower when comparing the reference model. So, a score 3 is assigned to the adjusted model while a score 2 is assigned to the reference.

At peak B (39Hz) the modal shape of the reference model does not correlate with the bench test in any area of the structure. In contrast, the modal shape of the adjusted model, even not precisely following the bench test results, shows higher level of similarity. The transfer function analysis also shows the reference FE model with no correlation with the bench test results, while a higher similarity level is noticed at the adjusted model curve. As a result, a score 0 is given to the reference FE model, while a 2 is given to the adjusted.

At peak C (45Hz) a low level of similarity is noticed comparing the reference model with the bench test results, since only the structure of the front bumper follows its modal shape. On the other hand, the modal shape of the adjusted model shows good correlation with the bench test, following its modal shape at most of the structure areas. A similar behavior can be seen comparing the transfer functions, since there is no similarity between the reference FE model and the bench test while in the y-dir the adjusted model follows the test curve. A score 1 is assigned to the reference FE model while a score 2 is assigned to the adjusted.

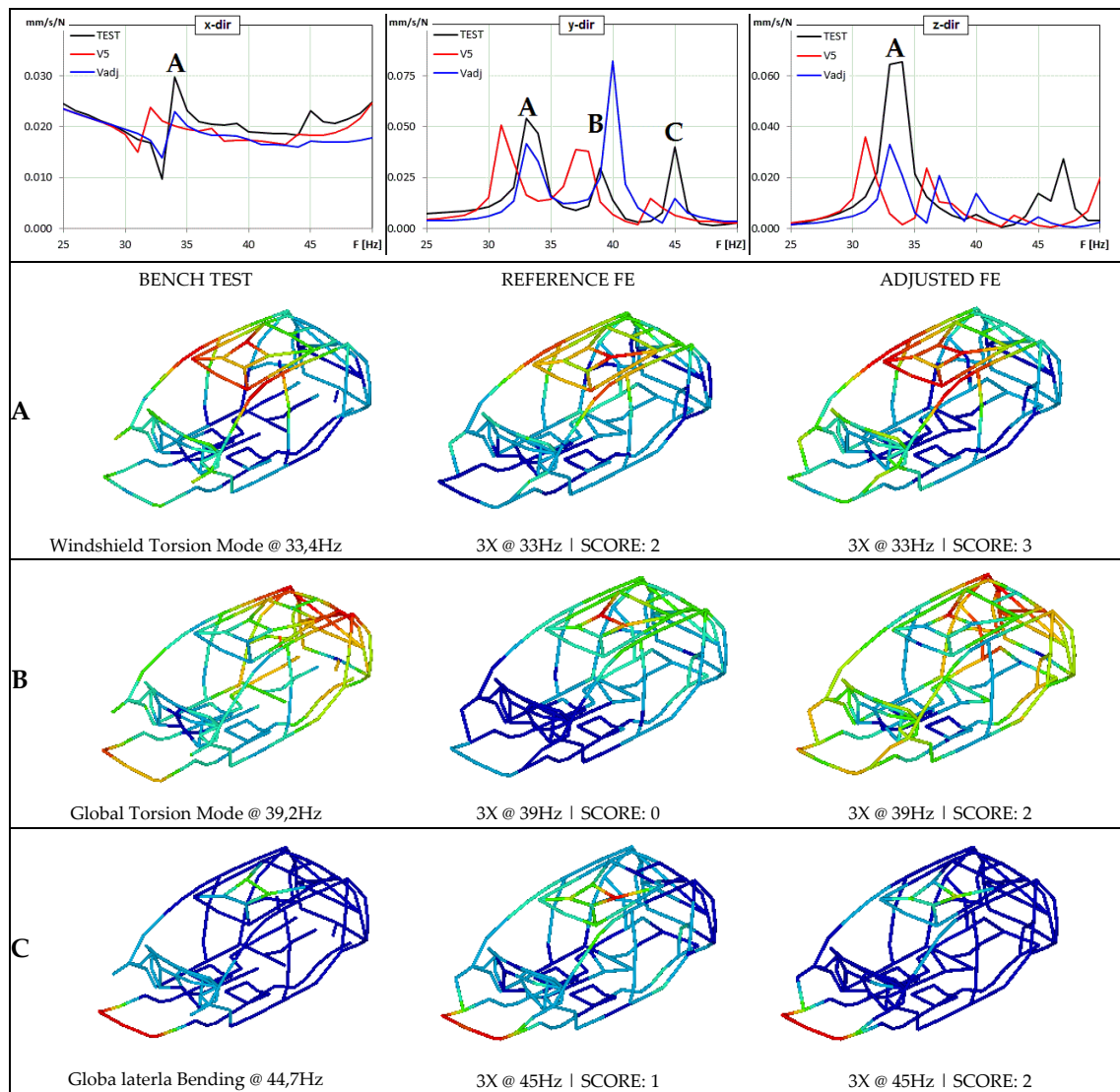


Figure 80. Experimental test *vs* reference *vs* adjusted FE model rear left shock tower x-dir FRFs, modal shape responses and similarity scores.

4.2. Reference vs Adjusted FE Model Results Comparison

In the case where the body structure is excited at the rear left shock tower in the y-direction, two relevant response peaks are noticed at 39Hz and 45Hz (Figure 81).

Considering the peak A (39 Hz) the analysis shows that both modal shapes of the reference FE model and adjusted FE model follow the bench test result, except for the front bumper structure in the reference model and the rear structure in the adjusted FE model.

Comparing the transfer functions, the reference FE model peaks have a 2Hz delta when compared with the bench test curve while the adjusted only 1Hz. Considering the overall results similarity, a score of 2 is assigned for both FE models.

At peak B (45 Hz), the reference FE model modal shape does not follow the bench test results at the roof, front bumper and rear structure areas. In contrast, the adjusted FE model modal shape accurately follows the bench test results, despite of an amplitude delta shown at the moonroof area. A similar behavior is noticed comparing the transfer functions. The reference FE model curves do not follow the bench test results while the adjusted, despite of an amplitude delta, have the same peak frequencies. So, scores 0 and 2 are given to the reference and adjusted FE models.

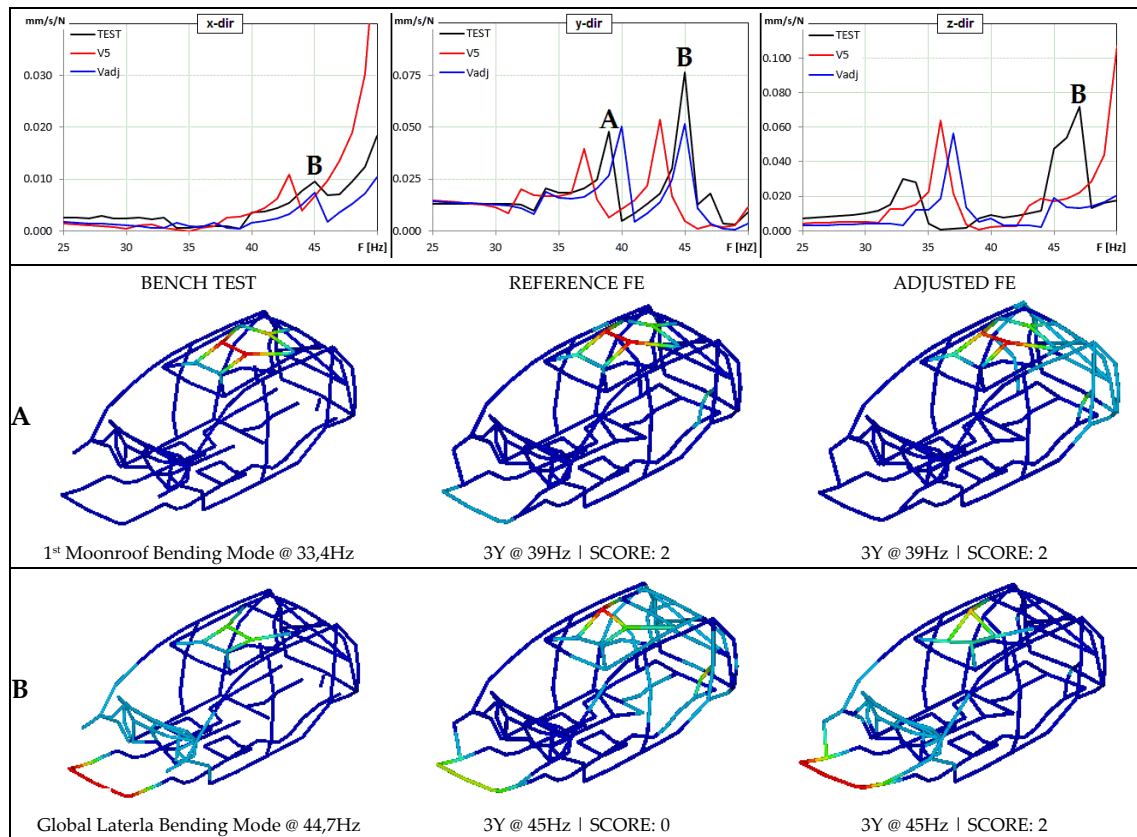


Figure 81. Experimental test *vs* reference *vs* adjusted FE model rear left shock tower y-dir FRFs, modal shape responses and similarity scores.

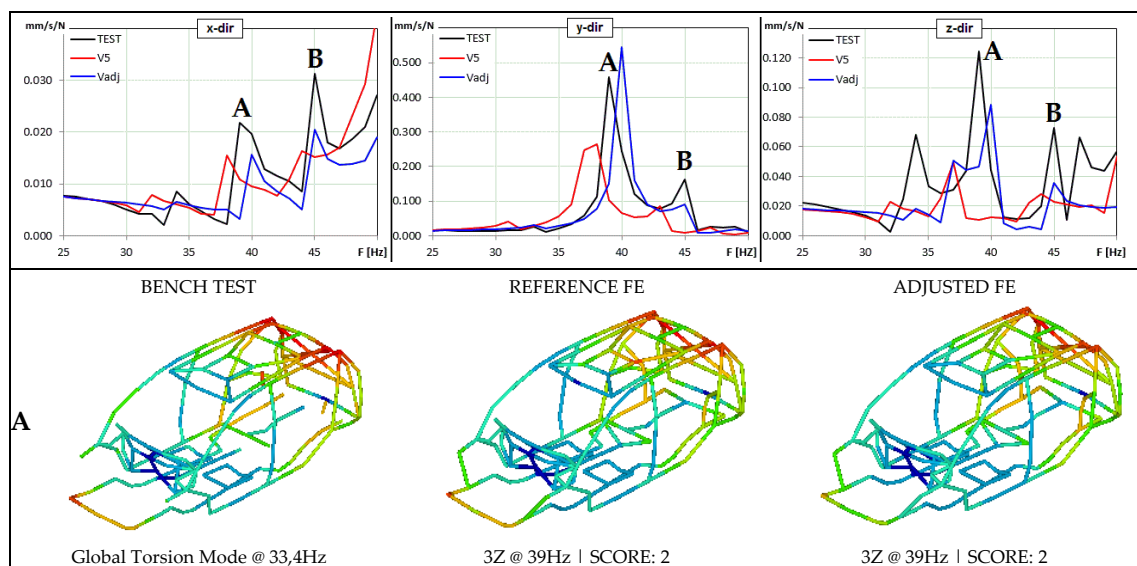
In the case where the body structure is excited at the rear left shock tower in the z-direction, two relevant response peaks are noticed at 39Hz and 45Hz (Figure 82).

The peak A (39 Hz) analysis shows that the modal shapes of the reference and adjusted FE models have very similarly behavior to the modal shape of the bench test. Indeed, comparing the modal shape of both models to each other and also against the bench test it is noticed that only small differences are seen in the rear structure and front bumper structure areas, remaining almost the same throughout the structure.

The analysis of the transfer function also shows a similar level of correlation between the curves of the reference and adjusted FE models. However, in this case the peak of the reference FE model is located 1Hz before the peak of the bench test curve, while the peak of the adjusted FE model is 1Hz after it. As a result, the same similarity score of 2 points is assigned for both reference and adjusted FE models.

At peak B (45Hz), the analysis of the modal shapes shows that there is no good correlation between the reference FE model and the bench test results. The modal behavior of the rear structure and front bumper structure areas, which characterize the global lateral bending mode, are not noticed in this model. On the other hand, it can be seen that the modal shape of the adjusted FE model follows the modal behavior of the bench test in these areas of the structure, showing its good level of correlation.

In the same way, the reference FE model transfer function do not follow the bench test curve at peak B. In contrast, the transfer function peaks of the adjusted FE model, despite of an amplitude difference, are find at the same frequency of the bench test curves. So, scores 0 and 2 are assigned to the reference and adjusted FE models respectively.



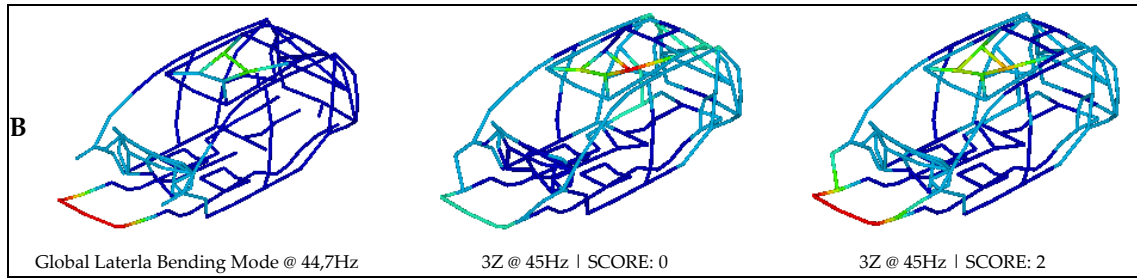


Figure 82. Experimental test *vs* reference *vs* adjusted FE model rear left shock tower z-dir FRFs, modal shape responses and similarity scores.

Now, the point mobility responses at the rear left shock tower of the bench test results, the reference, and the adjusted FE models (Figure 83) are compared. In this case, the better accuracy of the adjusted FE model curve is noticed along the curves when compared with the reference model, specially at y and z directions.

The peak A stands at the same frequency of the bench test and the adjusted curve follows the bench test results for a range of approximately 10Hz before and after the peak. The curve into range C, which is also the fully range of analysis, follows the bench test in the y direction. In z direction the point mobility curve of the adjusted FE model is very close to the test at both D range and B peak. All these conditions confirm the high correlation level achieved.

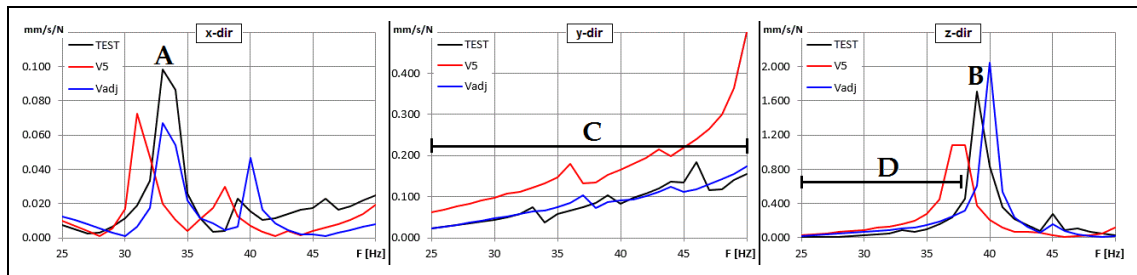


Figure 83. Experimental test *vs* reference *vs* adjusted FE model rear left shock tower x, y and z-dir point mobility FRFs.

In the case where the body structure is excited at the rear right shock tower in the x-direction, three relevant response peaks are noticed at 33Hz, 39Hz and 45Hz (Figure 84).

The first peak analyzed (peak A, at 33 Hz) shows good level of correlation not only between the reference FE and adjusted FE modal shapes, but also between these and the bench test, with the exception of the front bumper area of the reference model. The transfer functions analysis shows a 2Hz delay in the reference FE model curve respect the bench test, while the adjusted model copies it accurately. As a result, a score 0 is assigned to the reference FE model while a score 1 is assigned to the adjusted.

The peak B (39Hz) analysis shows low correlation between the reference FE model and the bench test modal shapes at the whole structure. The adjusted FE model, even not having high correlation with the bench test, shows some similarities at the rear and front bumper areas.

The transfer function of the reference mode also shows low similarity level with the bench test curve at this peak, which can be noticed for the adjusted FE model curve too. In this case, a score 2 is given to the reference FE model and a score 3 to the adjusted.

Then, the peak C (45Hz) analysis shows a low level of correlation between the modal shape of the reference FE model against the bench test results. On the other hand, the analysis of the adjusted FE model shows good similarity between its modal shape and the bench test.

The analysis of the transfer functions also shows that the adjusted FE model has a better correlation level than the reference, but with slightly better precision only. A score 0 is assigned to the reference while a score 2 is assigned to the adjusted FE model.

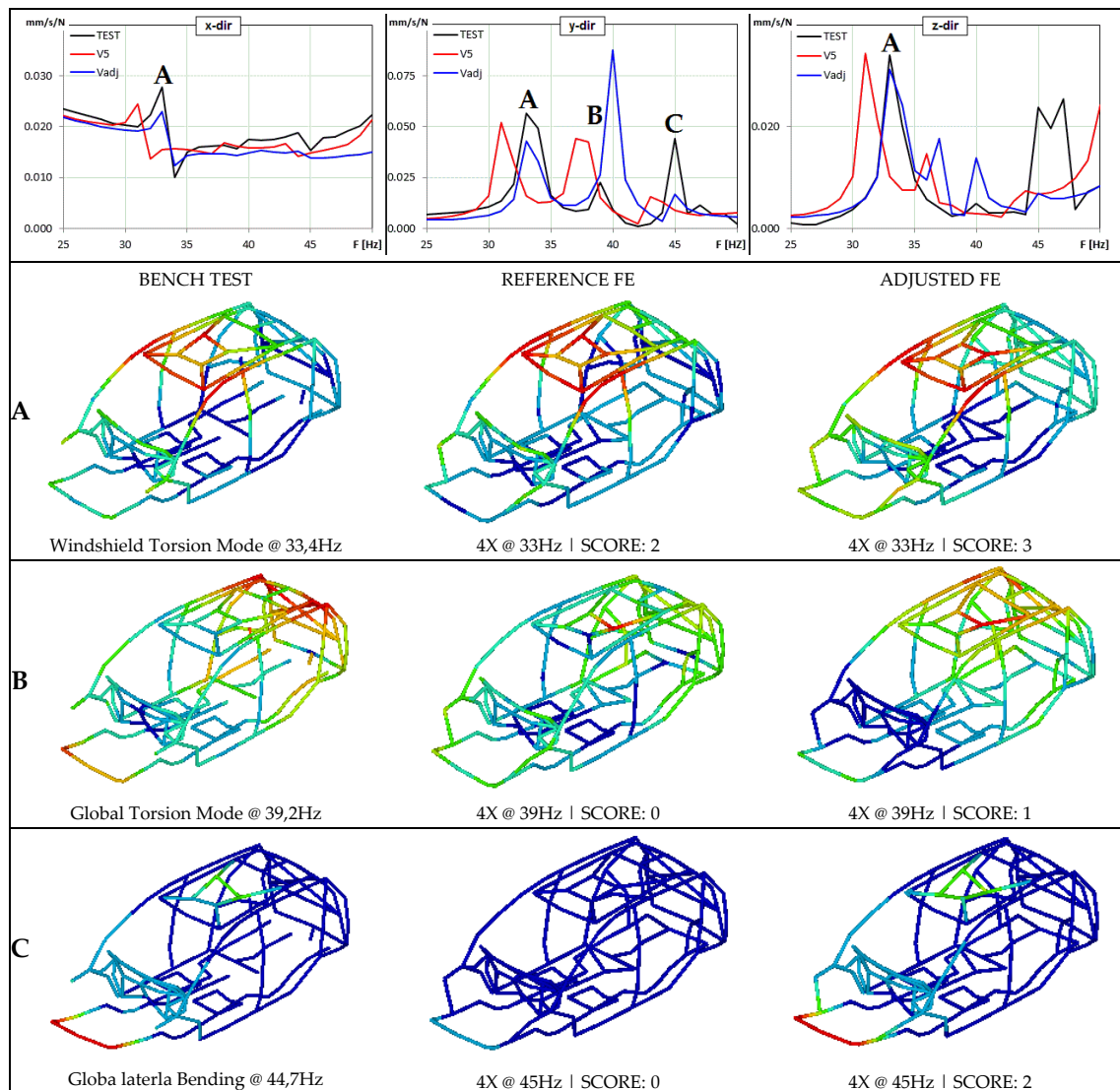


Figure 84. Experimental test *vs* reference *vs* adjusted FE model rear right shock tower x-dir FRFs, modal shape responses and similarity scores.

4.2. Reference vs Adjusted FE Model Results Comparison

In the case where the body structure is excited at the rear right shock tower in the y-direction, two relevant response peaks are noticed at 39Hz and 45Hz (Figure 85).

The peak A (39 Hz) shows good correlation between the adjusted FE model and the bench test modal shapes, despite small differences in the front bumper and rear structure. The reference FE model shows similar behavior, however having these differences more pronounced. The transfer function of the adjusted FE model is also closer to the bench test curve than the reference, showing better correlation too. So, scores 1 and 2 are assigned to the reference and the adjusted FE model.

The B peak analysis (45 Hz) shows that the modal shape of the reference FE model does not correlate with the bench test. In contrast, the modal shape of the adjusted FE model follows the behavior of the bench test, clearly showing the lateral bending mode. In the same way, the transfer function of the fitted model has a good correlation with the bench test curve, while the reference model does not correlate. As a result, a score 0 is given to the reference FE model while a score 2 is given to the adjusted.

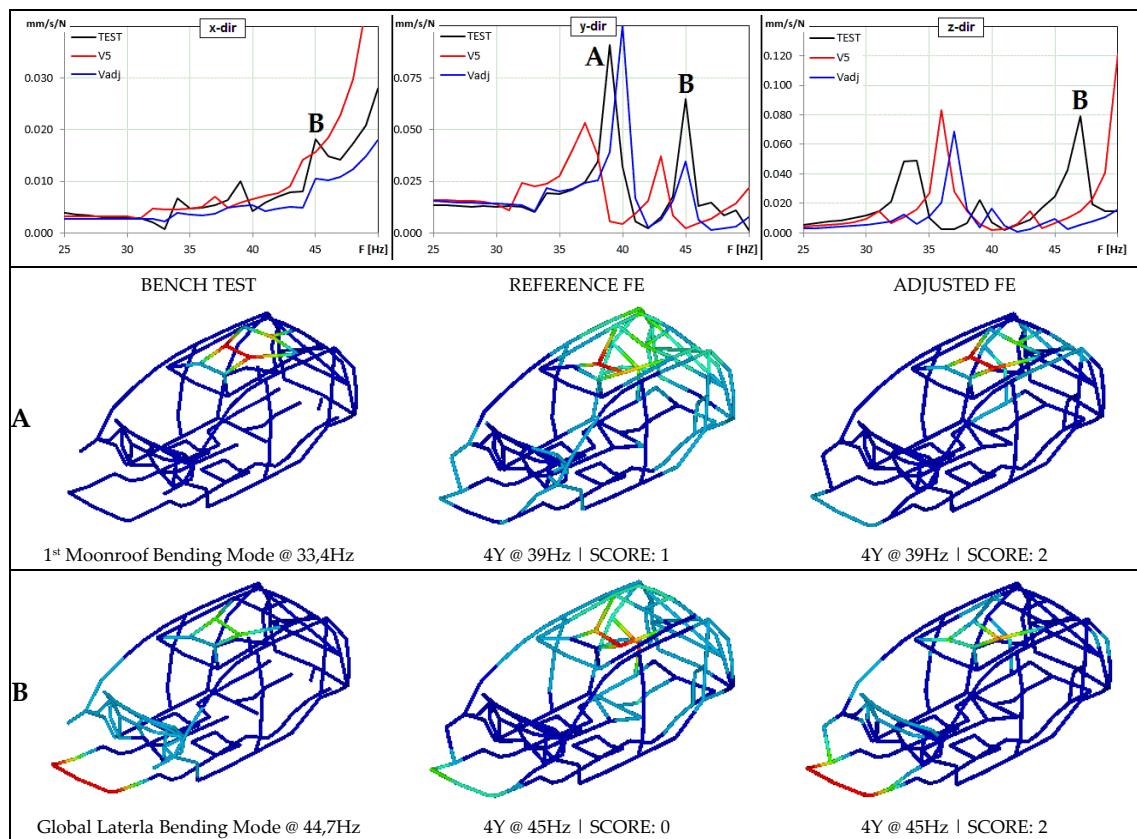


Figure 85. Experimental test *vs* reference *vs* adjusted FE model rear right shock tower y-dir FRFs, modal shape responses and similarity scores.

In the case where the body structure is excited at the rear right shock tower in the z-direction, two relevant response peaks are noticed at 39Hz and 45Hz (Figure 86).

The analysis of the peak A (39 Hz) shows that the modal shapes of the reference and the adjusted FE models are similar to each other considering all areas of the body structure. In addition to this, both FE models have good correlation with the modal shape of the bench test in all areas of the body structure.

The analysis of the transfer functions also shows a similar level of correlation between the reference and adjusted FE model curves and the bench test results. But in this case, the curve of the adjusted FE model correlates slightly better than the one shown by the reference FE model. Because of this difference, a score 2 is assigned to the reference FE model, while a score 3 is assigned to the adjusted.

Next, the peak B (45Hz) analysis shows that the modal shape of the reference FE model do not correlate well with the bench test, specially is the front bumper and rear structure areas. In contrast, despite of having opposite phase at the rear structure area, the adjusted FE model follows the test results modal shape accurately along the whole body, making the lateral bending mode correlation clear.

Also, the reference FE model transfer function has a 2Hz delay with respect to the bench test curve, but the adjusted FE model transfer function follows the bench test curve precisely when considering the peak frequency. So, a score 0 is given to the reference FE model and a score 2 to the adjusted.

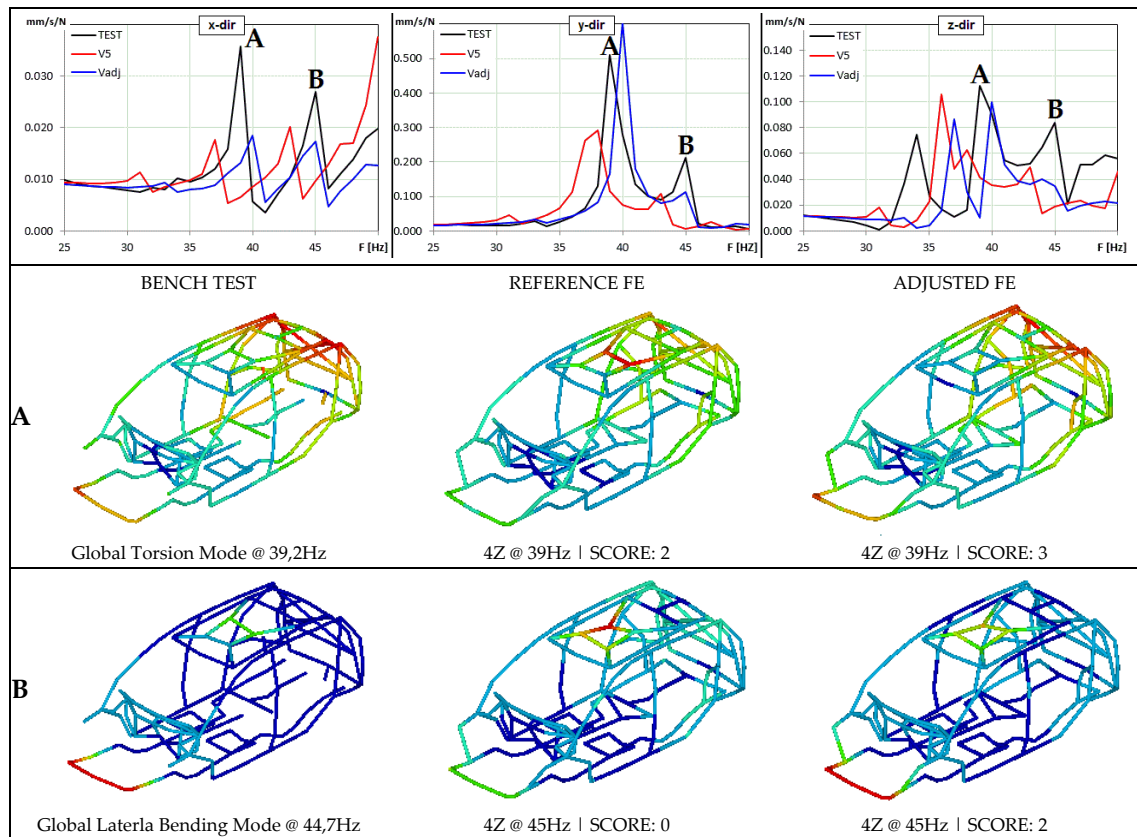


Figure 86. Experimental test *vs* reference *vs* adjusted FE model rear right shock tower z-dir FRFs, modal shape responses and similarity scores.

4.2. Reference vs Adjusted FE Model Results Comparison

Now, the point mobility responses at the rear right shock tower of the bench test results, the reference, and the adjusted FE models (Figure 87) are compared.

Like at the rear left shock tower point mobility comparison, also here the better accuracy of the adjusted FE model curve is noticed along the curves at the whole range of frequency, when compared with the reference model at all directions.

The analysis of the peaks A and B shows that the adjusted FE model peaks are closer to the bench test curve peaks than the reference FE model. In addition, considering the overall shape of the curves, both y and z directions of the adjusted model follows accurately the bench test curves considering the whole frequency ranges C and D.

Furthermore, considering the x direction, the adjusted model follows the bench test in most of the frequency range, reassuring its higher correlation level with the bench test results when compared against the reference FE model.

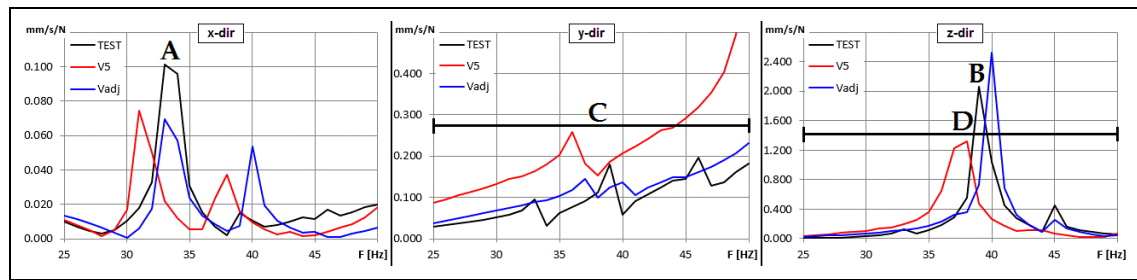


Figure 87. Experimental test *vs* reference *vs* adjusted FE model rear right shock tower x, y and z-dir point mobility FRFs.

4.2.2.2. Similarity Score Comparison

The similarity scores assigned to each peak of both reference and adjusted FE model transfer function are summarized in the Table 27:

EXCITATION POINT	DIRECTION	FREQUENCY	SIMILARITY SCORE	
			REFERENCE MODEL	ADJUSTED MODEL
FRONT LEFT SHOCK TOWER	x	34	1	3
	x	39	1	2
	x	45	0	3
	y	33	2	3
	y	45	0	1
	y	50	1	2
	z	34	1	2
	z	40	2	3

FRONT RIGHT SHOCK TOWER	x	34	2	3
	x	39	1	2
	x	45	1	3
	y	33	2	3
	y	45	0	2
	y	50	1	2
	z	34	2	3
	z	40	2	3
REAR LEFT SHOCK TOWER	x	33	1	3
	x	39	1	2
	x	45	0	3
	y	39	2	2
	y	45	0	2
	z	39	2	2
	z	45	0	2
REAR RIGHT SHOCK TOWER	x	33	2	3
	x	39	0	1
	x	45	0	2
	y	39	1	2
	y	45	0	2
	z	39	2	3
	z	45	0	2
TOTAL SCORE			30	71

Table 27. Similarity scores from reference and adjusted FE model modal shape correlation analyses.

The analysis of the transfer functions and point mobility curves shows that the adjusted FE model modal shapes better simulate the bench test modes than the reference FE model across the entire structure and in all frequency ranges.

Both the stiffness adjustments, which are mainly reflected in frequency variations, as well as the damping adjustments, more evident in amplitude variations, are efficient in provide more representative results with higher correlation levels.

In many cases the adjusted model simulation is able to copy the bench test results, which is never reached by the reference model. In the other hand, even in the worst case the adjusted model has a certain level of similarity, keeping its confidence level high.

Therefore, the simulations obtained from the adjusted model are more accurate than those generated from the reference model, which in the end means gains in quality, time and resources along the development of a project.

4.2.3. Frequency Response Function Statistical Analysis

Here, a set of analyzes is performed to determine how much the data set of each FE model statistically differs from the bench test measurement results set. For this, the frequency and amplitude of the bench test FRF peaks already analyzed in the previous section are measured. Then, the correspondent frequency and amplitude of the reference and adjusted FE model FRF curves are also measured.

So, the frequency and amplitude deltas between the bench test peaks, against the reference and adjusted FE model peaks is acquired. Based on this, the minimum, the maximum, and the medium absolute error (MAE) of these deltas are calculated.

Initially, the frequency and amplitude deltas are plotted in a histogram (Figure 88), making it possible to visualize their distribution and compare the correlation level between the results of the reference and adjusted FE model, and the bench tests.

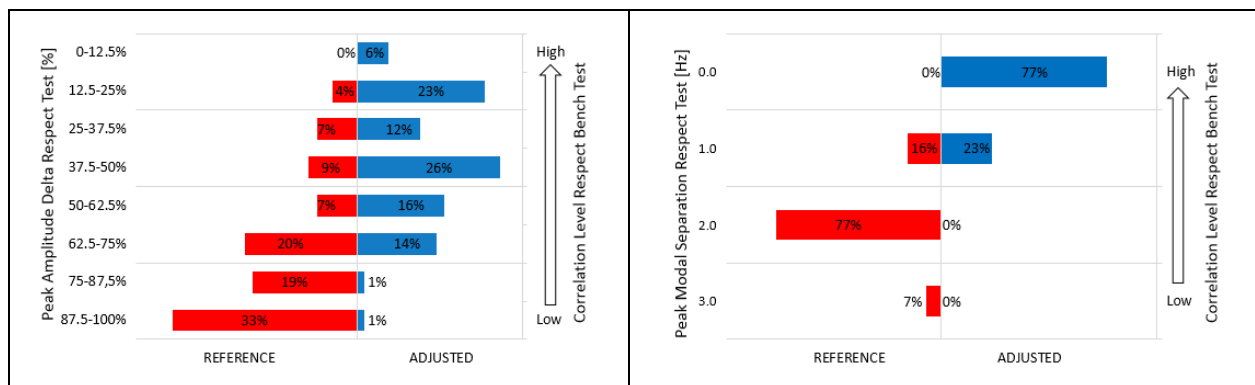


Figure 88. Reference and adjusted FE model histograms with amplitude (left) and frequency (right) FRF peaks distribution.

As a result, the distribution of the reference FE model peaks are concentrated in the lowest lower quartiles, where the correlation with the bench test results is lower. In contrast, analyzing the peaks of the adjusted FE model, it is noticed that its distribution is concentrated in the upper quartiles, where the correlation with the bench test results is higher.

As a result, considering the reference FE model, where the simulation amplitude values are very far from those they are intended to represent, higher criticality peaks can be neglected while other points of less importance are considered for analysis, as if they were the root cause of some excessive vibration problem. In contrast, this issue is not expected to be found with the adjusted FE model due to its distribution.

This statement is also very critical for the frequency distribution of the reference FE model. As already noticed along the modal shapes analysis, modal separations above 2Hz can jeopardize the characterization

of the eigenmodes of a system, usually leading its simulation to a misrepresentation of the real modal shape intended to be represented. On the other hand, considering the adjusted FE model, it can be noticed the absence of low correlation points (last quartile) in its distribution, leading to a higher correlation with the bench test results.

Next, the minimum, maximum and MAE of the reference and adjusted FE model FRF frequency and amplitude deltas respect the bench test results (Figure 89) are plotted.

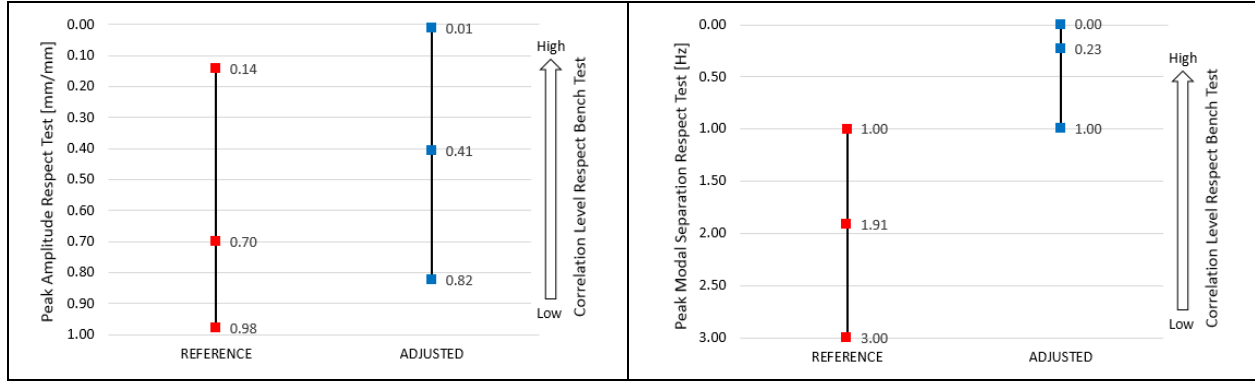


Figure 89. Reference and adjusted FE model box plots with minimum, maximum and MAE respect the amplitude (left) and frequency (right) bench test results.

The box plot graphics confirm the higher correlation already indicated by the histograms. In this case, aside of the lowest average amplitude and frequency deltas, it is also noticed that the minimum and maximum values are lower, as well as the function amplitude (difference between maximum and minimum values) of the adjusted FE model data sets when compared with the reference.

In addition to this, the medium squared error (MSE) is calculated in terms of amplitude and frequency. The use of this metric aims here to evaluate the difference between the results calculated by the simulations of the reference and adjusted FE models and the measures acquired from the bench test. The square weights penalize larger deltas between predicted values and test results by highlighting peak deviations, which is consistent with the directional that this metric is expected to provide. The MSE is calculated as follows [35]:

$$MSE = \frac{1}{n} \sum_{i=1}^n (\hat{y}_i - y_i)^2 \quad (5)$$

where $(\hat{y}_i)$ and (y_i) are respectively the result predicted by the simulations and the value acquired with the bench test. Considering this, the reference and adjusted FE model MSE (in m^2 units) for the amplitude and frequency deltas are:

Amplitude:	$MSE_{A-REF} = 0.0113$	$MSE_{A-ADJ} = 0.0057$
Frequency:	$MSE_{F-REF} = 3.8841$	$MSE_{F-ADJ} = 0.2319$

From the MSE results calculated by equation (5), it is shown in rounded numbers that the adjusted FE model amplitude results are twice as accurate as those obtained by the reference FE model and also, that the adjusted frequency results, are seventeen times more accurate than the reference FE model.

So, based in all statistical analysis outcomes, and considering that the closer the simulation results are to the test data, the more accurate this model is, it is concluded that, as already confirmed by the FRAC and FRF comparison analyzes, the simulations obtained from the adjusted FE model are more accurate than those generated from the reference FE model.

4.3. Optimization Analysis

In addition, a weight optimization analysis of the vehicle body structure is performed to evaluate the isolated effect of the proposed methodology. In this way, the enhancement provided can be quantified in terms of weight and cost savings. The following restrictions are also considered to perform the optimization analysis:

- Disregard panels, since they are already set to the minimum gauge allowed;
- Disregard component brackets, since their main function is not related to the main structure, but to component tests;
- Avoid gauge changes smaller than 0.1 mm due to manufacturing restrictions;
- Avoid gauge changes higher than 15% of the initial gauge (if $t < 1.0$ mm) due to manufacturing restrictions for new tooling;
- Avoid gauge changes higher than 10% of the initial gauge (if $t \geq 1.0$ mm) due to manufacturing restrictions for new tooling;
- Disregard critical parts related to frontal and lateral impact tests, since their main function is related to crash analysis; and
- Disregard critical parts related to chassis and powertrain attachments, since their main function is related to fatigue durability analysis.

The following calculation deck (Table 28) is used to perform the weight optimization analysis on the adjusted FE model of the vehicle body structure. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
SOL 200	Design Optimization solution, in this case applied to weight optimization
CEND	Designates the end of the Executive Control Section
ECHO=SORT (DESVAR, PARAM, DRESP1, DRESP2, DCONSTR)	Print bulk data response options
DESOBJ (MIN) =1	Defined the optimization objective variable and function, in this case to minimize the weight having the components thickness as variable parameter
DSAPRT (FORMATTED, NOEXPORT, END=SENS) = ALL	Specifies design sensitivity output parameters
DISPLACEMENT (PUNCH) = ALL	Requires displacement as output result parameter
ESE (PUNCH) = ALL	Requires strain energy density as output result parameter
SUBCASE 1	Delimits and identifies the subcase
ANALYSIS=STATICS	Defines the analysis solution to be applied for the optimization problem
LABEL= OPTIMIZATION	Defines the subcase label
LOAD = 2	Defines a static load set
SPC = 2	Defines a Single Point Constrain set, in this case, to be used as boundary condition
DESSUB=2001	Selects the design constraints to be used in a design optimization task for the current subcase
BEGIN BULK	Designates the start of the Bulk Data Section
SPC 2 18 123 0.0	Specifies the Single Point Constrain set nodes and degree of freedom
SPC 2 19 13 0.0	
SPC 2 100 3 0.0	
FORCE 2 11 1.0 0.0 0.0 2646.23	Specifies the load set where the loads are applied, as well as the force magnitude and direction
FORCE 2 12 1.0 0.0 0.0 -2646.23	
DOPTPRM DELP 0.05 DESMAX20 CONV2 1.0E-6 P1 1 P2 15	Overrides default values of parameters used in design optimization
PARAM, OPTEXIT, 4	Select the design sensitivity variable, in this case, the components thickness

DSCREEN DISP -100.0 9999	Defines screening data for constraint deletion
	Defines a set of structural responses that are used for design constraints.
DRESP1 1 WEIGHT WEIGHT	DRESP1: body parts weigh sensitivity
DRESP1 101 FSH_L DISP 3 11	DRESP 101: left side shock tower displacement in z direction
DRESP1 102 FSH_R DISP 3 12	DRESP 102: right side shock tower displacement in z direction
DRESP2 200 TORSTIFF166 + + DRESP1 101 102 + DNODE 11 2 12 2	Defines equation responses that are used for sensitivity analysis purposes
DEQATN 166 KT(LW,RW,LY,RY) = 3.0/((ABS(LW)+ABS(RW))/(ABS(LY)+ABS(RY)))	Defines a design variable for design optimization
DESVAR 1 LABEL 0.80 0.60 0.60	Defines the design variable lower and upper limits
DVPREL1 1 PSHELL 50201 4 + 60 1.0	Defines the relation between an analysis model property and design variables
END DATA	Designates the end of the Bulk Data Section

Table 28. Optistruct deck used for weight optimization analysis on the adjusted FE model

The optimization analysis provide a gauge reduction in four parts of the vehicle body structure (Figure 90), resulting in a weight saving of 0.566 Kg.

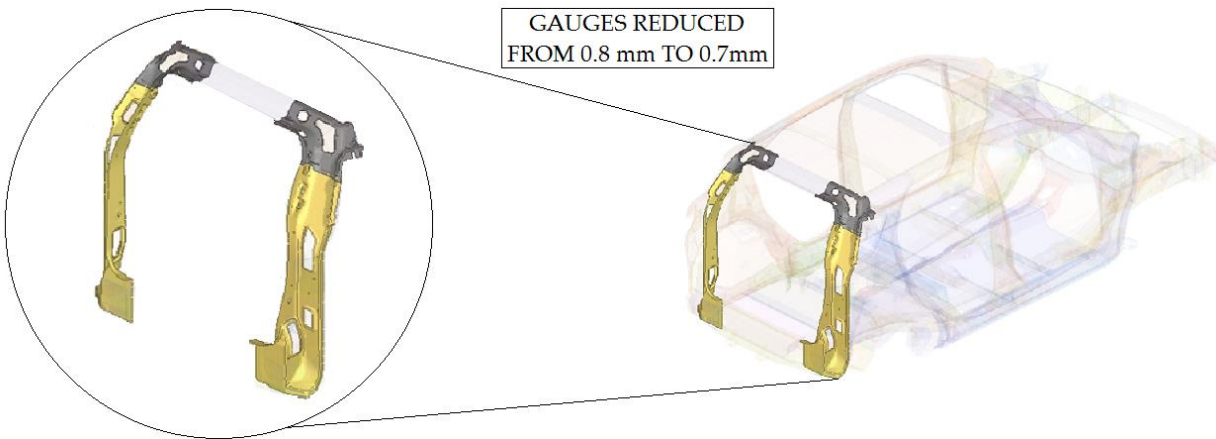


Figure 90. Optimized adjusted FE model of the vehicle body structure highlighting the parts with gauge reduction.

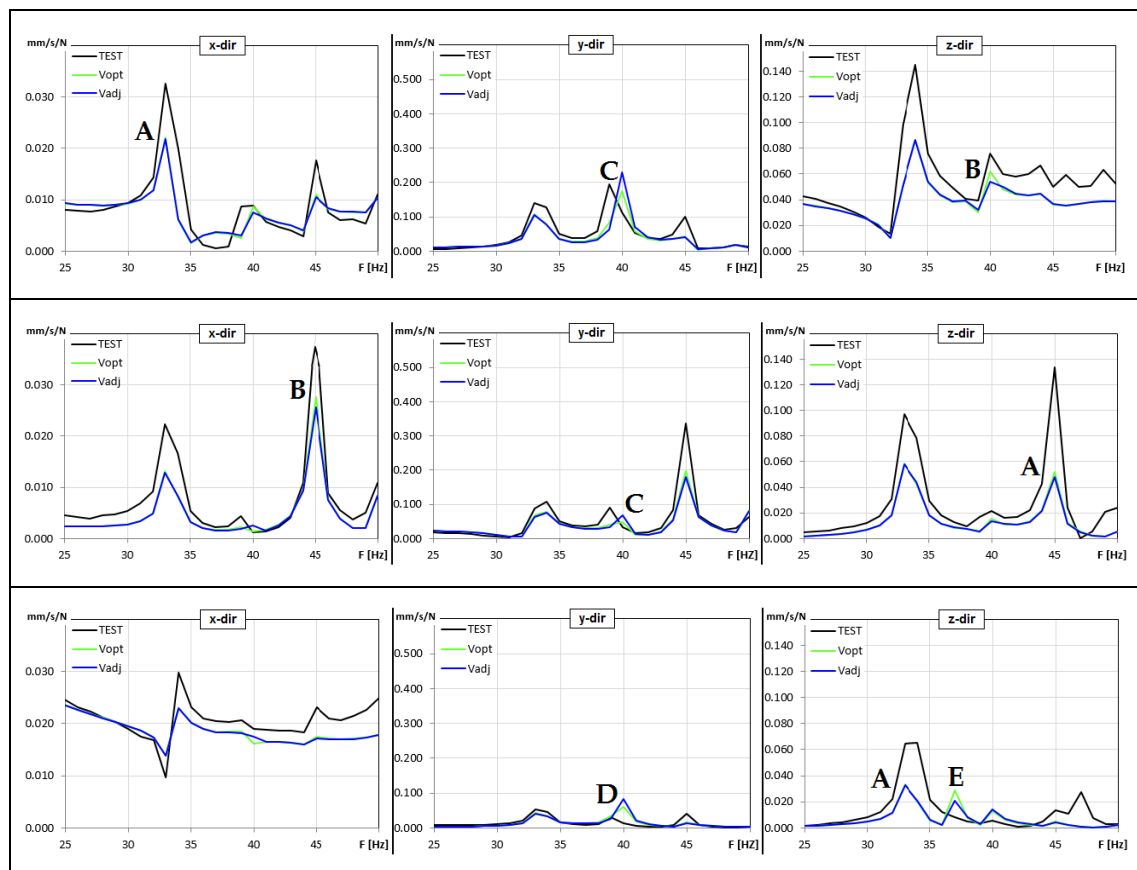
As a result, the peaks of the Optimized FE model FRF shows only slightly changes when compared with the Adjusted FE model.

As a result, the peaks of the optimized FE model FRFs show only small changes when compared to the curves of the adjusted FE model. No frequency changes and also no large amplitude changes are noticed.

In many cases, no changes can be noticed (Figure 91, peaks "A"). In some cases, the optimized peaks move toward the direction of the bench test curves but without exceeding their maximum amplitude (Figure 91, peaks "B" and "C").

And in other cases, the optimized peaks move toward the direction of the FE model curves, but without showing large variations of amplitude (Figure 91, peaks "D" and "E"). In all of those cases, as the frequency and amplitude levels are preserved, no behavior change is expected respect the modal shapes of the severity level of these peaks.

The most relevant peaks at each shock tower are shown in the Figure 91. All other FRF graphics comparing the adjusted with the optimized FE model are shown in Appendix IX.



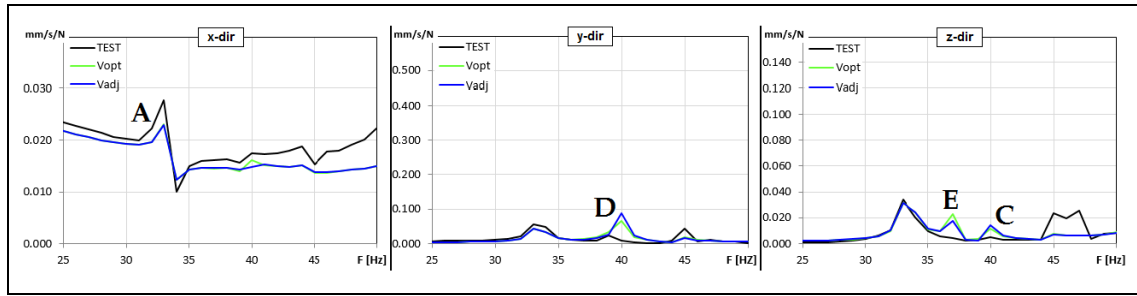


Figure 91. Experimental test *vs* gauge optimized *vs* adjusted FE model front left z-dir, front right y-dir, rear left x-dir and rear right x-dir shock tower FRFs.

4.4. Stress Analysis

In addition to this, the stresses on the down-gauged components (0.7 mm thickness on the optimized adjusted FE model) are also analyzed. They are compared with the stresses on the same components with the original gauge (0.8 mm thickness on the adjusted FE model). To this end, a linear static calculation is performed using an input torque of 3000 N.m applied in the front shock towers by opposite vertical loads.

Based on the vehicle manufacturer's experience in durability simulation test analysis, the rear shock towers attachment points are partially constrained in order to simulate a critical road test condition.

Then the predicted lifetime [36] for each condition is calculated using the Neuber plasticity correction and the Smith–Topper–Watson mean stress correction, considering the properties of the SAE-1020 material.

As a result, the original lifetime initially calculated decrease from 5.68110^6 to $4.120 \cdot 10^6$ cycles, but still complying with the material endurance limit [37].

The following calculation deck (Table 29) is used to perform the stress analysis on the adjusted optimized FE model of the vehicle body structure. The main commands and setup are detailed and commented next:

COMMANDS	COMMENTS
SOL 101	Static analysis solution
CEND	Designates the end of the Executive Control Section
SUBCASE 1	Delimits and identifies the subcase
LABEL= VM_STRESS-STRAIN	Defines the subcase label
LOAD = 2	Defines a static load set
SPC = 2	Defines a Single Point Constrains set, in this case, to be used as boundary condition

STRAIN (VONMISES) = 10	Requires Von-Mises strain output at the set of elements 10
STRESS (VONMISES) = 10	Requires Von-Mises stress output at the set of elements 10
DISPLACEMENT (PUNCH) = ALL	Requires displacement as output result parameter
ESE (PUNCH) = ALL	Requires strain energy density as output result parameter
SET=10	Defines the subcase set of elements for the results output subcase
BEGIN BULK	Designates the start of the Bulk Data Section
SPC 2 18 123 0.0	Specifies the Single Point Constrain set nodes and degree of freedom
SPC 2 19 13 0.0	
SPC 2 100 3 0.0	
FORCE 2 11 1.0 0.0 0.0 2646.23	Specifies the load set where the loads are applied, as well as the force magnitude and direction
FORCE 2 12 1.0 0.0 0.0 -2646.23	
END DATA	Designates the end of the Bulk Data Section

Table 29. Optistruct deck used for stress analysis on the adjusted optimized FE model

The maximum stress is also calculated and shown (Figure 92) in the components considering both the original and the down-gauged conditions.

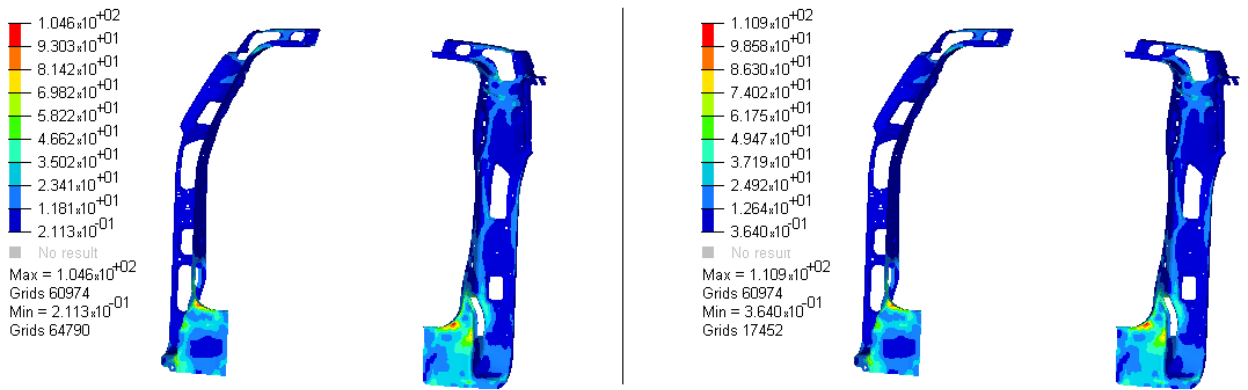


Figure 92. Maximum stress under torsional critical load on the adjusted FE model (left) and the adjusted optimized FE model (right).

The optimization analysis confirm the improvement of the MAC and FRAC coefficients given by the adjusted model, thus corroborating the significant improvement of the correlation level that the proposed adjustment methodology provides.

To better illustrate the impact of the modelling approach proposed, a weight-saving estimate as a result of this action is performed based on the following parameters:

- Sales in 2019 of the tested vehicle [38-40]: 282,371 units; and
- Weight savings per vehicle: 0.566 kg

As a result, an estimated material saving of 160 tons of steel per year is calculated taking in account only this vehicle model. Considering that the life cycle of a vehicle can extend for approximately ten years [3] and also the great diversity of vehicle models on the market, the total

savings can reach very impactful numbers not only in economic terms, but also in environmental and social issues.

In this case, the reduction in the use of steel, which brings about important cost savings, also helps to preserve natural resources.

In addition, the weight reduction provided by this optimization improves the vehicle's fuel economy performance, bringing another positive impact on the preservation of natural resources and the reduction of carbon emissions.

5. DISCUSSION

Improving the correlation level between FE simulations and field tests is a key factor in improving the time and cost-efficiency of many engineering projects. In particular, it is essential in the automotive industry. A possible solution to achieve this goal would be to add nonlinear elements and boundary conditions to traditional linear FE simulation models.

However, this change from linear to nonlinear conditions would substantially increase the computational time and resources required to perform an analysis.

Therefore, when large structures are involved, such as in the case of vibration analysis of the vehicle body structure, the use of nonlinear models becomes an unfeasible alternative, especially when considering the massive introduction of electric vehicles on the market.

In this case, the higher frequency ranges reached by the electric engines when compared against the traditional internal combustion units become an important challenge for the calculation process, which will require much higher computational capacity to analyze the simulation models even under linear conditions, and especially if any non-linear behavior is intended to be considered.

Also, as a consequence of having most of the new vehicles (cars, buses and trucks) equipped with electric engines, the noise level into the vehicles became much lower when compared against the traditional internal combustion engine vehicles.

So, despite of the beneficial effects carried by this, it also significantly increase the vibration and noise perception of the vehicle occupants, turning small squeaks, rattles and resonances, previously imperceptible for drivers and passengers, into very annoying vibration and noises that could not be accepted at any project.

In this work, a new methodology to increase the accuracy of linear models without compromising their complexity and feasibility is proposed. For this, the MAC coefficients, the FRF curves and the FRAC coefficients obtained for both the reference and the adjusted FE models of the vehicle body structure are compared, getting the following findings:

- Both the reference and the adjusted models have a consistent correspondence with respect to the bench test data ($MAC > 0.9$). The adjusted model shows slightly better MAC correspondence than the reference model;

- The adjusted FE model FRF curves fit the bench test results better than the reference model FRF curves over the entire frequency range studied. This correlation is certified by the analysis of the modal shapes of the most relevant FRF peaks and therefore confirmed by their statistical analysis.
- The overall average FRAC coefficient of the adjusted model ($\text{FRAC}_{\text{adj}} = 0.4751$) shows an improvement of 9.1% compared to the overall average FRAC coefficient of the reference model ($\text{FRAC}_{\text{ref}} = 0.4355$).

In addition, the optimization analysis based on the results of the adjusted FE model confirms a significant improvement in the efficiency of the vehicle body structure weight.

The analysis of these results confirms the effectiveness of the proposed methodology. It fills the knowledge gap, initially identified, through a novel process for adjusting the stiffness and damping of spot-welded joints on thin-wall structures using small FE test models and including them into linear FE models of the entire vehicle body structure.

However, considering the results obtained, the following research opportunities to expand the correlation improvements achieved here can be further exploited:

- Improve the accuracy of the mesh representation of welding spots by proposing new element formulations;
- Improve the accuracy of the mesh representation of bolted connections by proposing new element formulations;
- Improve the accuracy of the mesh representation of adhesive connection joints by proposing new element formulations;
- Refine the representation of the FE models by incorporating variations in sheet metal thickness from stamping manufacturing processes;
- Improve the accuracy of experimental bench test results by proposing new acquisition strategies for raw data measurements; and
- Improve the accuracy of experimental bench test results by proposing new data processing methods for both vibration and damping evaluation.

Some of these results are also discussed and presented in the article “Enhancing the Accuracy of Linear Finite Element Models of Vehicle Structures Considering Spot-Welded Flanges” shown in the Publication section and published in 11th of October of 2021 by MPDI Materials Q1 Journal.

These considerations, among others, can be introduced in future research models applied to vehicle structures for their refinement, either for vibration and noise analysis or even for durability and fatigue studies, aiming to increase the accuracy of their simulations and consequently improve the correlation level between the analytical and test results.

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APPENDIX I

Script used for MAC calculation:

```
%File data input
[filename,pathname] = uigetfile({'*.xlsm; *.xlsx; *.xls','Excel files'}, 'Please choose modal file');
inputfile = strcat(pathname, '\', filename);
cd(pathname);

%Pre-allocation of variables
MAC_x = [];
MAC_y = [];
MAC_z = [];
MAC_xmag = [];
MAC_ymag = [];
MAC_zmag = [];
MAC_x_headers = {};
MAC_y_headers = {};
MAC_z_headers = {};

%Variables list assignment
xsheet_test = sheetsfile(xfindposition(1),1);
ysheet_test = sheetsfile(yfindposition(1),1);
zsheet_test = sheetsfile(zfindposition(1),1);
xdata_test = xdata_test(:,1:col(1)-1);
ydata_test = ydata_test(:,1:col(1)-1);
zdata_test = zdata_test(:,1:col(1)-1);

%Conversion of mag and phase of test data into imaginary number
xcomplex_test= xdata_test(:,colmag).*exp(xdata_test(:,colphase)*1i);
ycomplex_test= ydata_test(:,colmag).*exp(ydata_test(:,colphase)*1i);
zcomplex_test= zdata_test(:,colmag).*exp(zdata_test(:,colphase)*1i);

%Conversion of mag and phase of fem data into imaginary number
xcomplex_fem= xdata_fem(:,colmag).*exp(xdata_fem(:,colphase)*1i);
ycomplex_fem= ydata_fem(:,colmag).*exp(ydata_fem(:,colphase)*1i);
zcomplex_fem= zdata_fem(:,colmag).*exp(zdata_fem(:,colphase)*1i);

%MAC using operator
for j = 1:size(xcomplex_fem,2)
    MAC_x_temp = (cumsum((xcomplex_fem(:,j))*xcomplex_test(:,j)))/
        ((cumsum(xcomplex_fem(:,j))^2)* (cumsum(xcomplex_test(:,j))^2));
```

```

MAC_y_temp = (cumsum((ycomplex_fem(:,j))*ycomplex_test(:,j)))/
((cumsum(ycomplex_fem(:,j))^2)* (cumsum(ycomplex_test(:,j))^2));
MAC_z_temp = (cumsum((zcomplex_fem(:,j))*zcomplex_test(:,j)))/
((cumsum(zcomplex_fem(:,j))^2)* (cumsum(zcomplex_test(:,j))^2));
MAC_xmag = [MAC_xmag ;
sqrt((real(MAC_x_temp)^2)+(imag(MAC_x_temp)^2))];
MAC_ymag = [MAC_ymag ;
sqrt((real(MAC_y_temp)^2)+(imag(MAC_y_temp)^2))];
MAC_zmag = [MAC_zmag ;
sqrt((real(MAC_z_temp)^2)+(imag(MAC_z_temp)^2))];
%Creation of identification tag based on sheet, test column and fem column
MAC_x_header_temp =
[xsheet_test '_' xsheet_fem '_' xheaders{colmag(j)}(1:end-2)];
MAC_y_header_temp =
[ysheet_test '_' ysheet_fem '_' yheaders{colmag(j)}(1:end-2)];
MAC_z_header_temp =
[zsheets_test '_' zsheet_fem '_' zheaders{colmag(j)}(1:end-2)];
MAC_x = [MAC_x ; MAC_x_temp];
MAC_y = [MAC_y ; MAC_y_temp];
MAC_z = [MAC_z ; MAC_z_temp];
MAC_x_headers = [MAC_x_headers {MAC_x_header_temp}];
MAC_y_headers = [MAC_y_headers {MAC_y_header_temp}];
MAC_z_headers = [MAC_z_headers {MAC_z_header_temp}];
end
%Conversion of mag and phase of test and fem data into imaginary number
complex_test= data_test(:,colmag).*exp(data_test(:,colphase)*1i);
complex_fem= data_fem(:,colmag).*exp(data_fem(:,colphase)*1i);
%MAC coefficient calculation
for j = 1:size(complex_fem,2)
MAC_temp = (cumsum((complex_fem(:,j))*complex_test(:,j))) /
((cumsum(complex_fem(:,j))^2) * (cumsum(complex_test(:,j))^2));
MAC_mag = [MAC_mag ; sqrt((real(MAC_temp)^2)+(imag(MAC_temp)^2))];
MAC = [MAC ; MAC_temp];
end

```

APPENDIX II

Script used for FRAC calculation:

```
%File data input
[filename,pathname] = uigetfile({'*.xslm; *.xlsx; *.xls','Excel
files'},'Please choose curve file');
inputfile = strcat(pathname,'\ ',filename);
cd(pathname);

%Pre-allocation of variables
FRAC_x = [];
FRAC_y = [];
FRAC_z = [];
FRAC_xmag = [];
FRAC_ymag = [];
FRAC_zmag = [];
FRAC_x_headers = {};
FRAC_y_headers = {};
FRAC_z_headers = {};

%Variables list asignment
xsheet_test = sheetsfile{xfindposition(1),1};
ysheet_test = sheetsfile{yfindposition(1),1};
zsheet_test = sheetsfile{zfindposition(1),1};
xdata_test = xdata_test(:,1:col(1)-1);
ydata_test = ydata_test(:,1:col(1)-1);
zdata_test = zdata_test(:,1:col(1)-1);

%Conversion of mag and phase of test data into imaginary number
xcomplex_test= xdata_test(:,colmag).*exp(xdata_test(:,colphase)*1i);
ycomplex_test= ydata_test(:,colmag).*exp(ydata_test(:,colphase)*1i);
zcomplex_test= zdata_test(:,colmag).*exp(zdata_test(:,colphase)*1i);

%Conversion of mag and phase of fem data into imaginary number
xcomplex_fem= xdata_fem(:,colmag).*exp(xdata_fem(:,colphase)*1i);
ycomplex_fem= ydata_fem(:,colmag).*exp(ydata_fem(:,colphase)*1i);
zcomplex_fem= zdata_fem(:,colmag).*exp(zdata_fem(:,colphase)*1i);

%FRAC using operator
for j = 1:size(xcomplex_fem,2)
    FRAC_x_temp= ((cumsum((xcomplex_fem(:,j))')*xcomplex_test(:,j)))^2) /
    (cumsum((xcomplex_fem(:,j))')*xcomplex_fem(:,j)) *
    (cumsum((xcomplex_test(:,j))')*xcomplex_test(:,j))));
    FRAC_y_temp= ((cumsum((ycomplex_fem(:,j))')*ycomplex_test(:,j)))^2) /
    (cumsum((ycomplex_fem(:,j))')*ycomplex_fem(:,j)) *
```

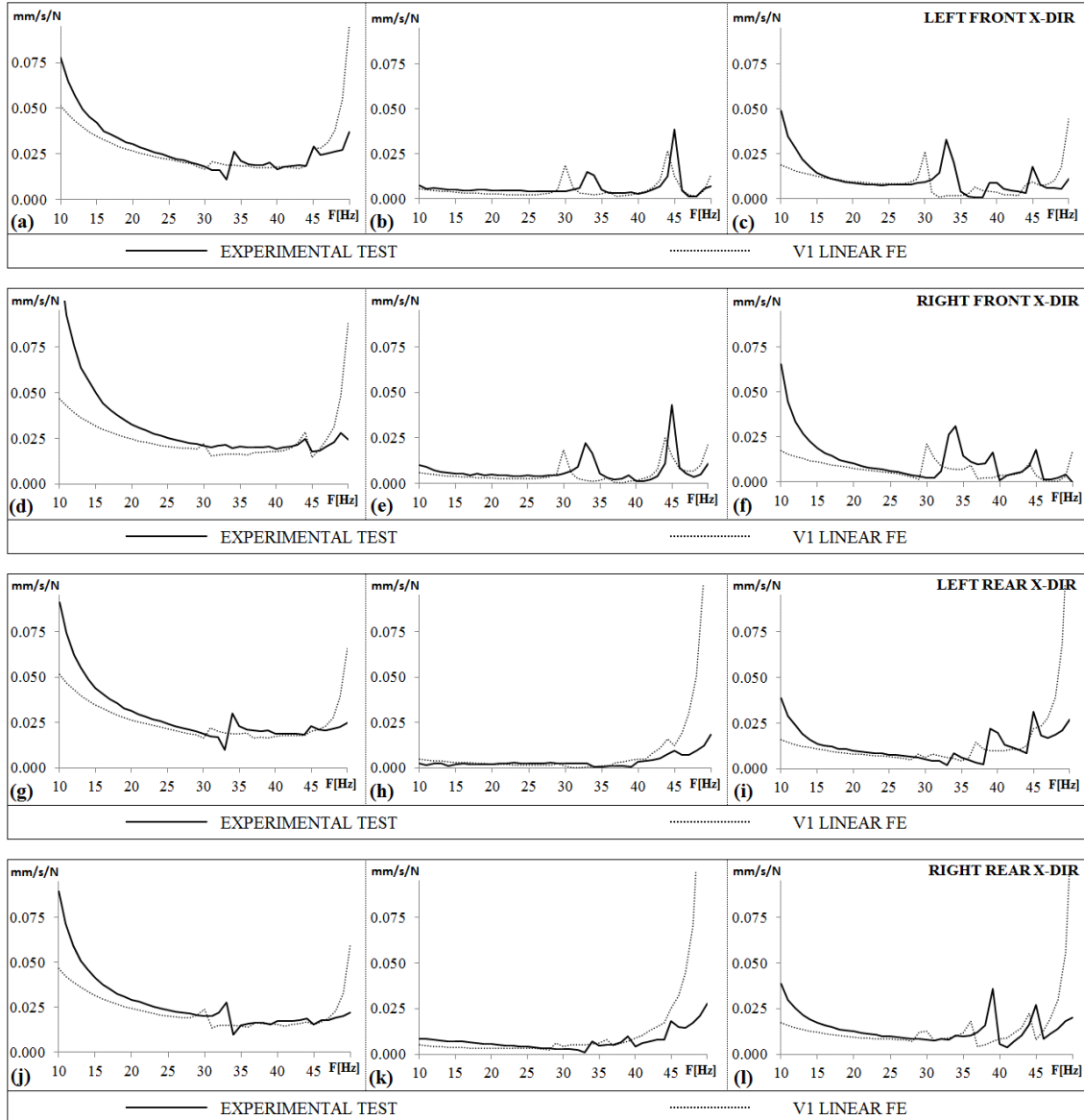
```

(cumsum(((ycomplex_test(:,j))')*ycomplex_test(:,j))));
FRAC_z_temp= ((cumsum(((zcomplex_fem(:,j))')*zcomplex_test(:,j)))^2) /
(cumsum(((zcomplex_fem(:,j))')*zcomplex_fem(:,j)) *
(cumsum(((zcomplex_test(:,j))')*zcomplex_test(:,j))));
FRAC_xmag = [FRAC_xmag ;
sqrt((real(FRAC_x_temp)^2)+(imag(FRAC_x_temp)^2))];
FRAC_ymag = [FRAC_ymag ;
sqrt((real(FRAC_y_temp)^2)+(imag(FRAC_y_temp)^2))];
FRAC_zmag = [FRAC_zmag ;
sqrt((real(FRAC_z_temp)^2)+(imag(FRAC_z_temp)^2))];
%Creation of identification tag based on sheet, test column and fem column
FRAC_x_header_temp =
[xsheet_test '_' xsheet_fem '_' xheaders{colmag(j)}(1:end-2)];
FRAC_y_header_temp =
[ysheet_test '_' ysheet_fem '_' yheaders{colmag(j)}(1:end-2)];
FRAC_z_header_temp =
[zsheetsheet_test '_' zsheet_fem '_' zheaders{colmag(j)}(1:end-2)];
FRAC_x = [FRAC_x ; FRAC_x_temp];
FRAC_y = [FRAC_y ; FRAC_y_temp];
FRAC_z = [FRAC_z ; FRAC_z_temp];
FRAC_x_headers = [FRAC_x_headers {FRAC_x_header_temp}];
FRAC_y_headers = [FRAC_y_headers {FRAC_y_header_temp}];
FRAC_z_headers = [FRAC_z_headers {FRAC_z_header_temp}];
end
%Conversion of mag and phase of test and fem data into imaginary number
complex_test= data_test(:,colmag).*exp(data_test(:,colphase)*1i);
complex_fem = data_fem(:,colmag).*exp(data_fem(:,colphase)*1i);
%FRAC coefficient calculation
for j = 1:size(complex_fem,2)
FRAC_temp= ((cumsum(((complex_fem(:,j))')*complex_test(:,j)))^2) /
(cumsum(((complex_fem(:,j))')*complex_fem(:,j)) *
(cumsum(((complex_test(:,j))')*complex_test(:,j))));
FRAC_mag = [FRAC_mag ; sqrt((real(FRAC_temp)^2)+(imag(FRAC_temp)^2))];
FRAC = [FRAC ; FRAC_temp];
end

```

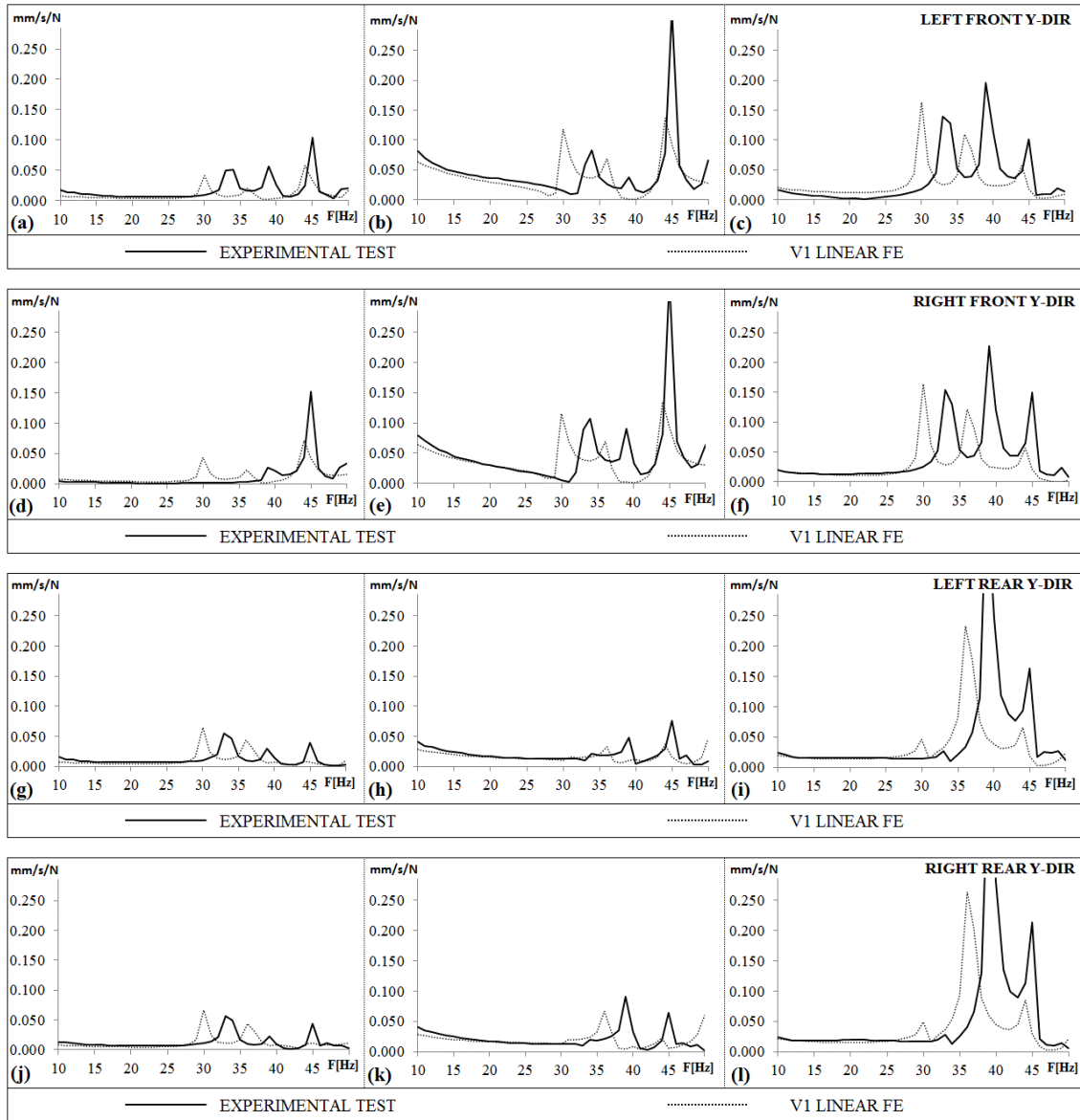
APPENDIX III

Experimental test vs FE model V1 FRF plots:



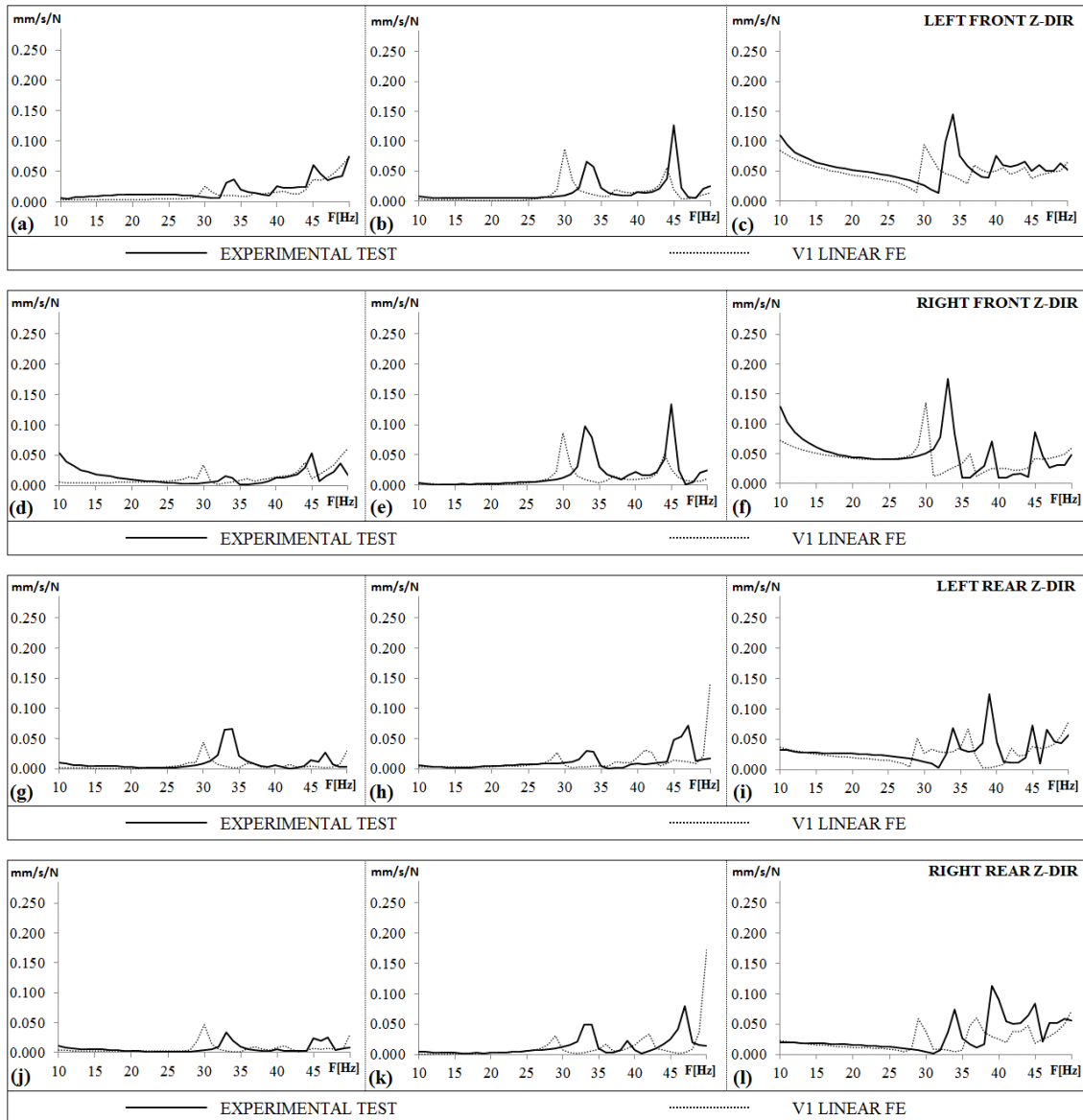
Experimental test *vs* FE linear model V1 FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* FE linear model V1 FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

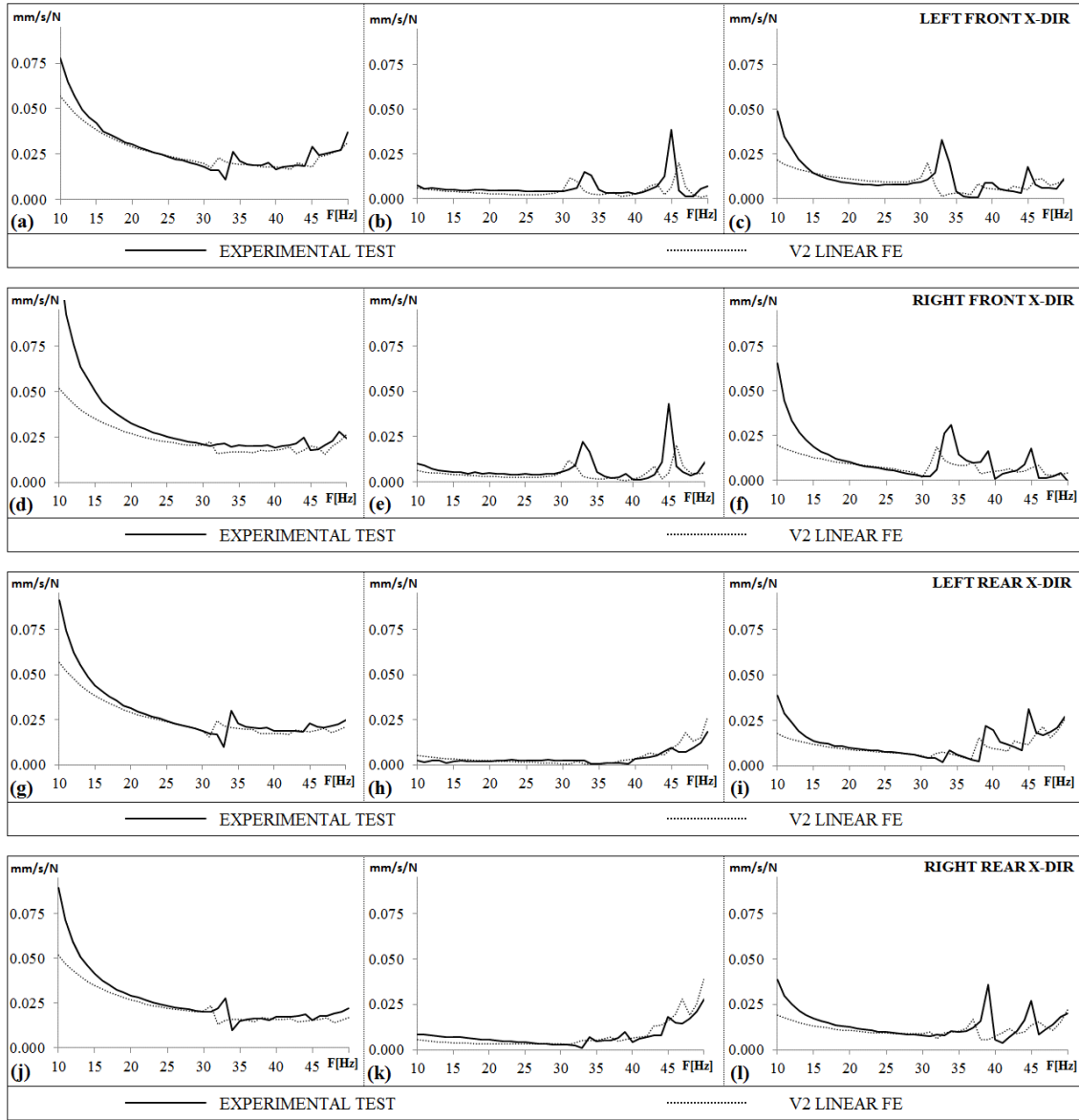


Experimental test *vs* FE linear model V1 FRFs (vibration magnitude z-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

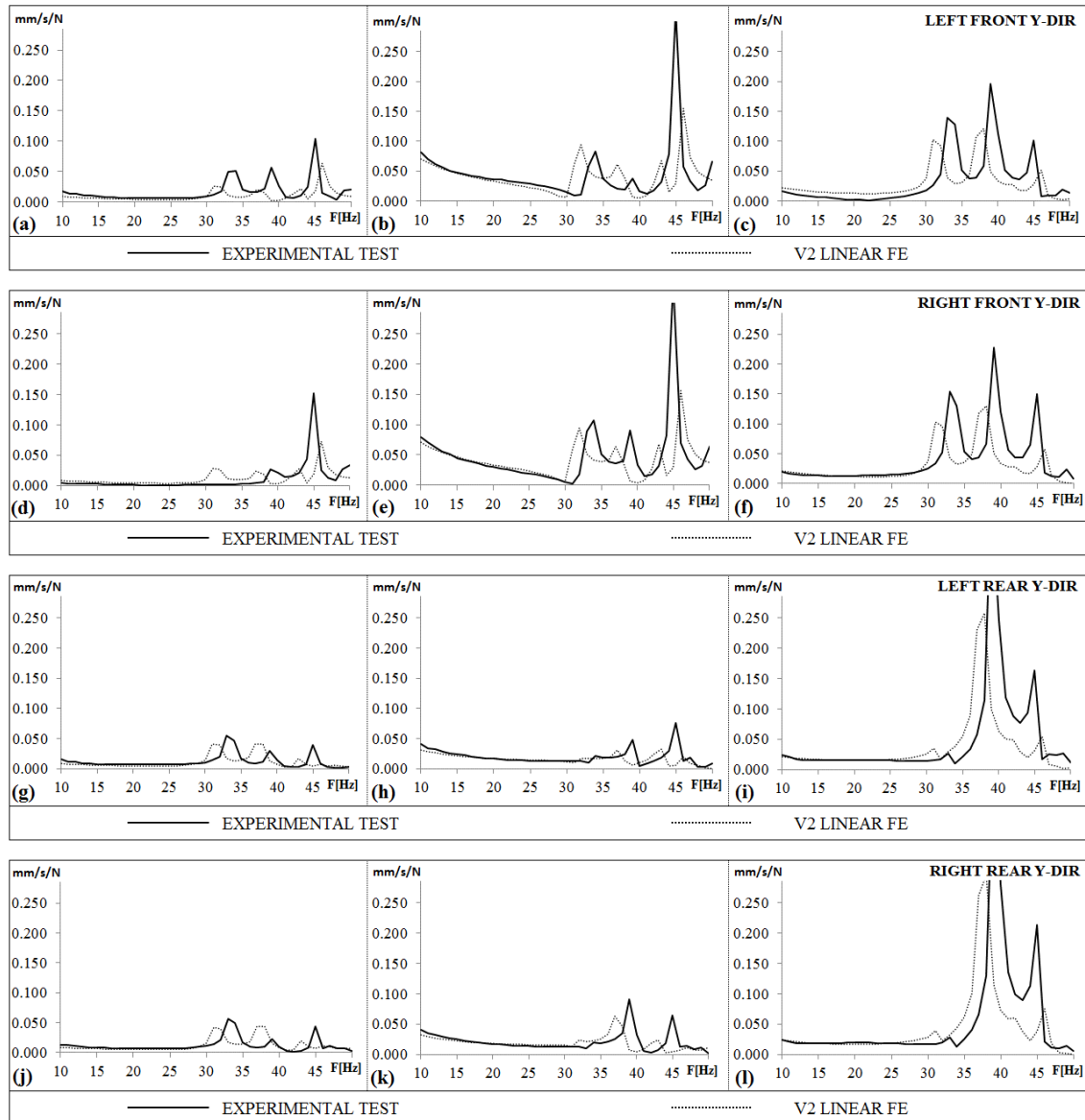
APPENDIX IV

Experimental test vs FE model V2 FRF plots:



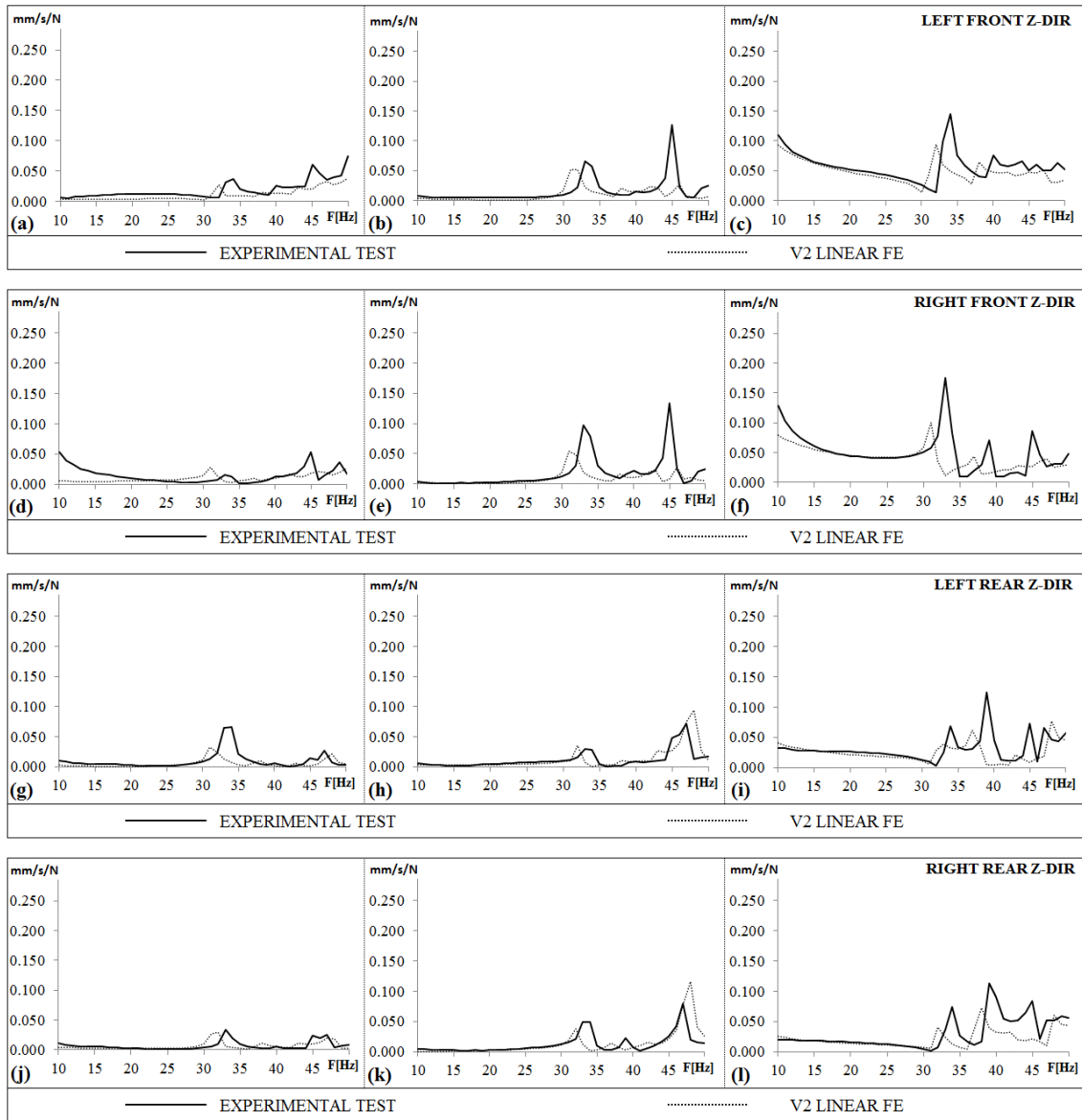
Experimental test vs FE linear model V2 FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* FE linear model V2 FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

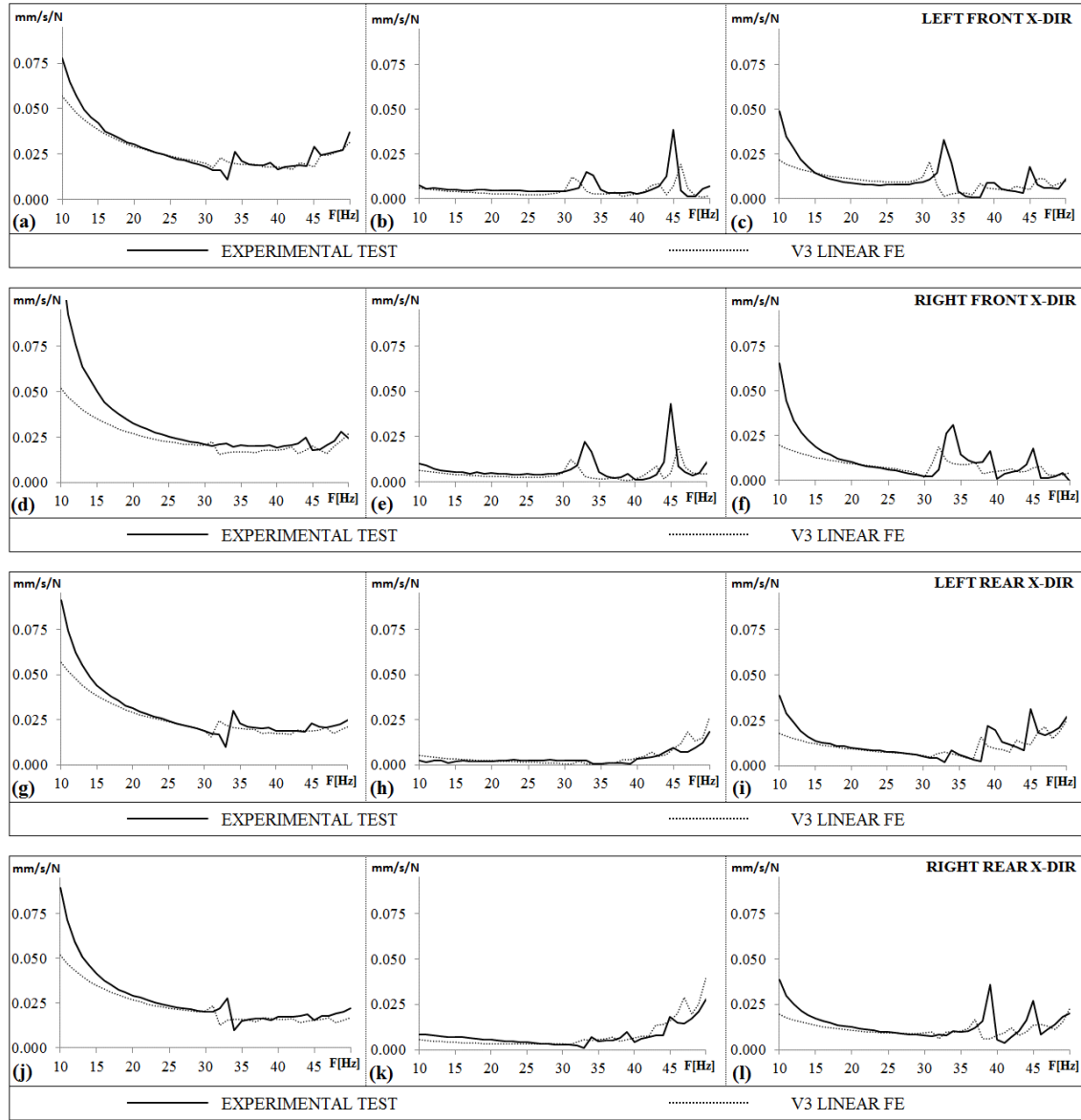


Experimental test *vs* FE linear model V2 FRFs (vibration magnitude z-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

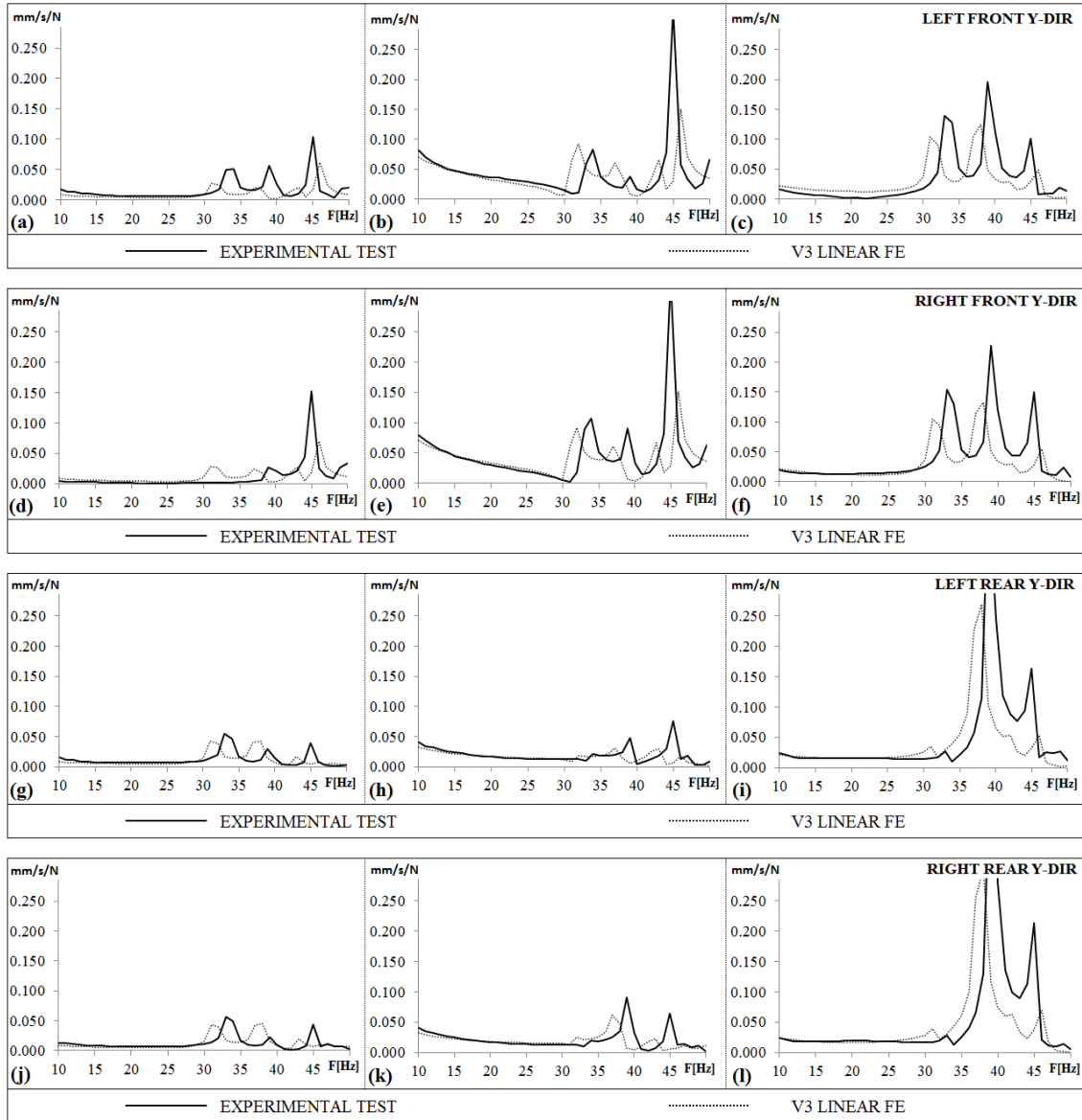
APPENDIX V

Experimental test vs FE model V3 FRF plots:



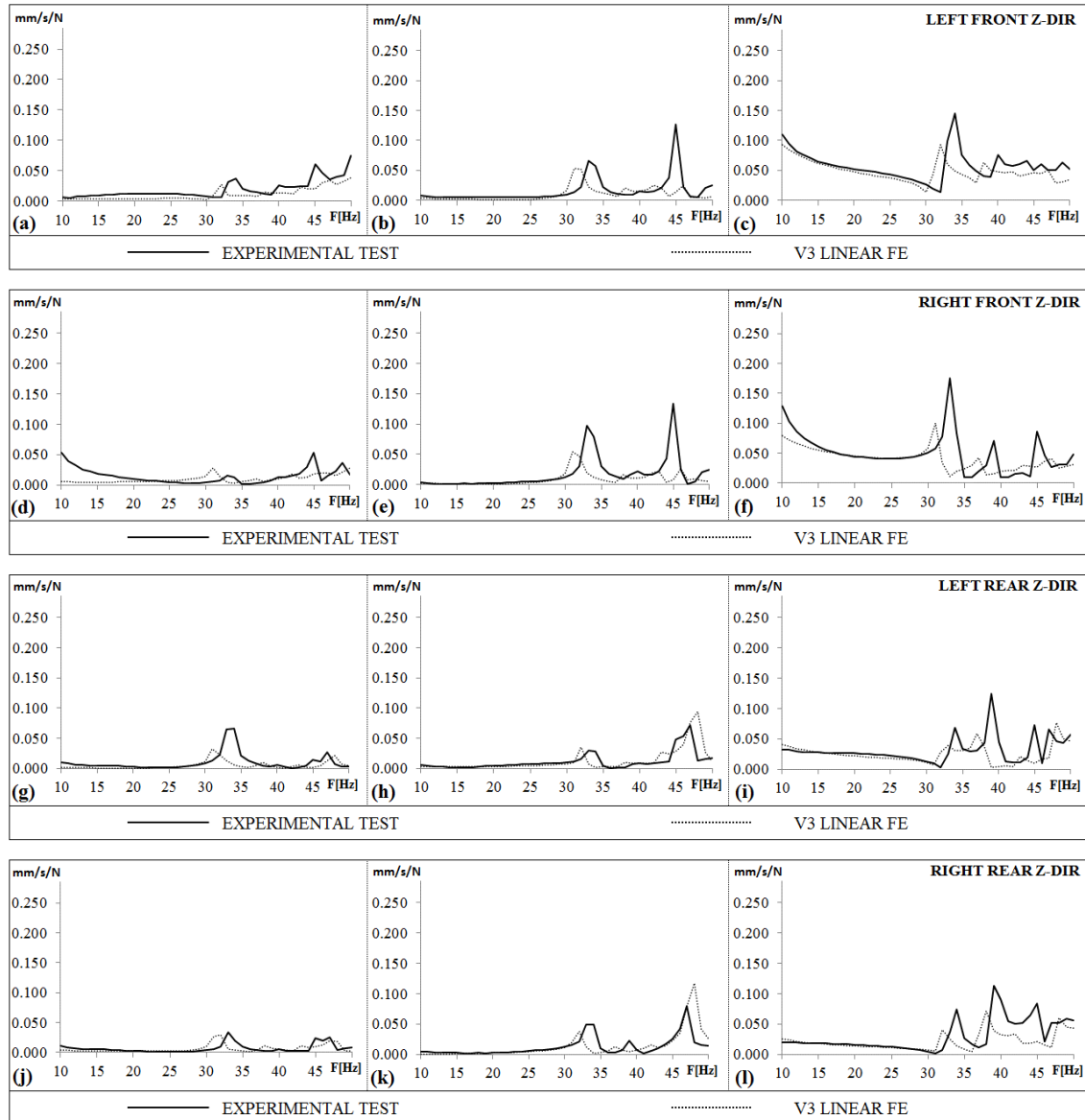
Experimental test vs FE linear model V3 FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* FE linear model V3 FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

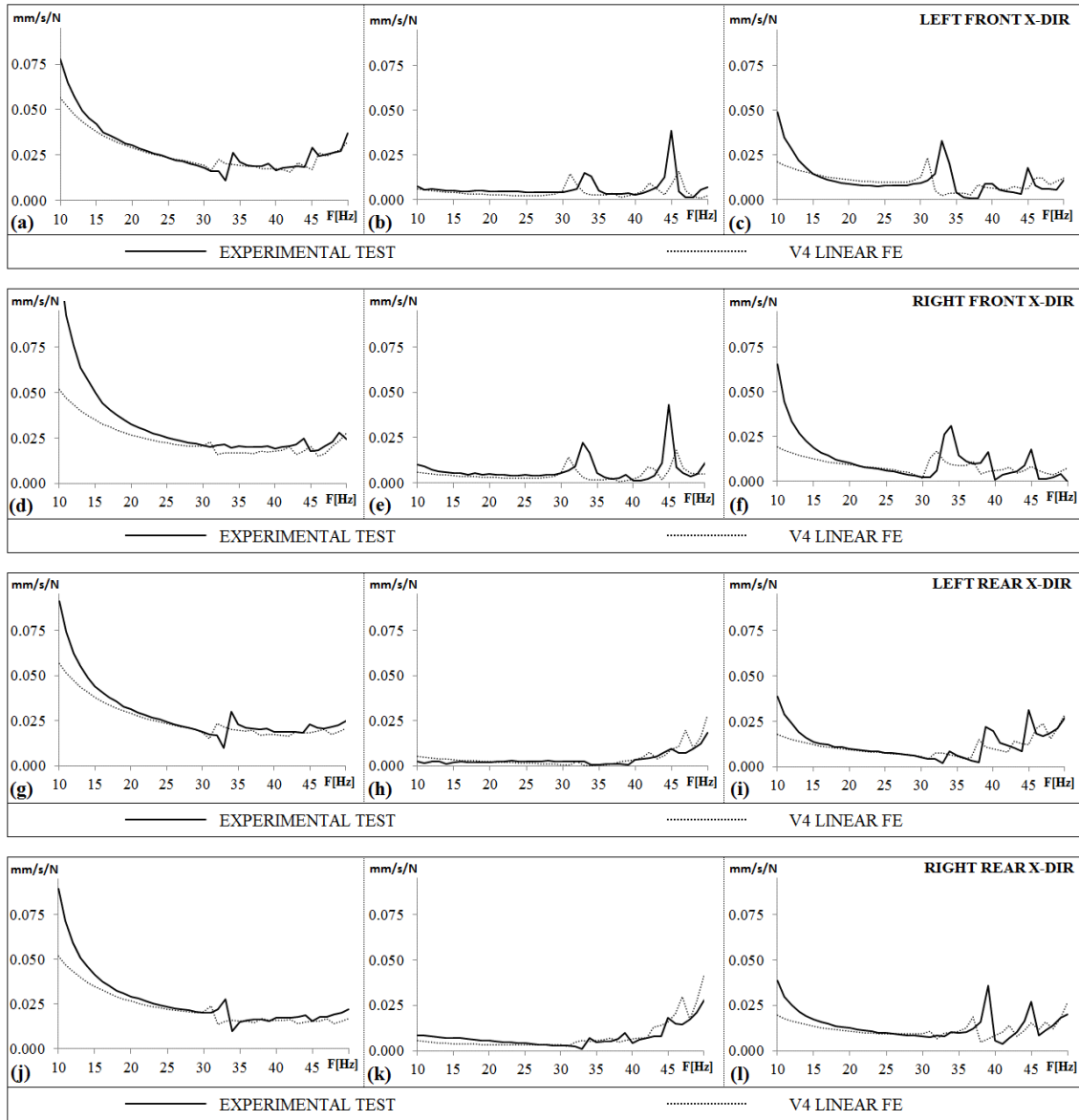


Experimental test *vs* FE linear model V3 FRFs (vibration magnitude z-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

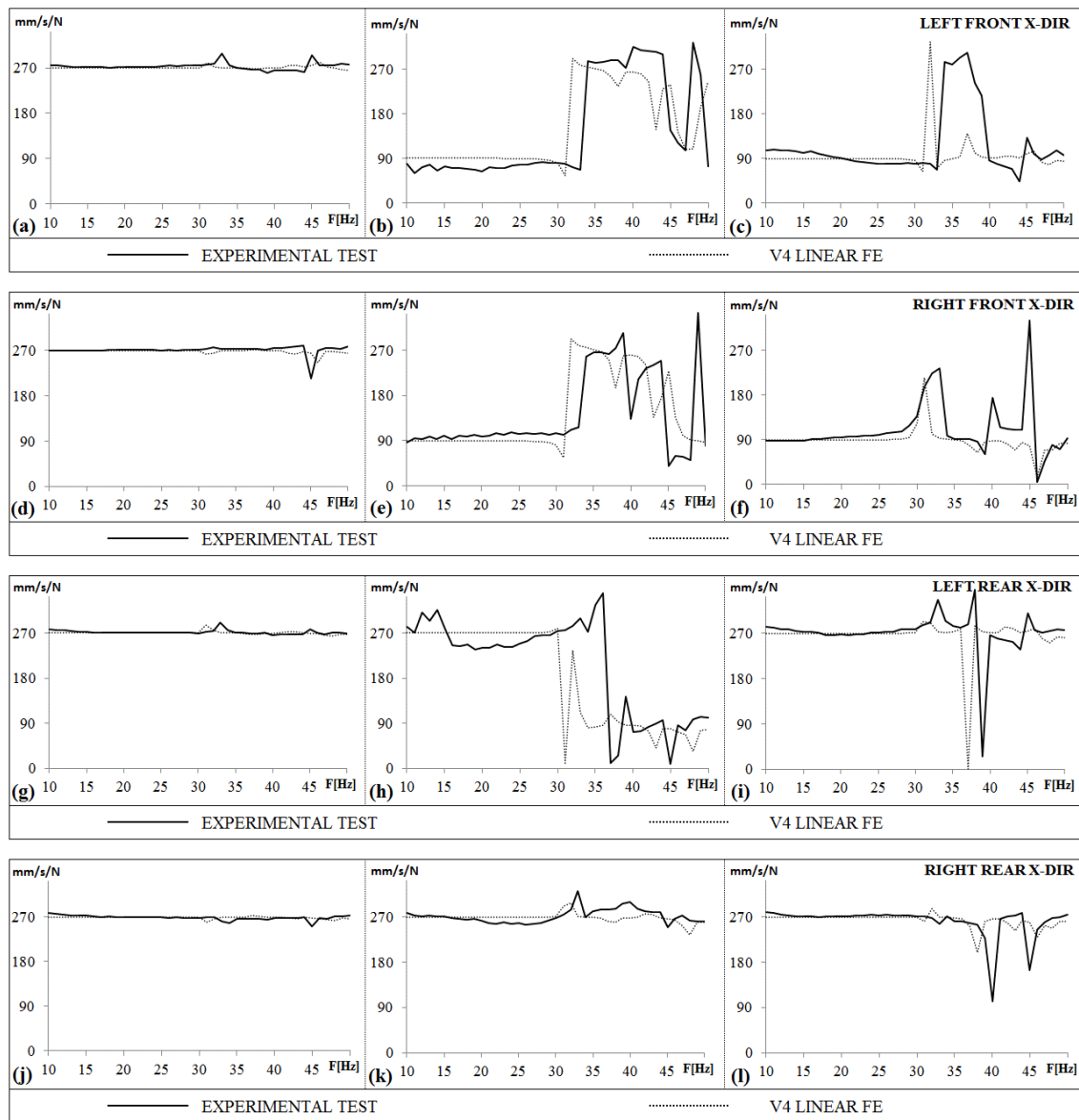
APPENDIX VI

Experimental test vs FE model V4 FRF plots:



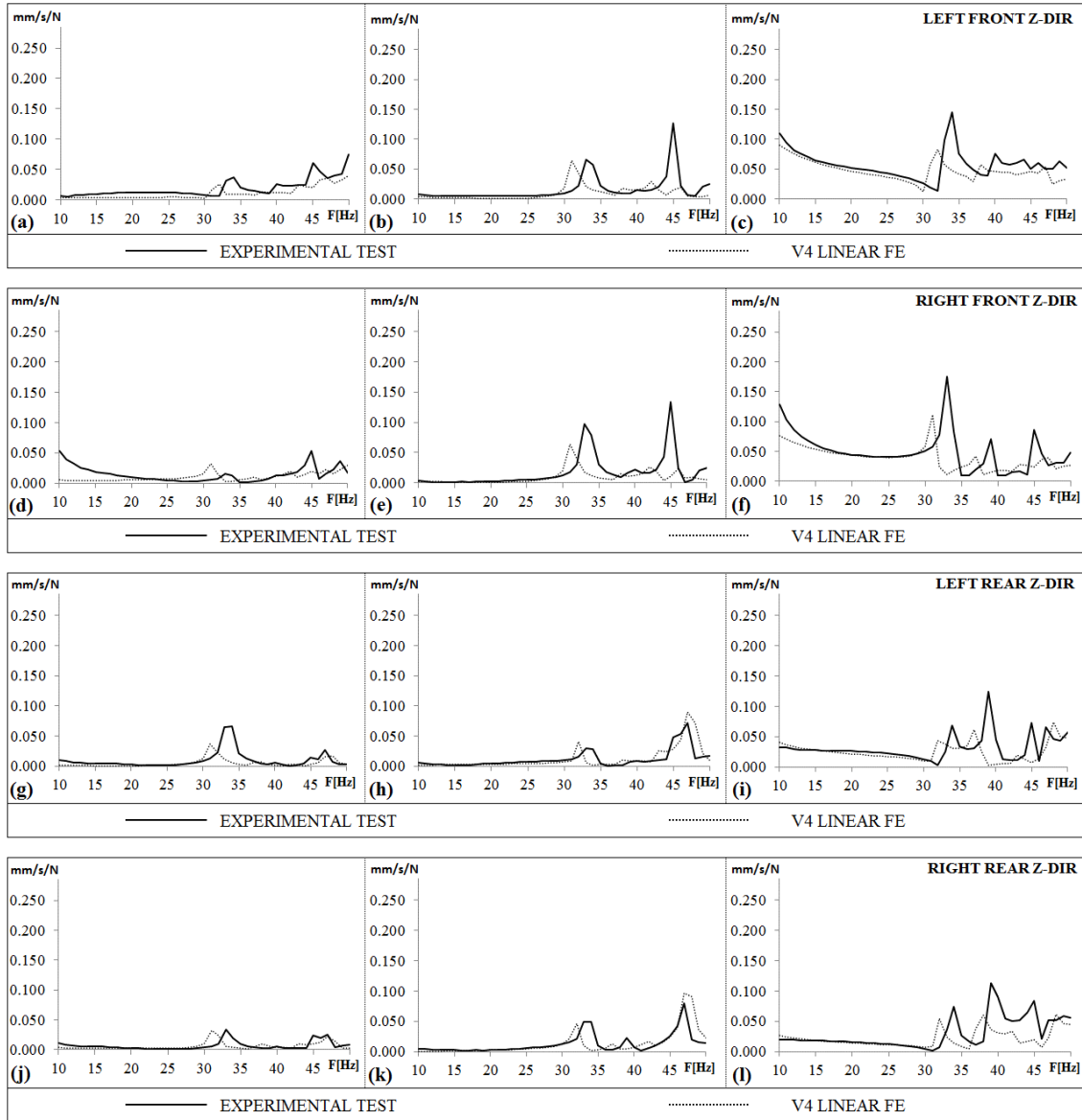
Experimental test *vs* FE linear model V4 FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* FE linear model V4 FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

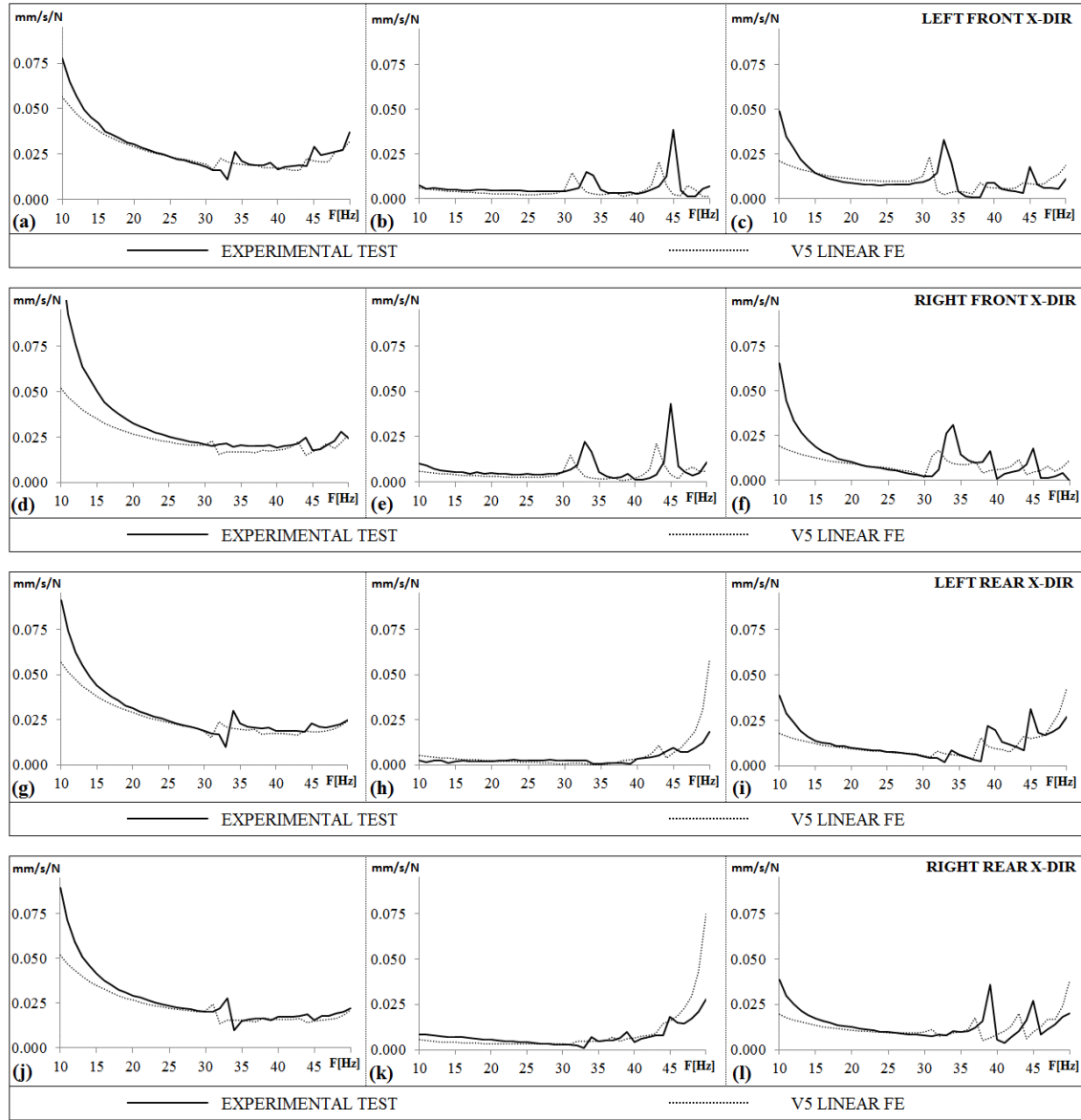


Experimental test *vs* FE linear model V4 FRFs (vibration magnitude z-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

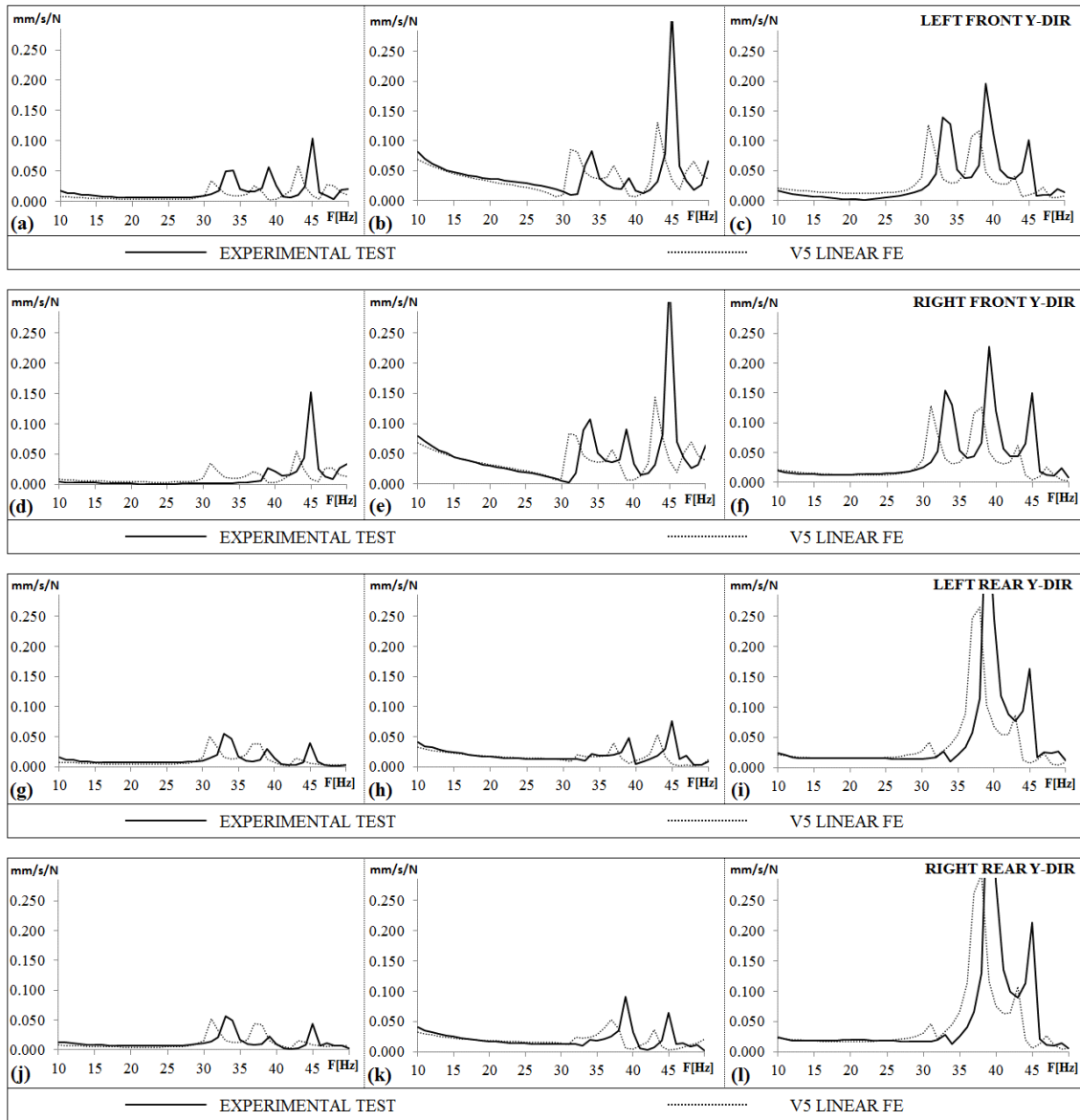
APPENDIX VII

Experimental test vs FE model V5 FRF plots:



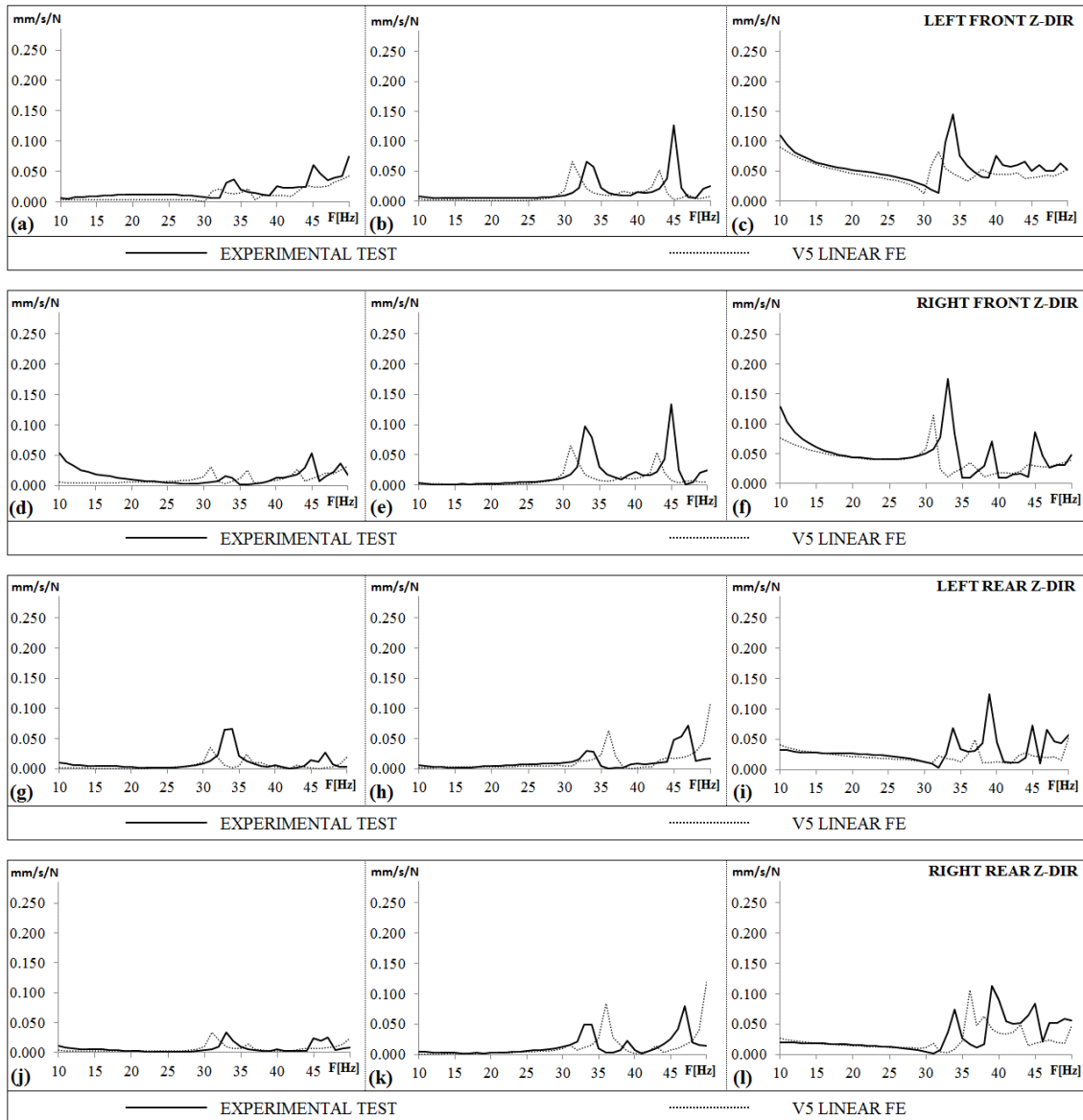
Experimental test vs FE linear model V5 FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* FE linear model V5 FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

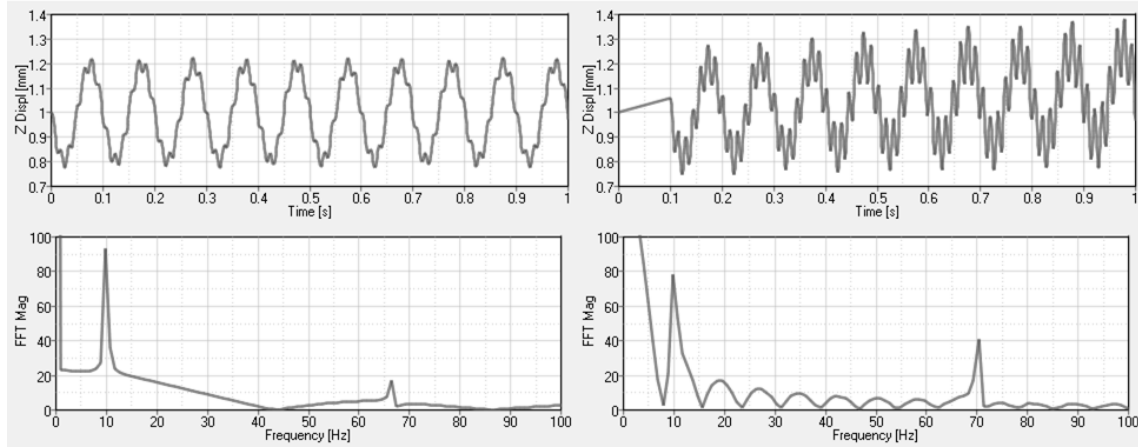


Experimental test *vs* FE linear model V5 FRFs (vibration magnitude z-dir response to unitary excitation) at:

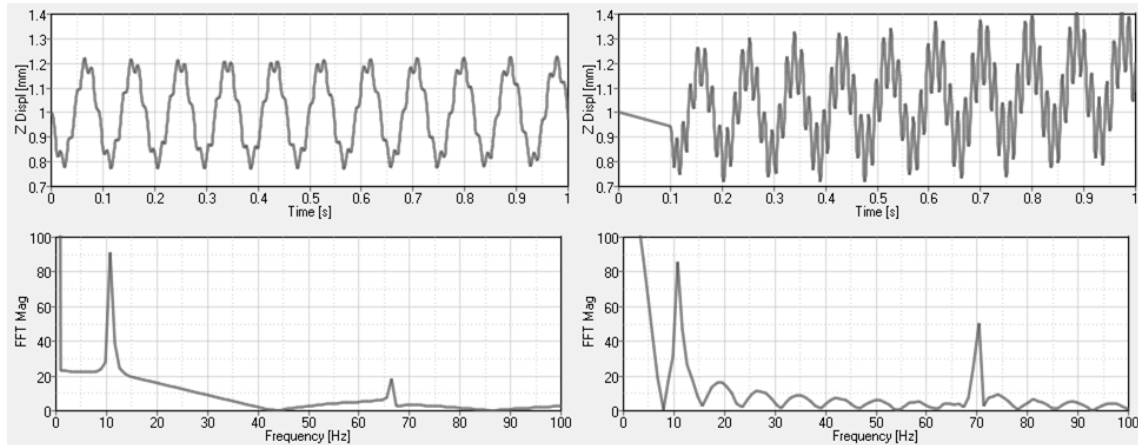
- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

APPENDIX VIII

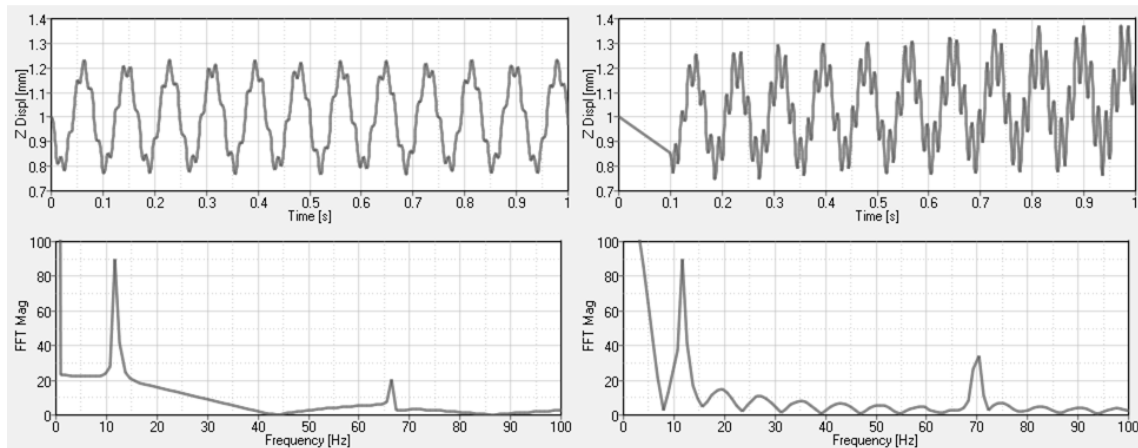
Linear and non-linear FE reduced model time-domain to frequency-domain conversion of FFT response under sinusoidal excitation from 10Hz to 60Hz:



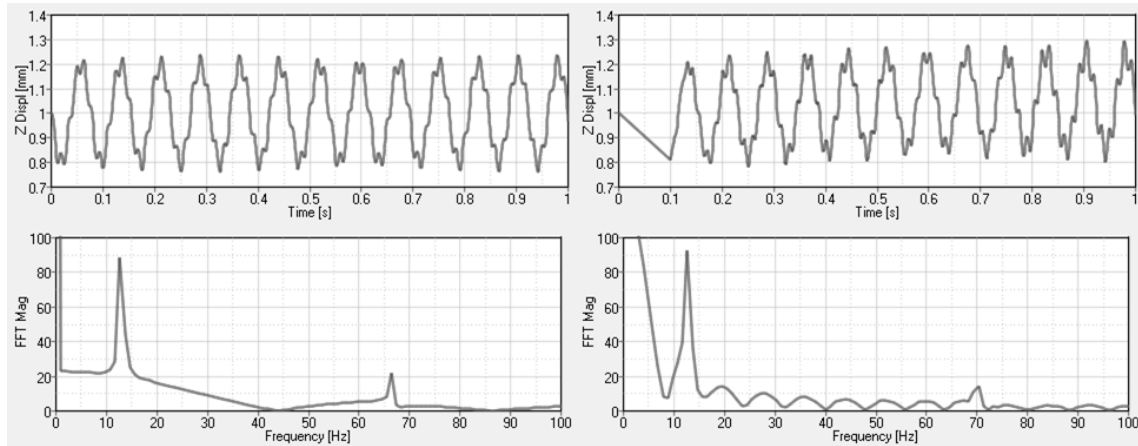
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 10Hz.



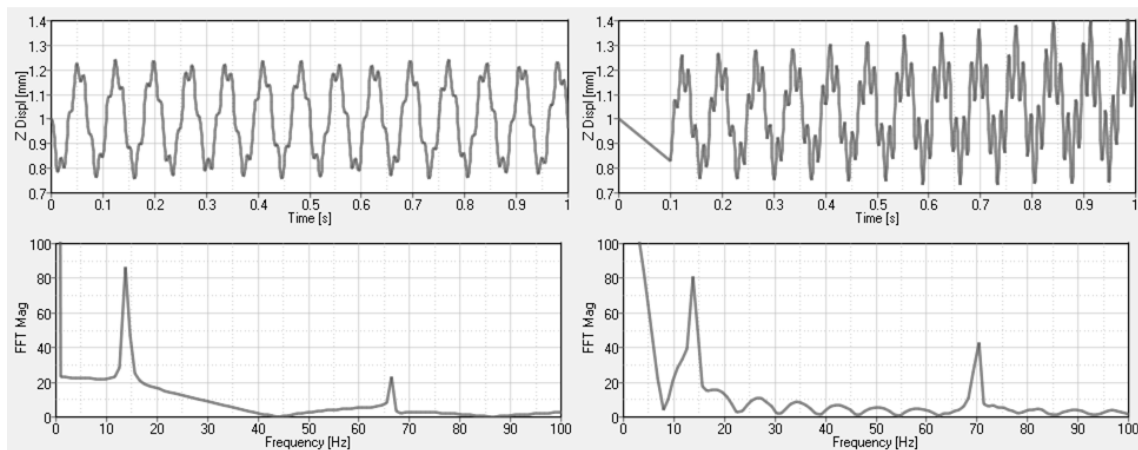
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 11Hz.



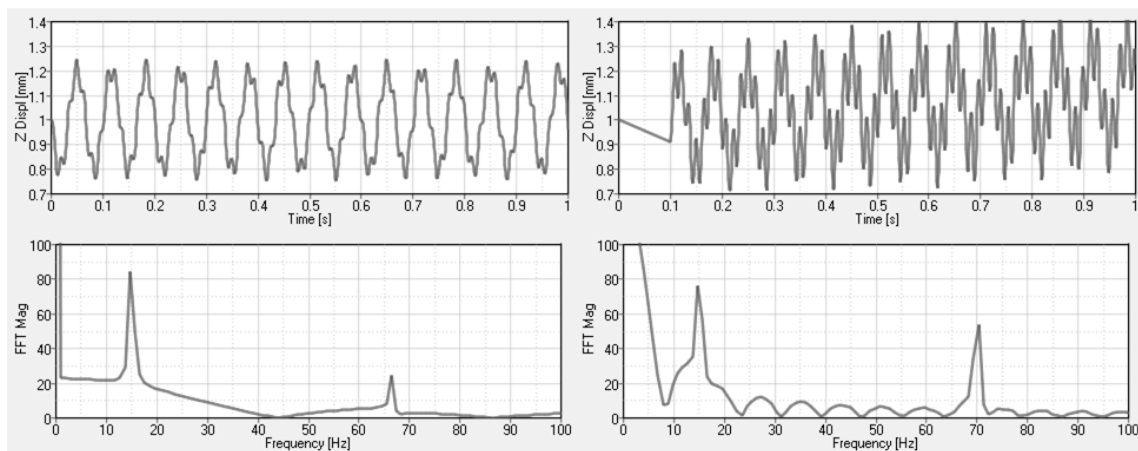
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 12Hz.



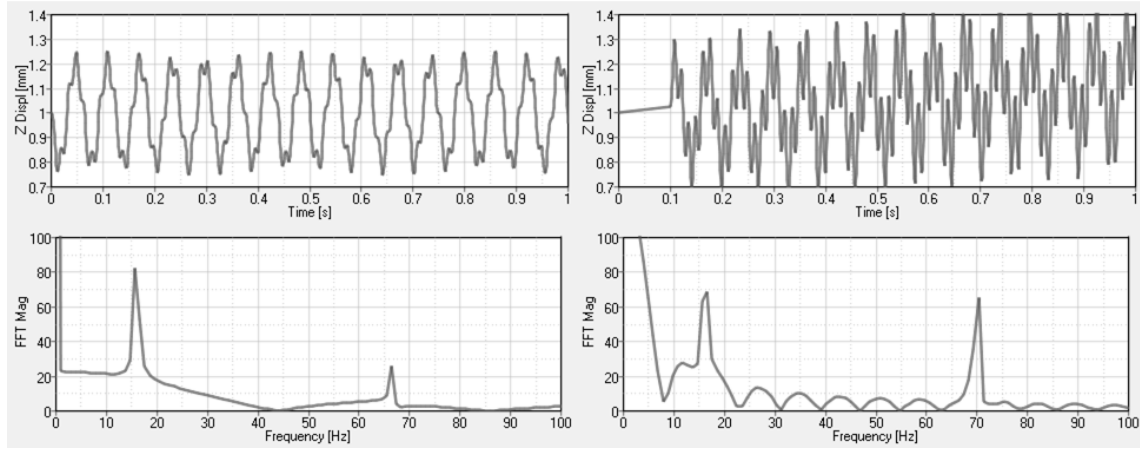
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 13Hz.



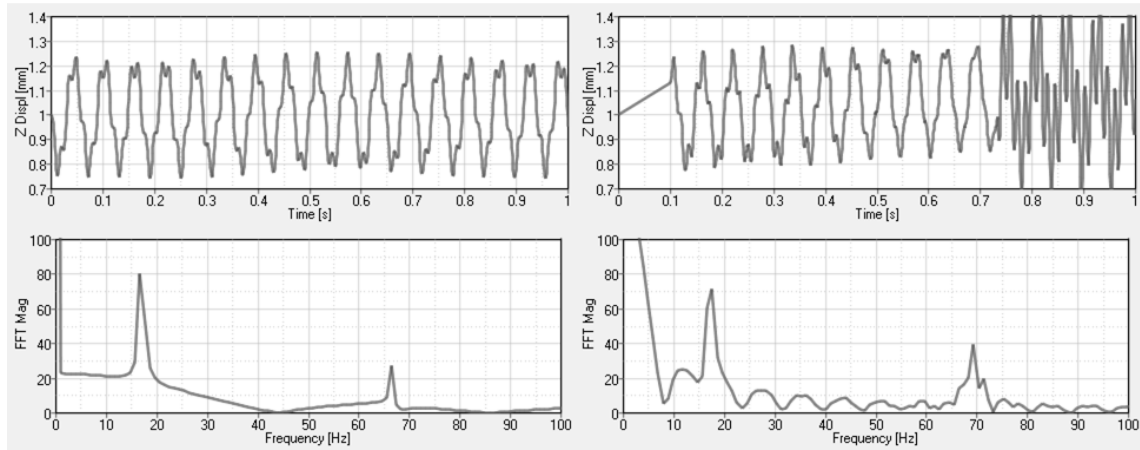
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 14Hz.



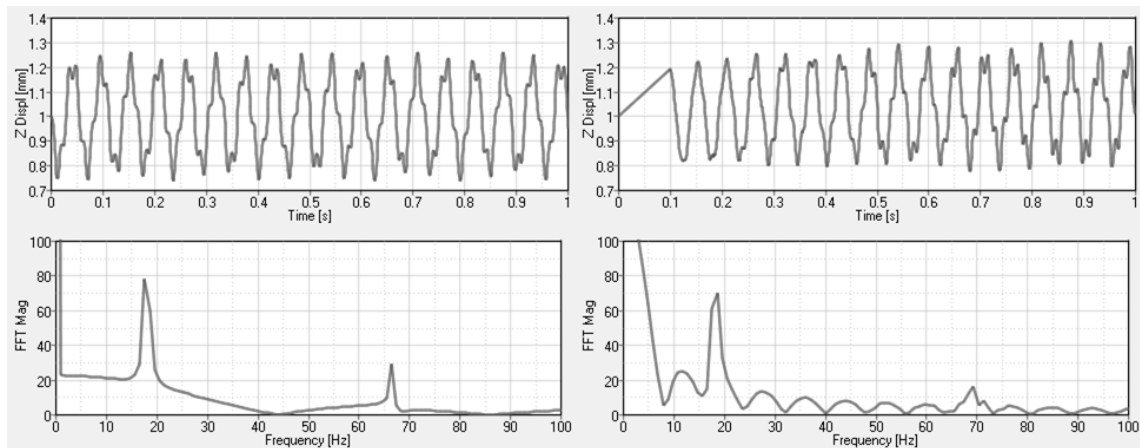
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 15Hz.



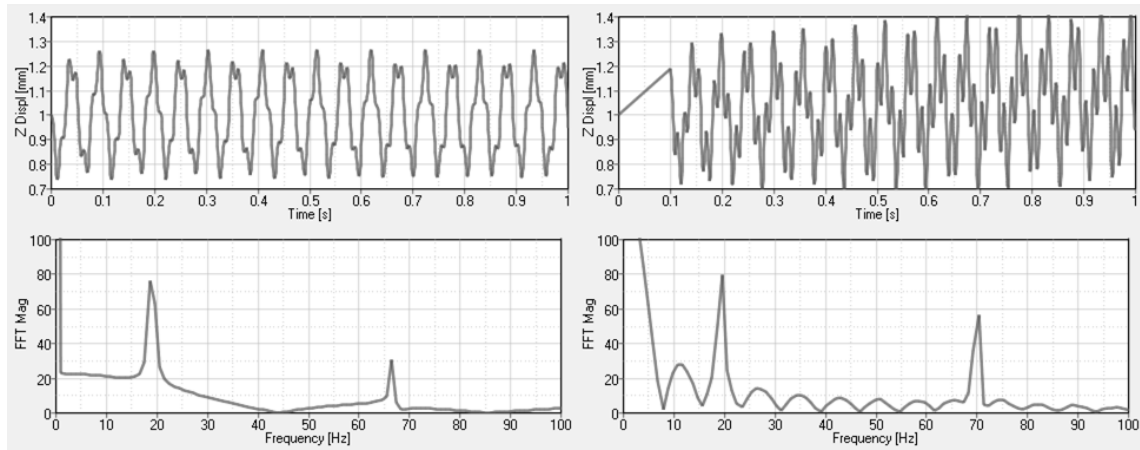
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 16Hz.



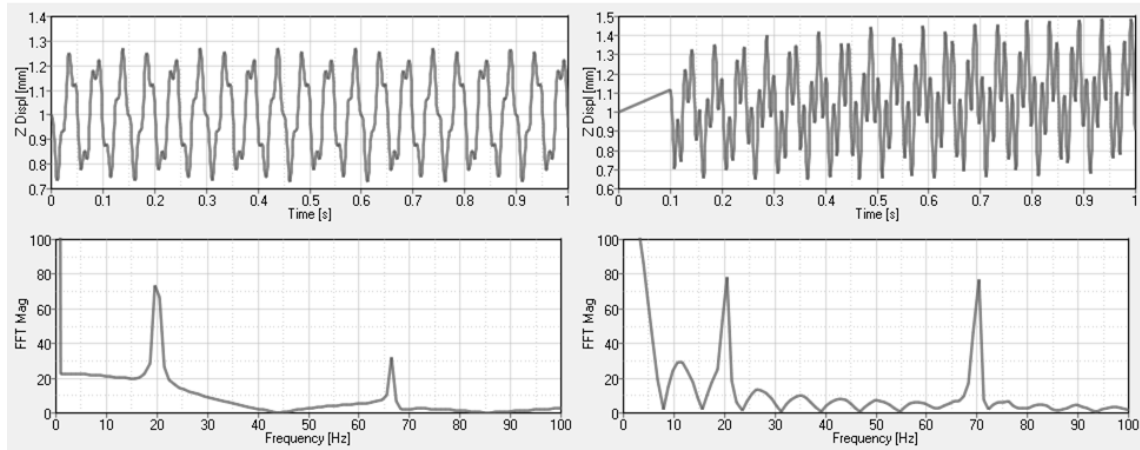
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 17Hz.



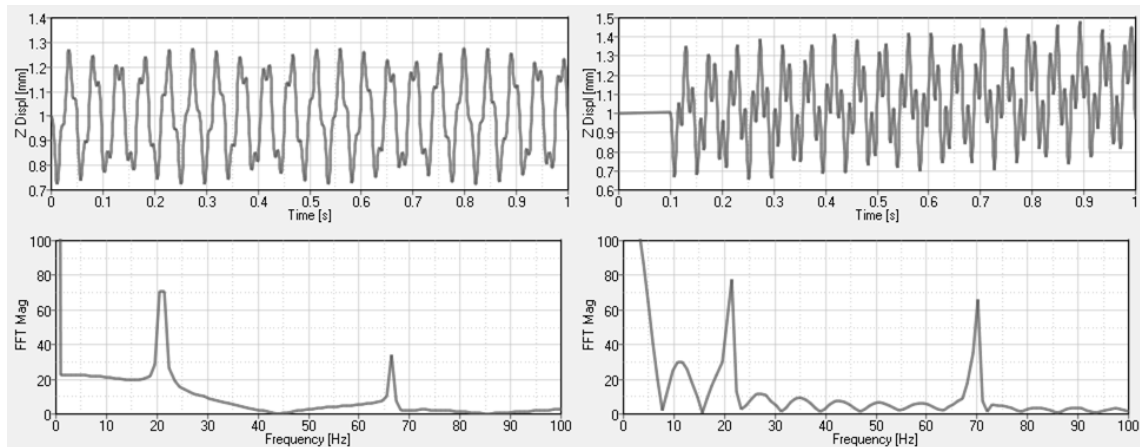
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 18Hz.



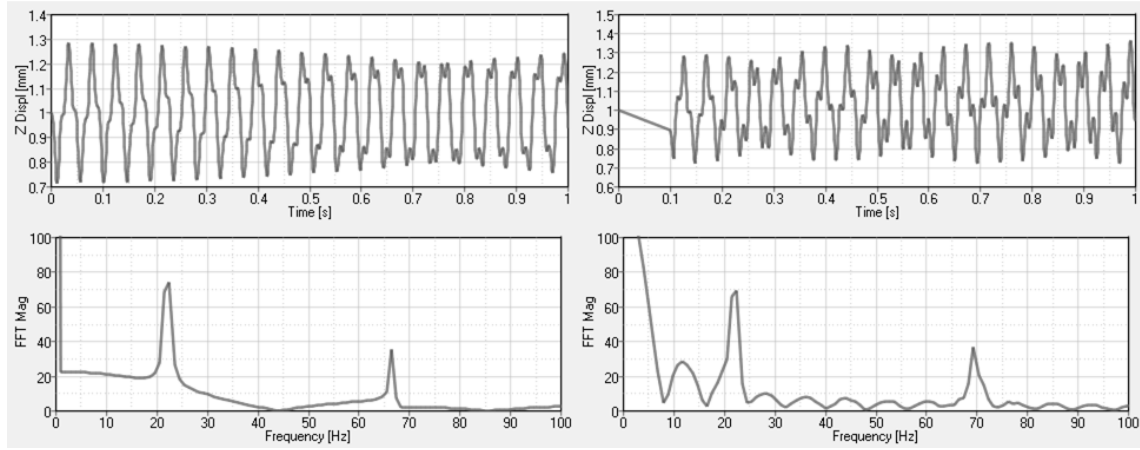
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 19Hz.



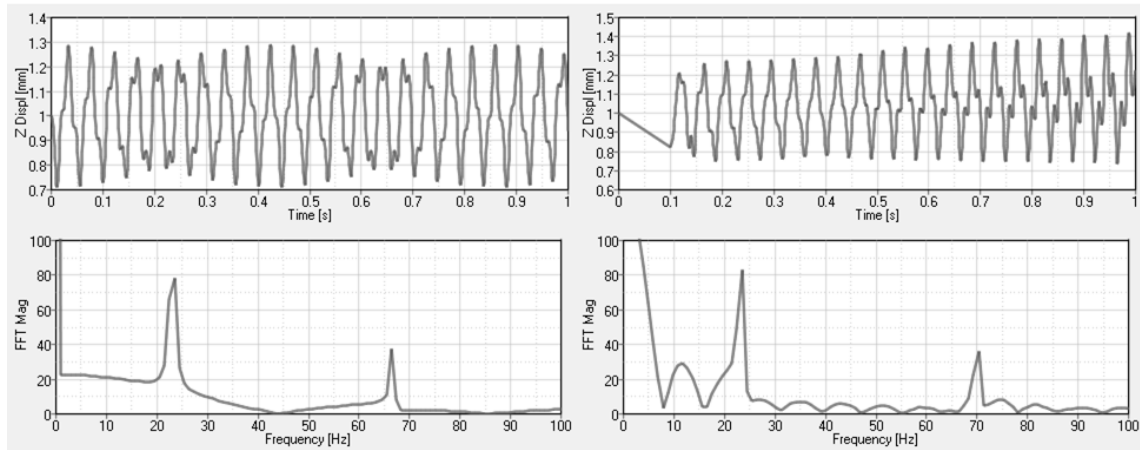
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 20Hz.



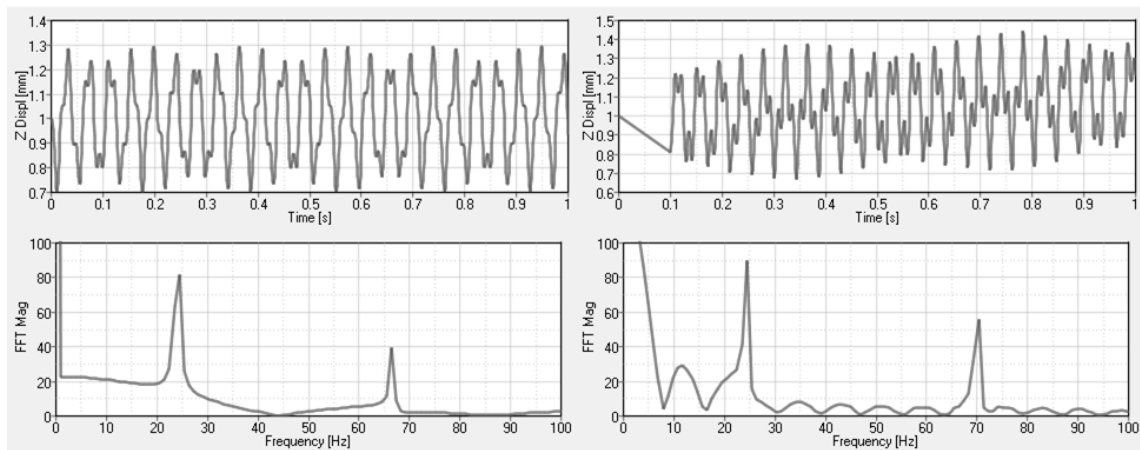
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 21Hz.



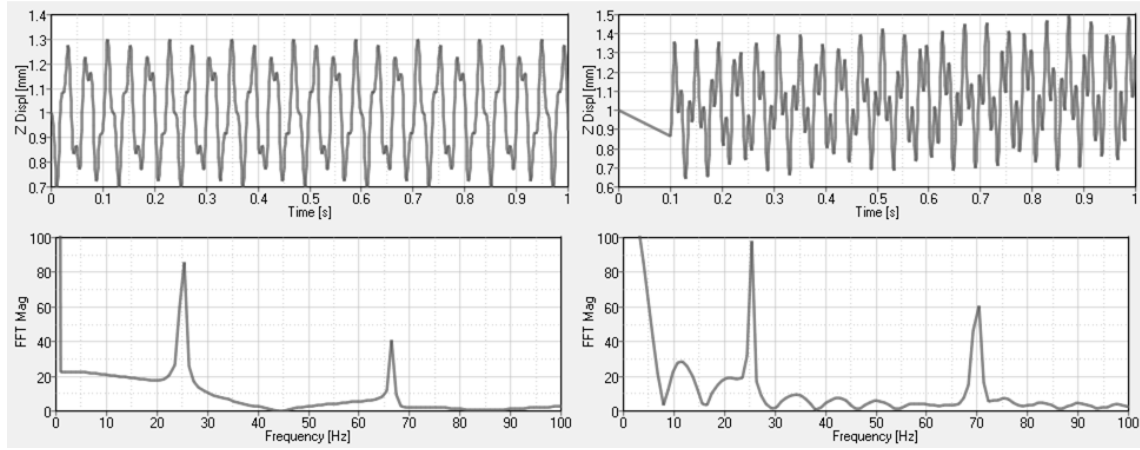
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 22Hz.



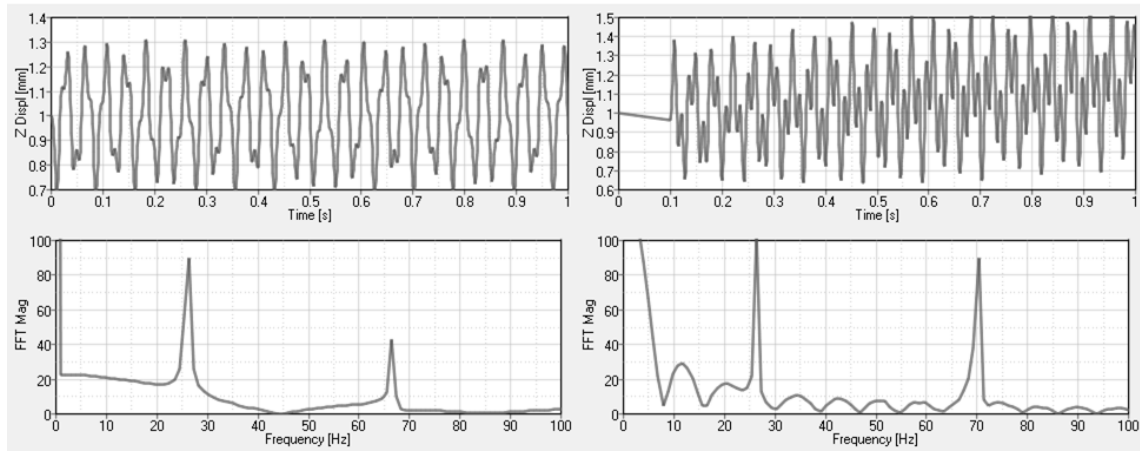
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 23Hz.



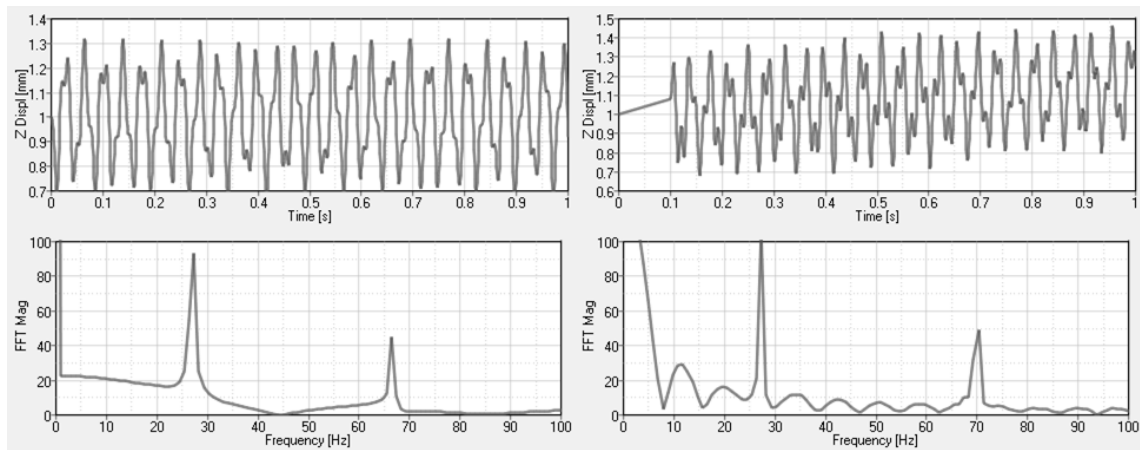
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 24Hz.



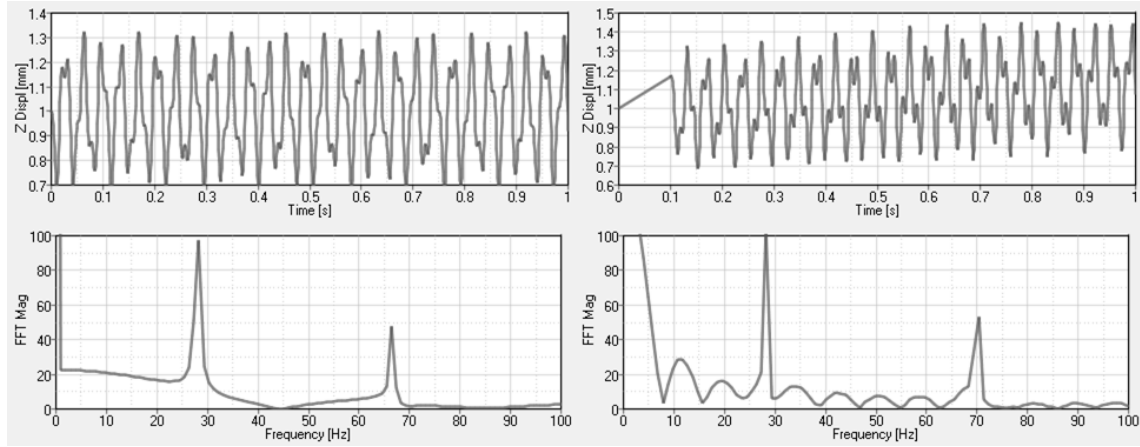
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 25Hz.



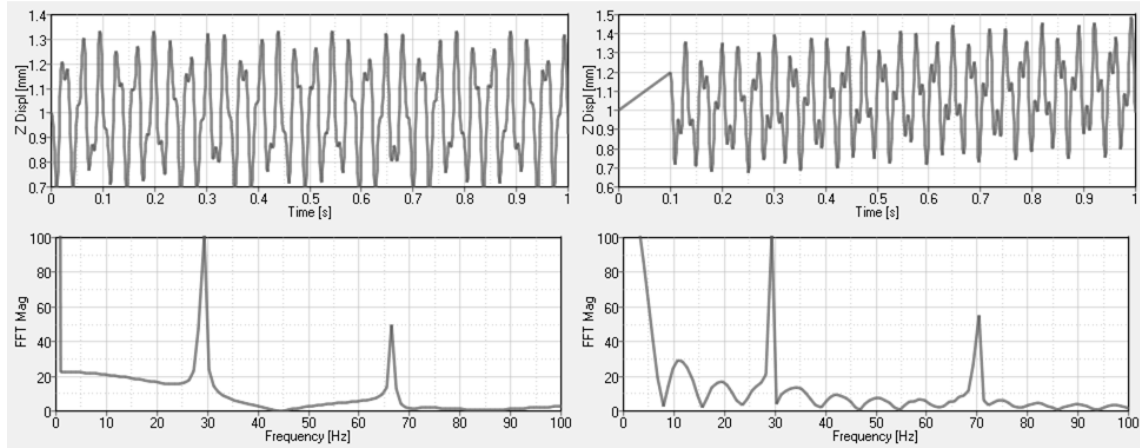
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 26Hz.



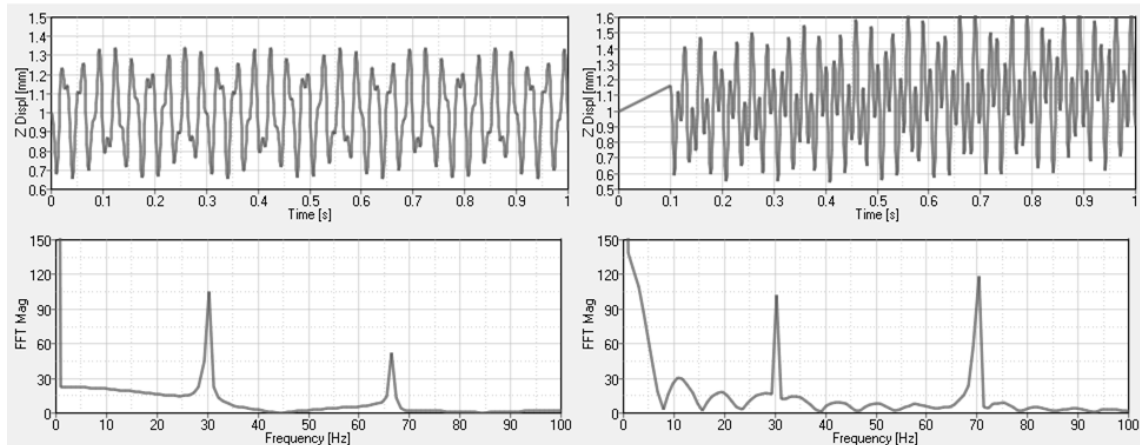
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 27Hz.



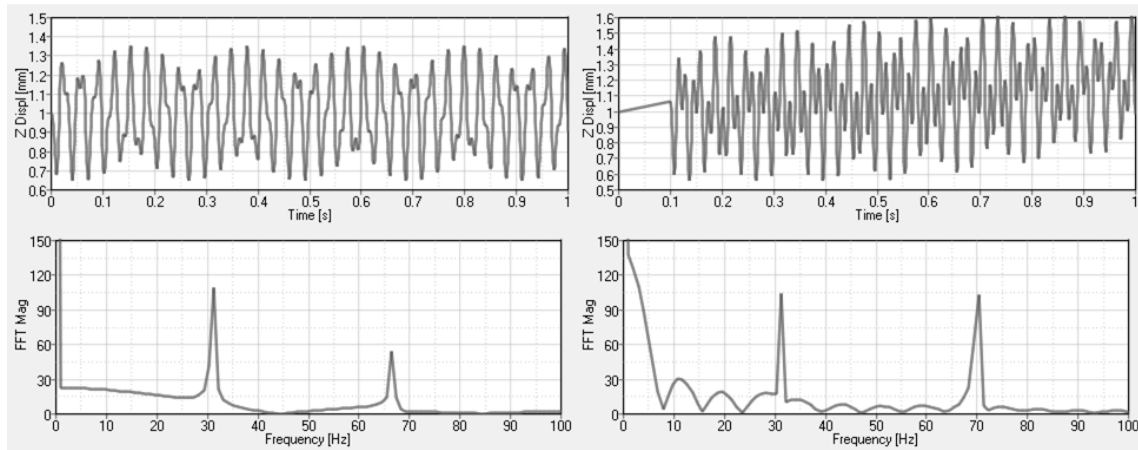
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 28Hz.



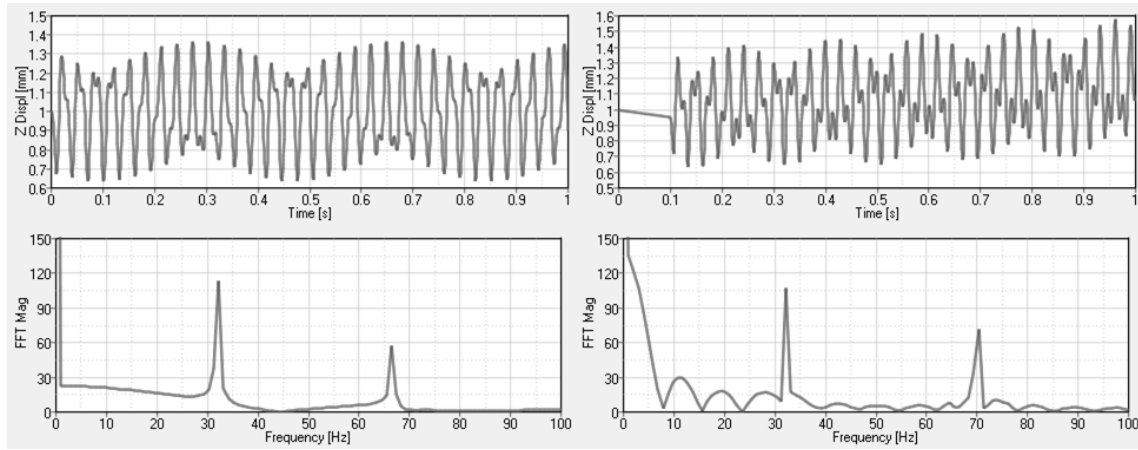
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 29Hz.



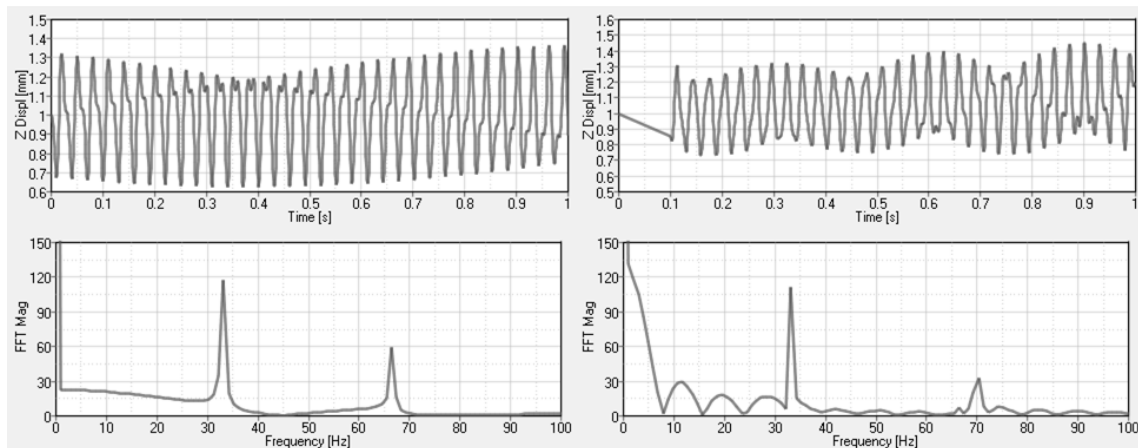
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 30Hz.



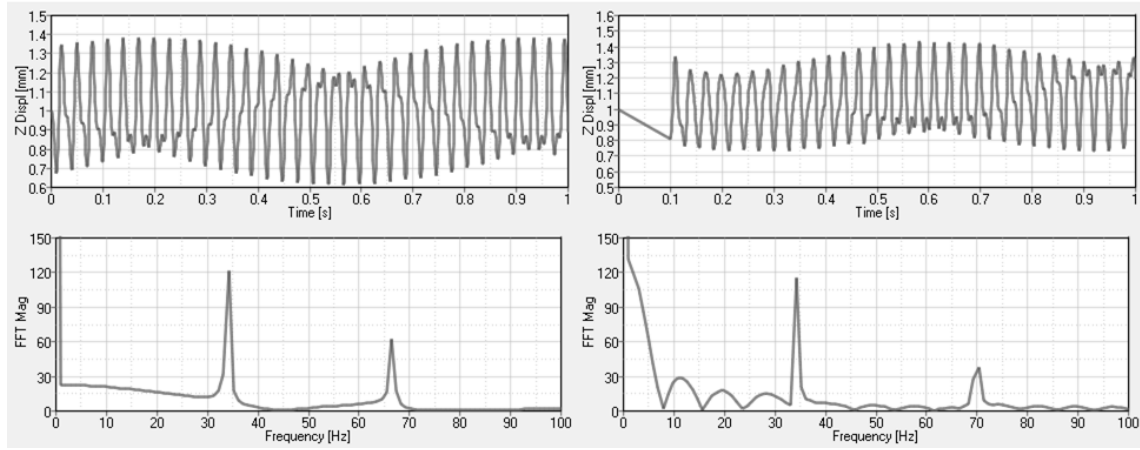
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 31Hz.



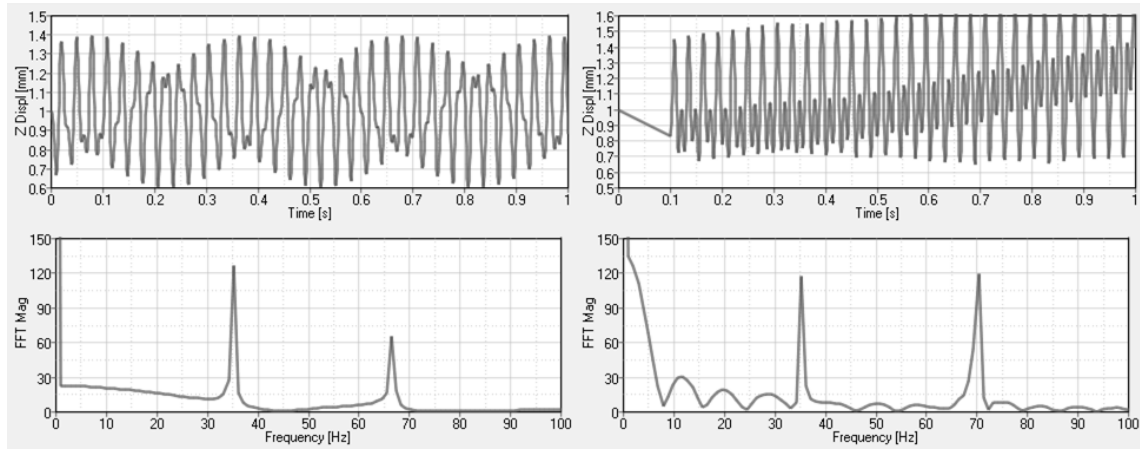
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 32Hz.



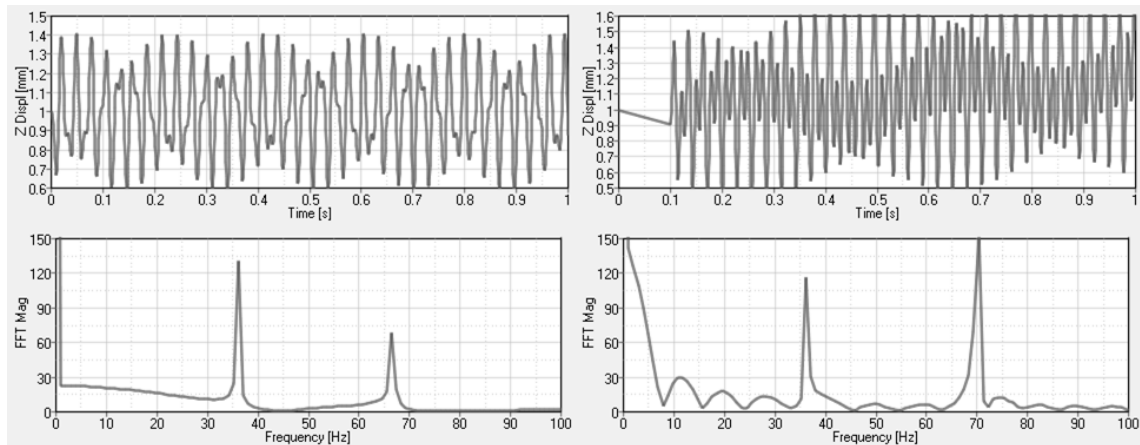
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 33Hz.



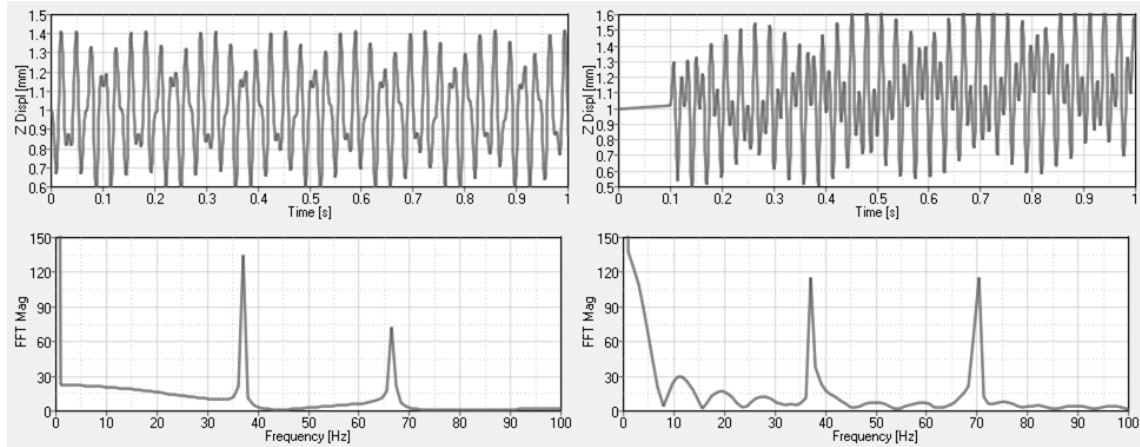
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 34Hz.



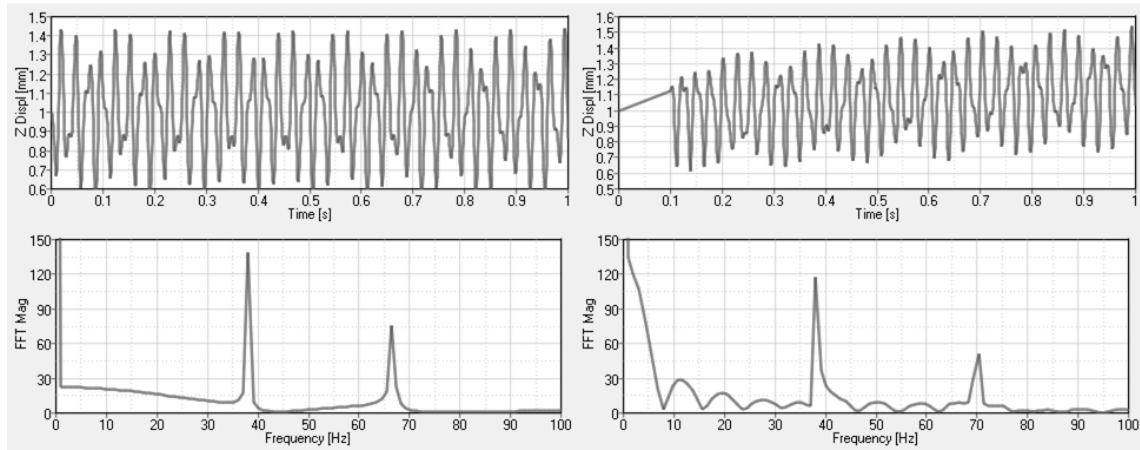
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 35Hz.



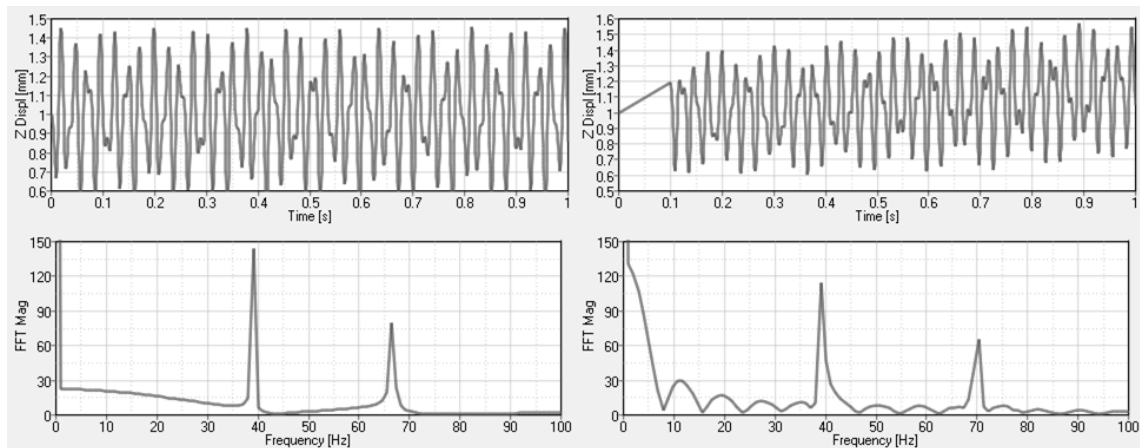
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 36Hz.



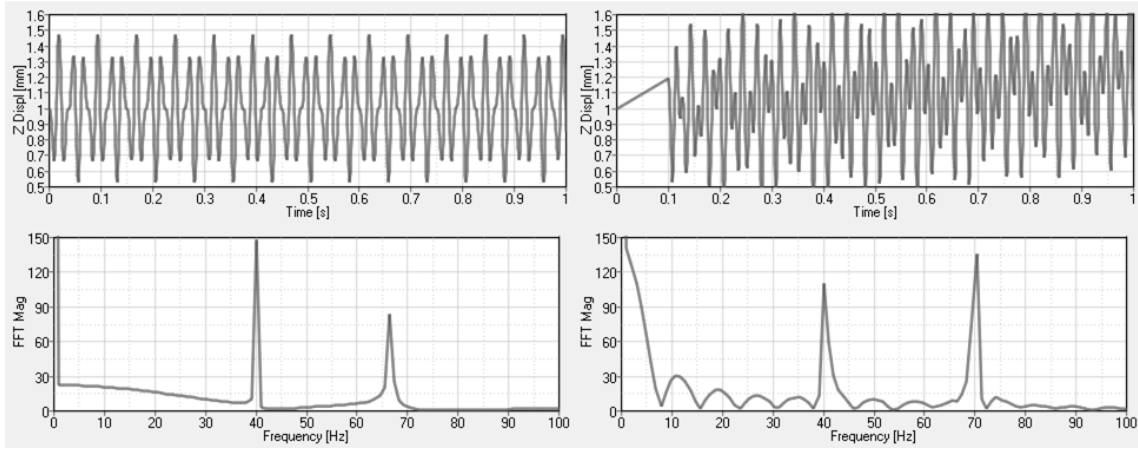
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 37Hz.



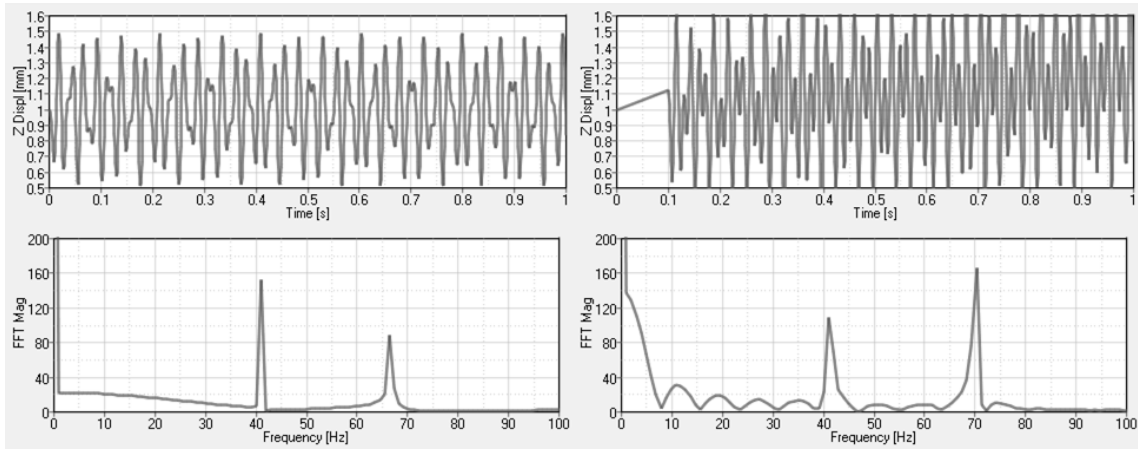
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 38Hz.



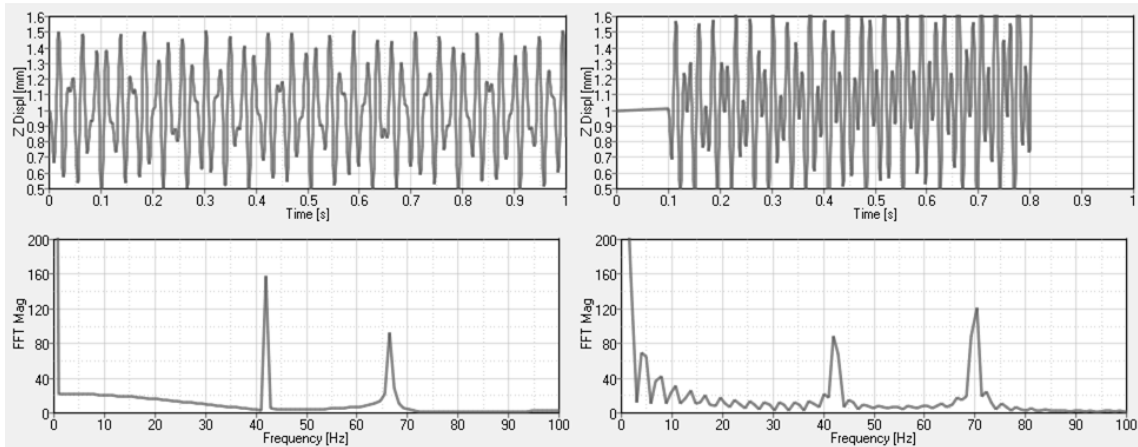
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 39Hz.



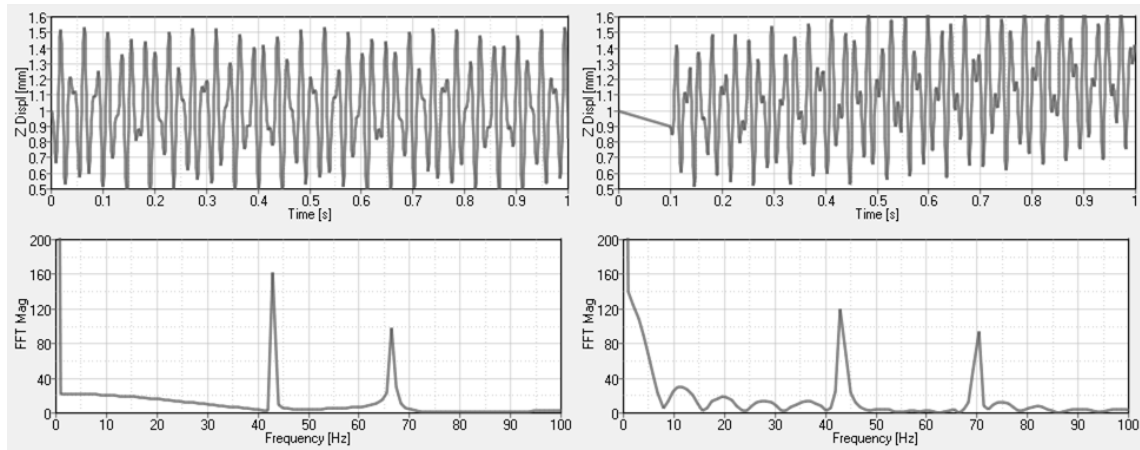
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 40Hz.



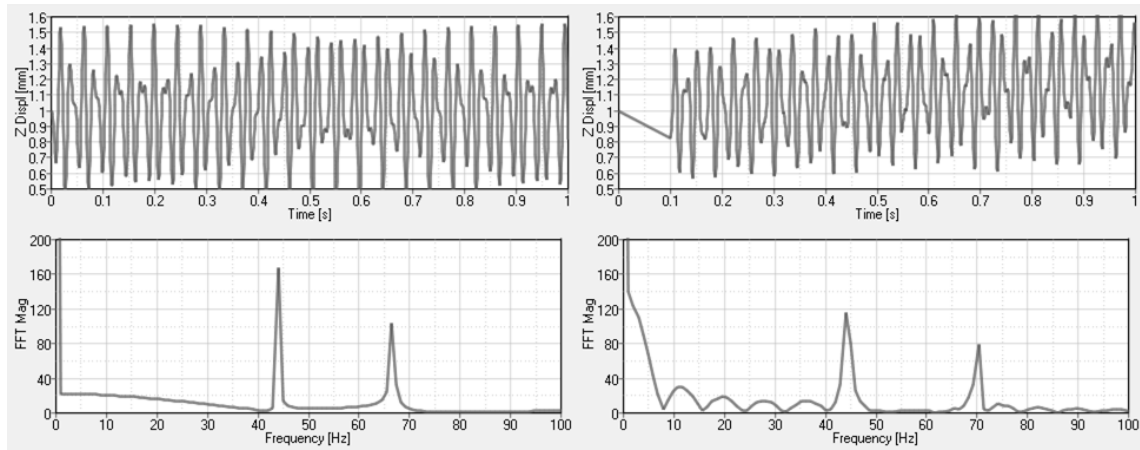
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 41Hz.



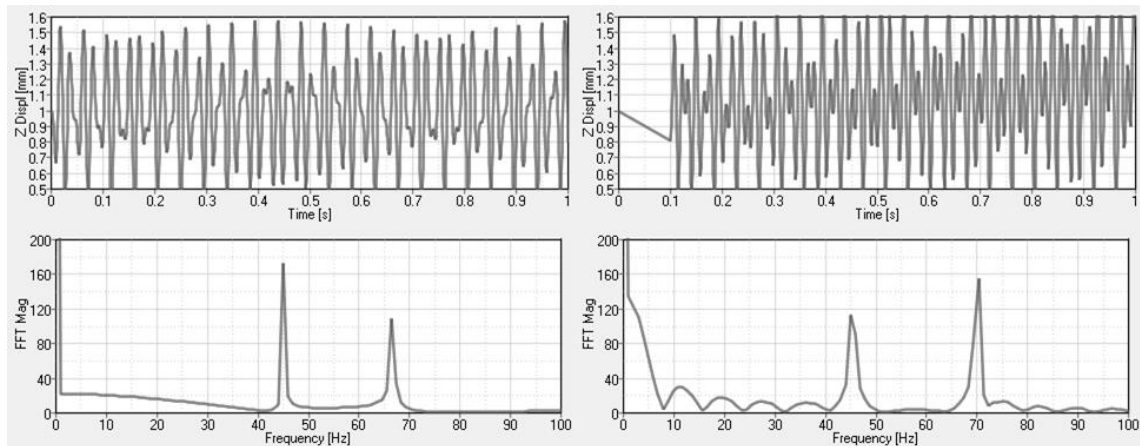
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 42Hz.



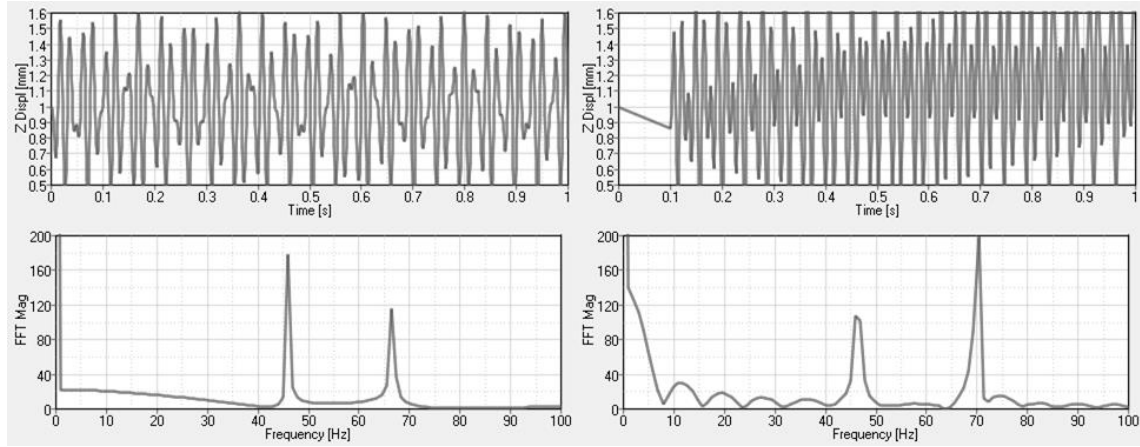
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 43Hz.



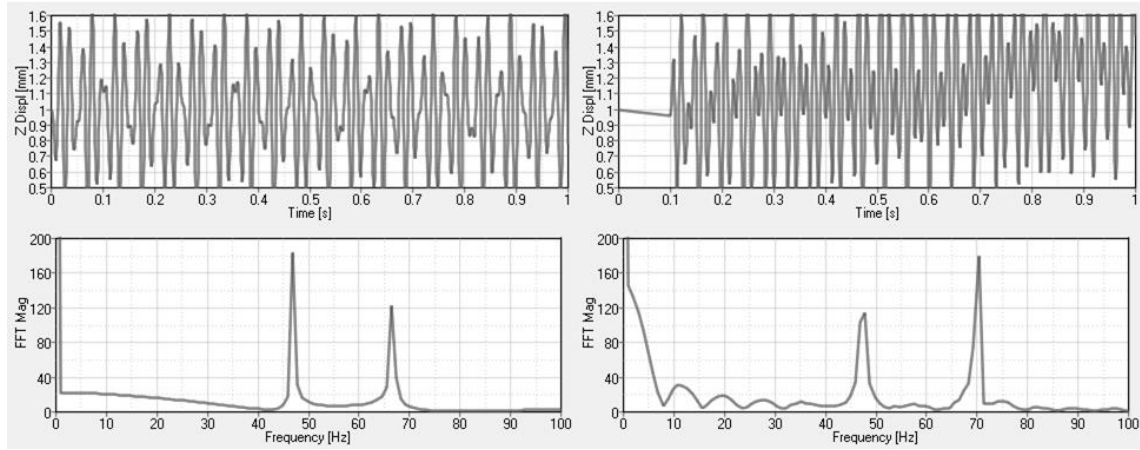
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 44Hz.



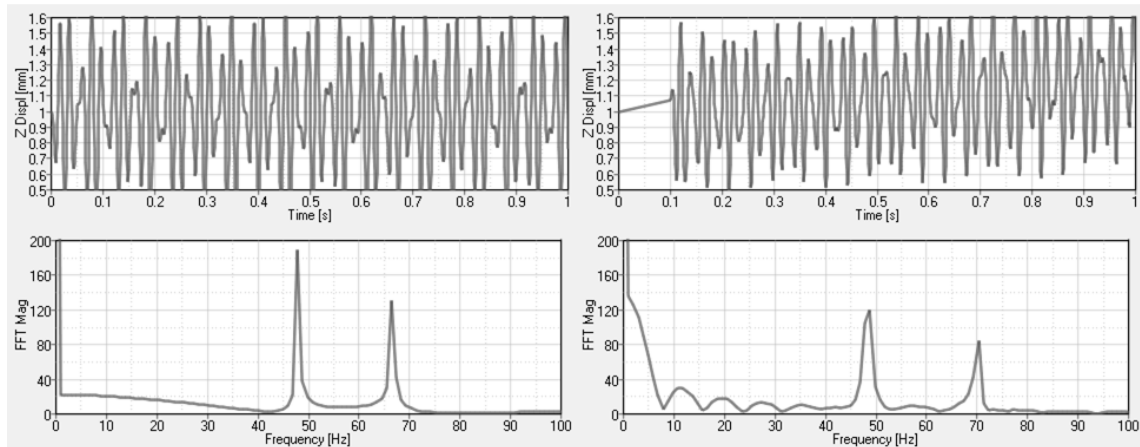
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 45Hz.



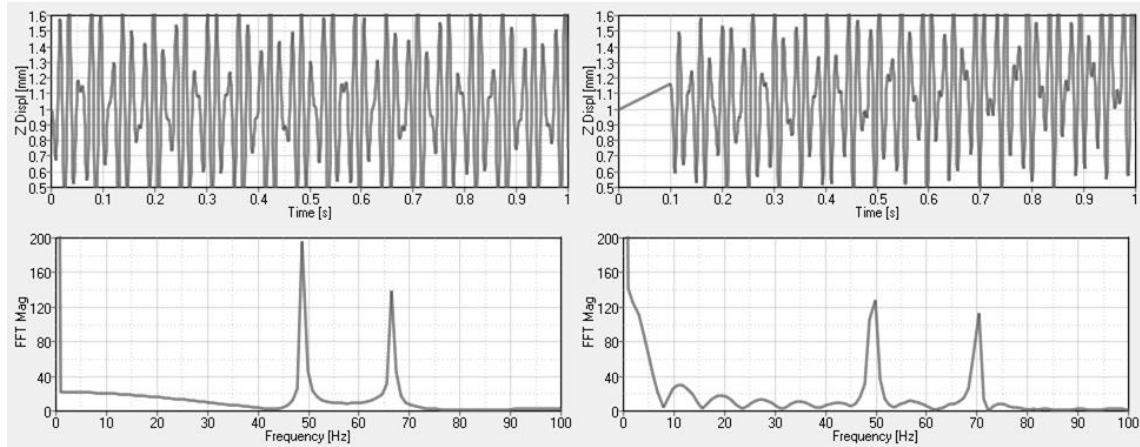
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 46Hz.



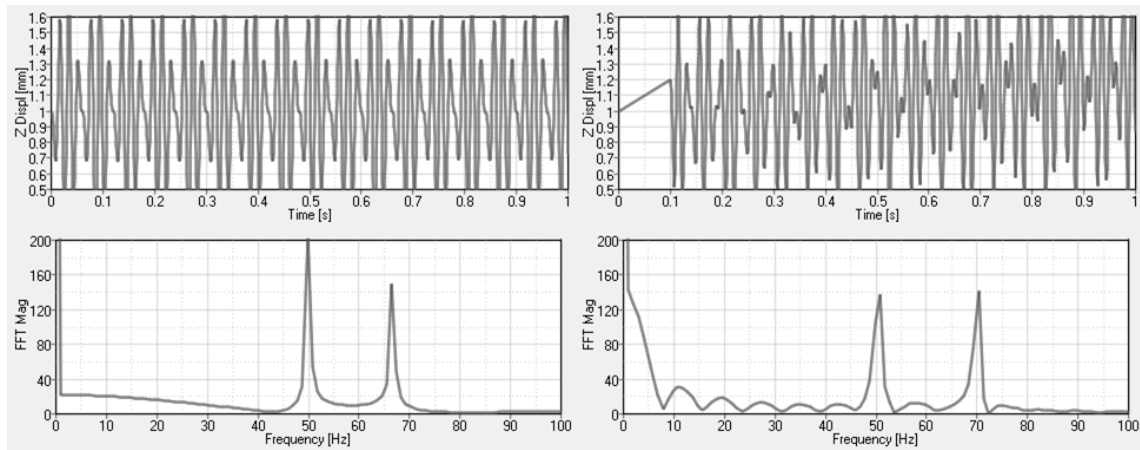
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 47Hz.



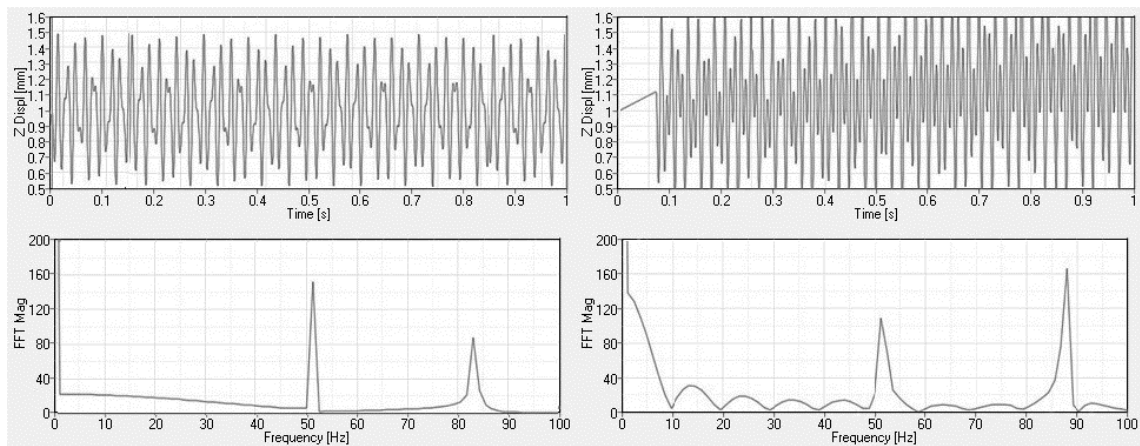
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 48Hz.



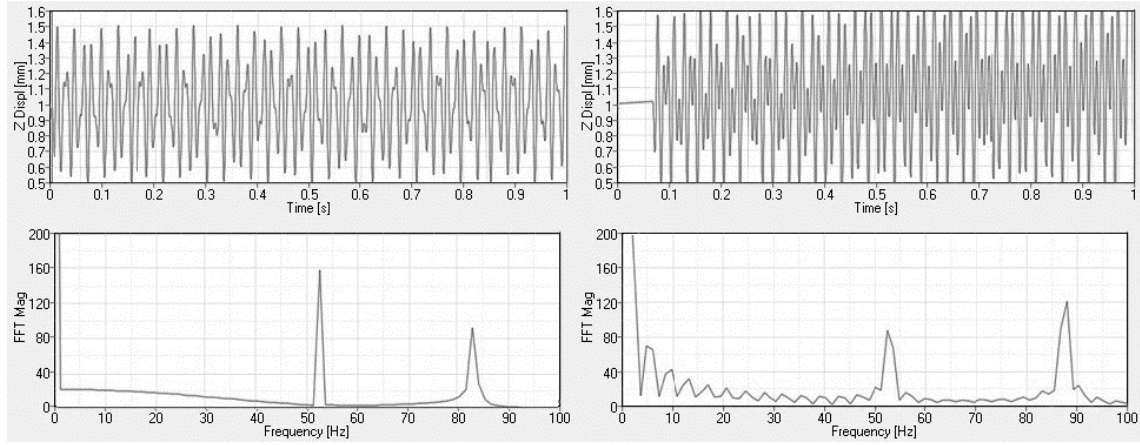
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 49Hz.



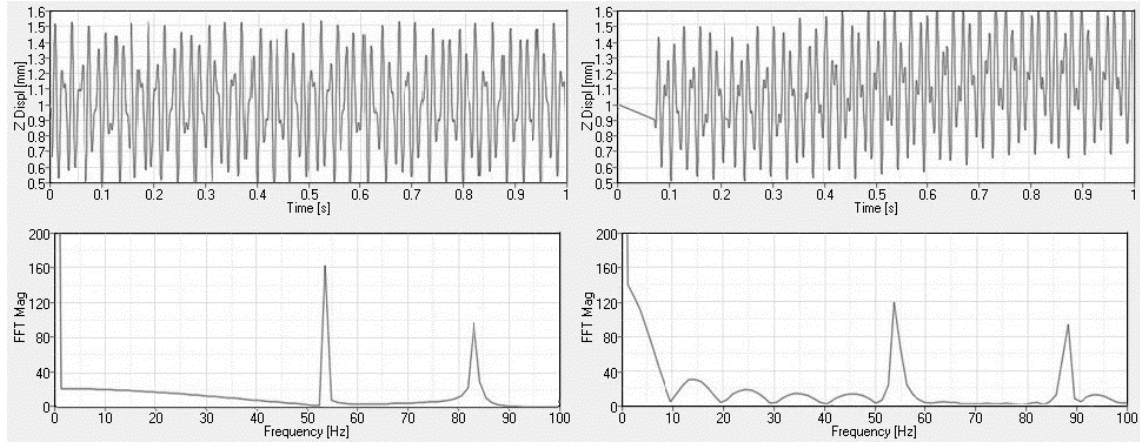
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 50Hz.



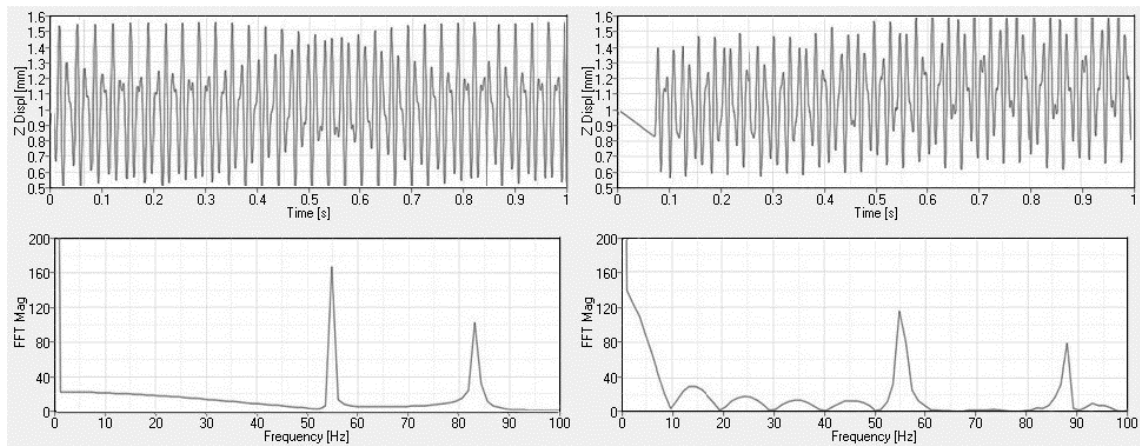
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 51Hz.



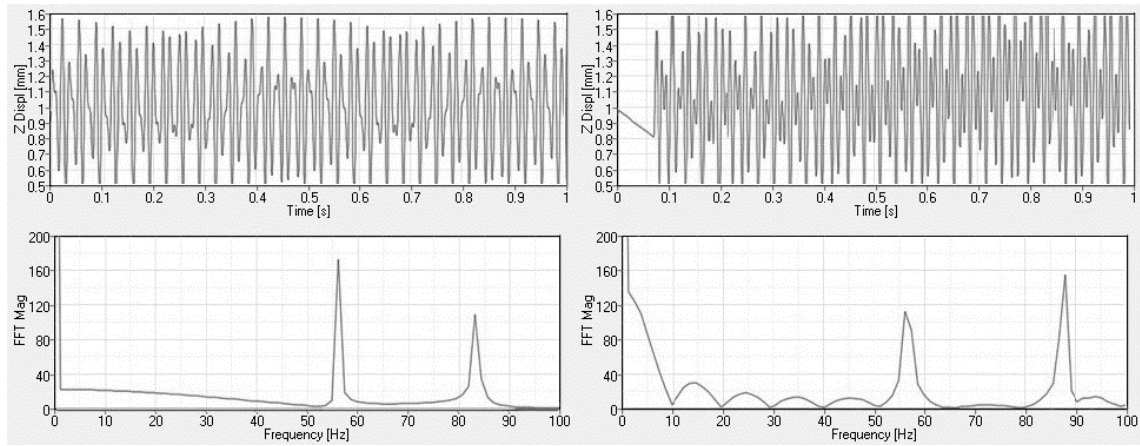
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 52Hz.



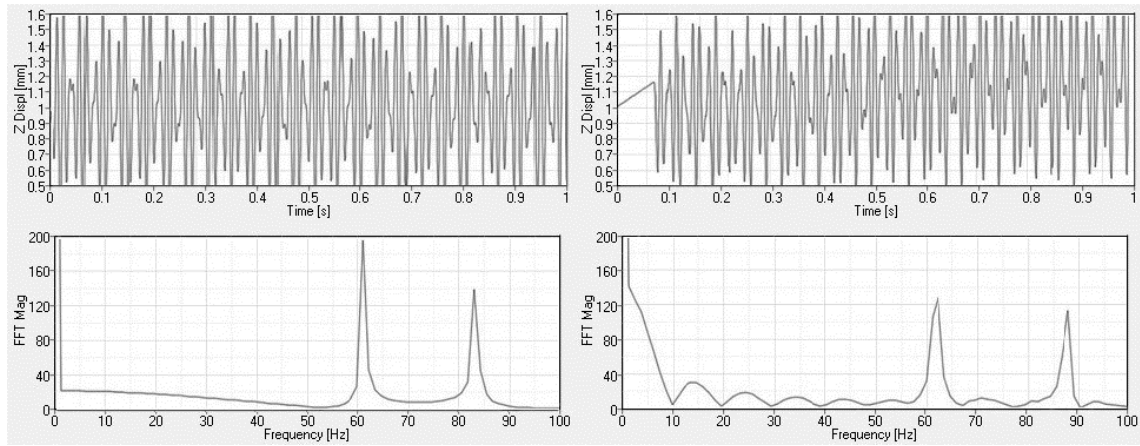
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 53Hz.



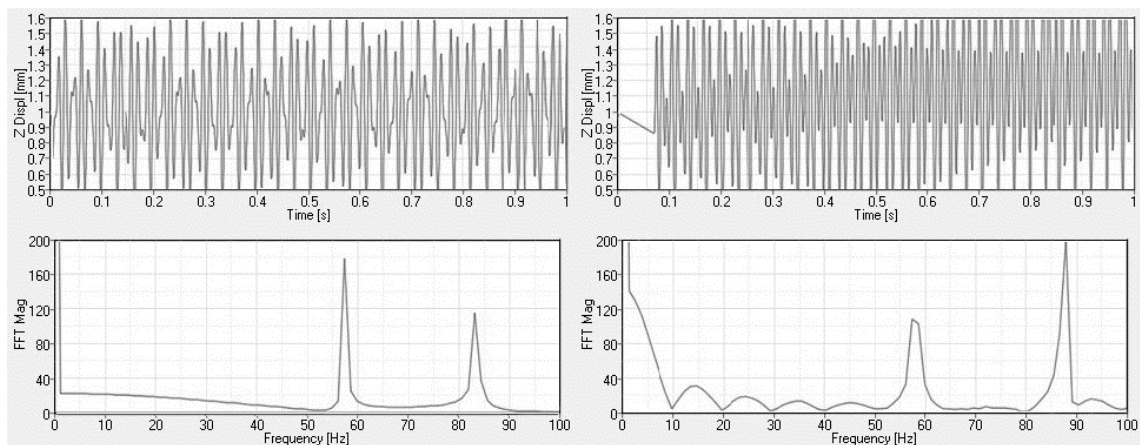
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 54Hz.



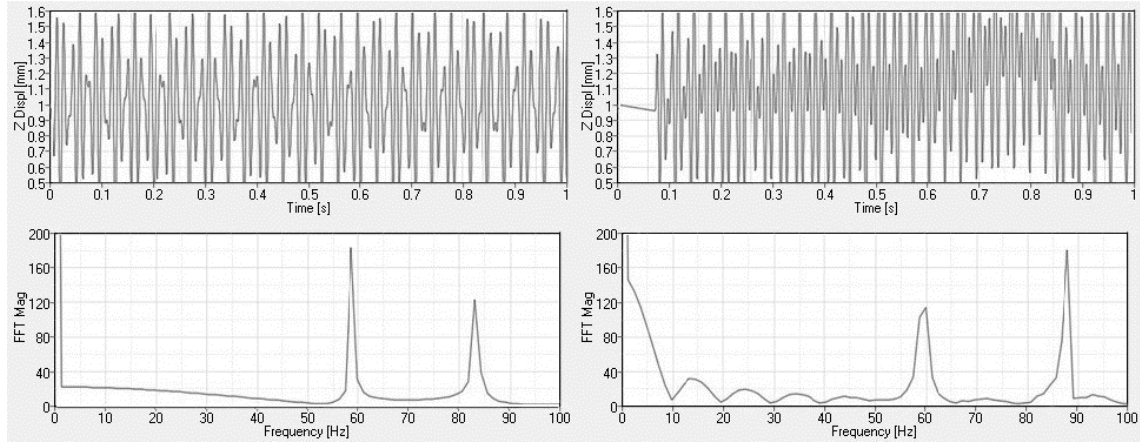
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 55Hz.



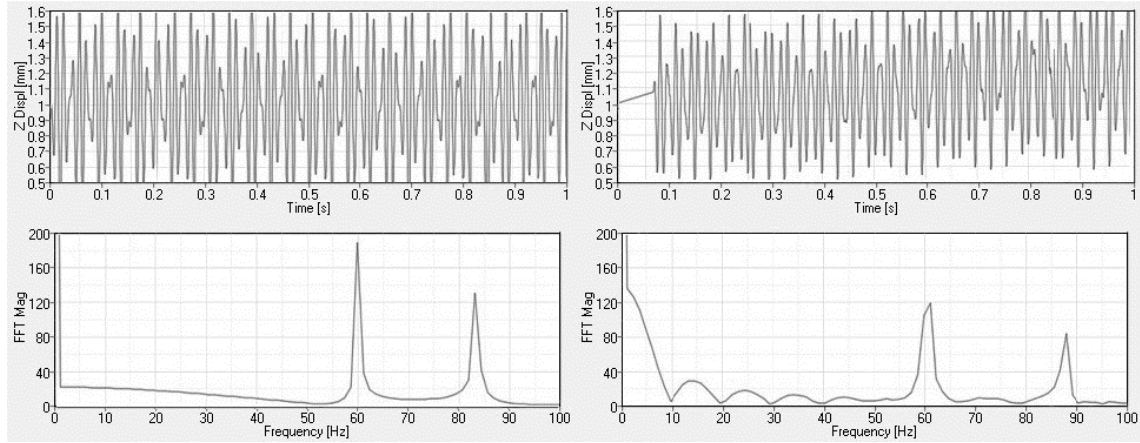
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 56Hz.



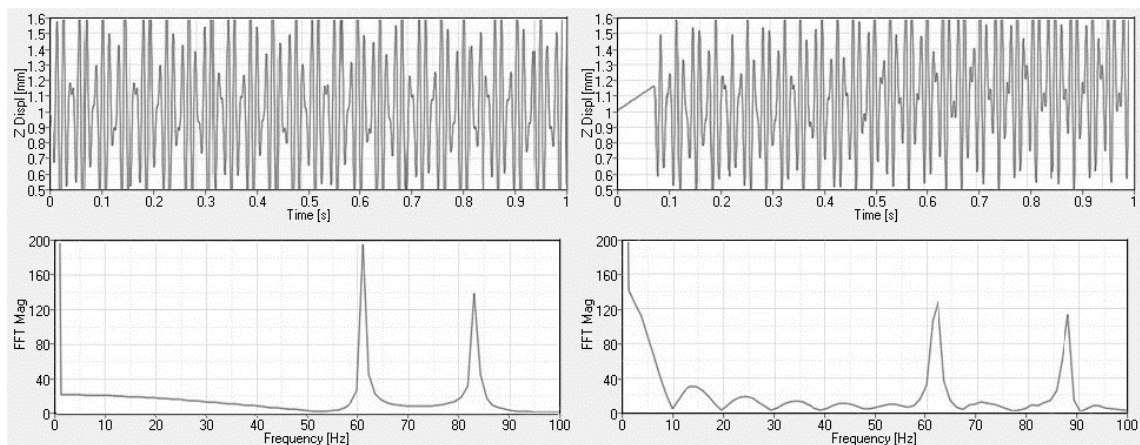
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 57Hz.



Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 58Hz.



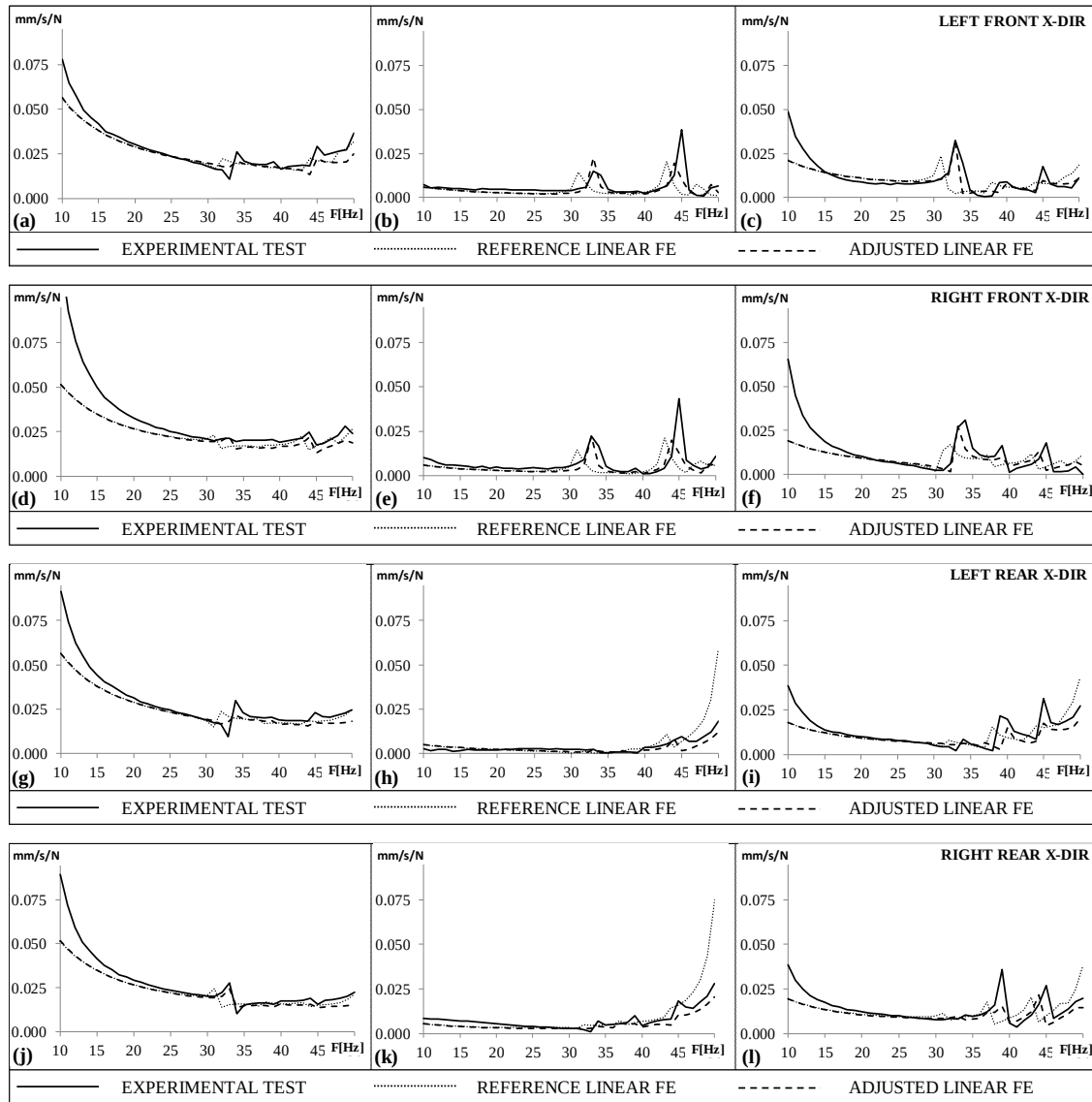
Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 59Hz.



Linear (left) and non-linear (right) FE reduced model time-domain (upper) to frequency-domain (bottom) conversion of FFT response under sinusoidal excitation at 60Hz.

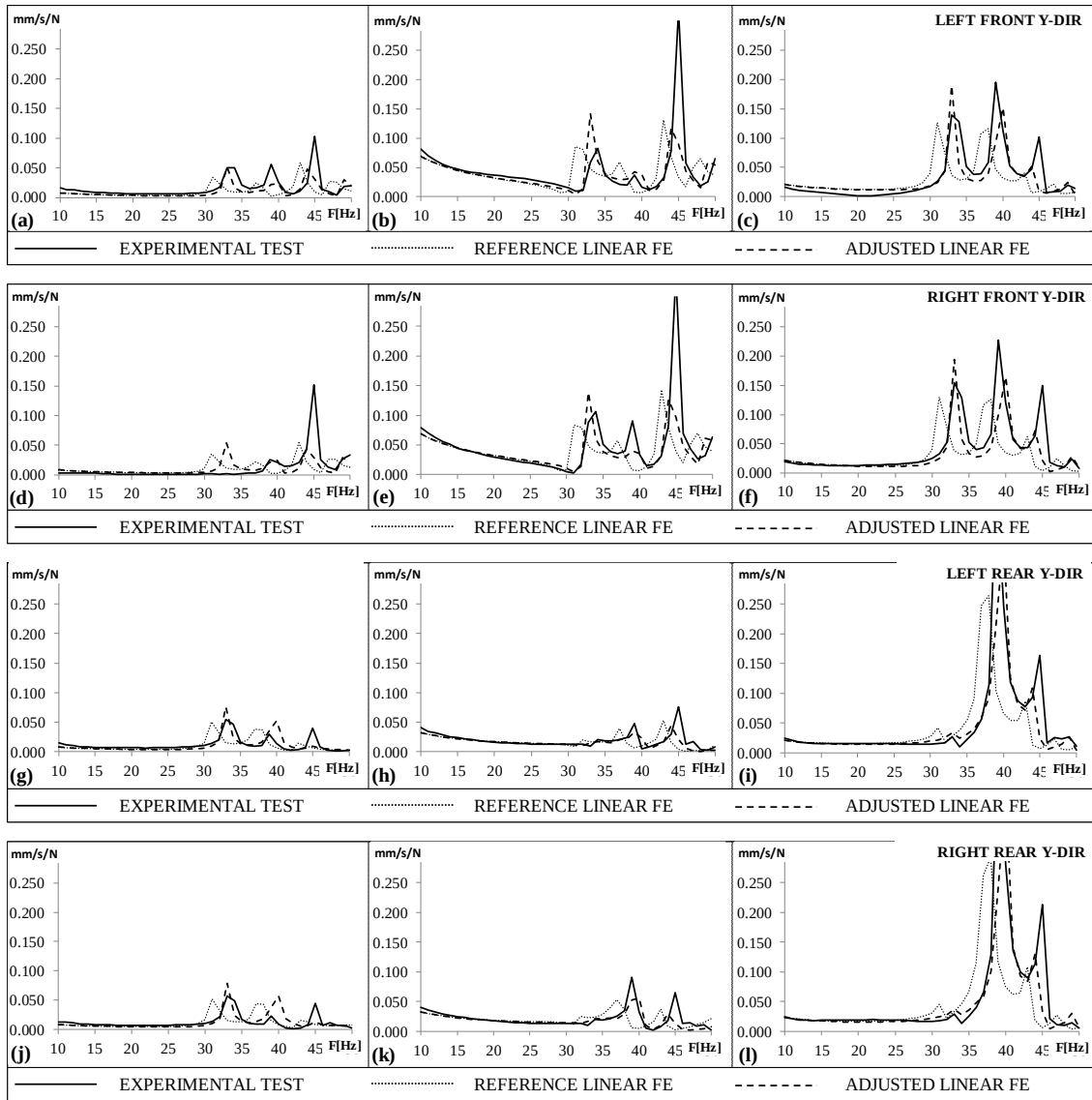
APPENDIX IX

Experimental test vs FE reference model vs FE adjusted model FRF plots:



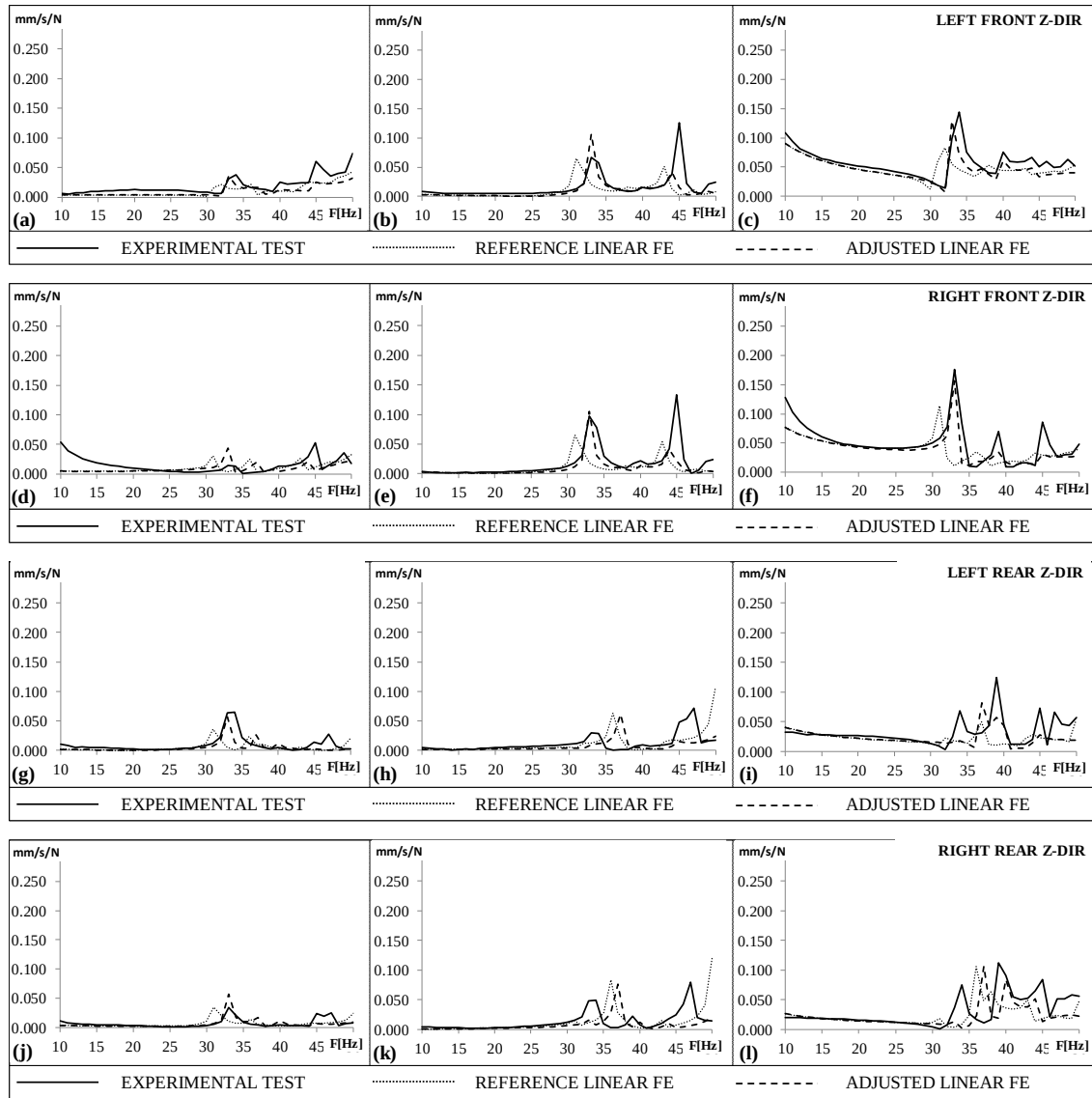
Experimental test *vs* reference linear FE *vs* adjusted linear FE Seat Track FRFs (vibration magnitude x-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* reference linear FE *vs* adjusted linear FE FRFs (vibration magnitude y-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir



Experimental test *vs* reference linear FE *vs* adjusted linear FE FRFs (vibration magnitude z-dir response to unitary excitation) at:

- (a) front left shock tower x-dir
- (b) front left shock tower y-dir
- (c) front left shock tower z-dir
- (d) front right shock tower x-dir
- (e) front right shock tower y-dir
- (f) front right shock tower z-dir
- (g) rear left shock tower x-dir
- (h) rear left shock tower y-dir
- (i) rear left shock tower z-dir
- (j) rear right shock tower x-dir
- (k) rear right shock tower y-dir
- (l) rear right shock tower z-dir

Article

Enhancing the Accuracy of Linear Finite Element Models of Vehicle Structures Considering Spot-Welded Flanges

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Abstract: Structural engineering simulations have required increasingly complex computational models to replace physical tests accurately. This work focuses on the numerical analysis of vehicle body structures, whose size and complexity make the use of very accurate nonlinear models unfeasible due to the prohibitive computational costs involved. The purpose of this study is to find a new approach to model spot-welded joints in linear finite element models of thin-wall vehicle body structures, improving the accuracy of the model without increasing its complexity. Using a set of simplified nonlinear models, we fitted the stiffness and damping properties of these welded joints and used those adjusted values into a linear model of the entire vehicle body structure. The results were compared with experimental tests, showing a clear improvement in the accuracy of the modal and frequency responses provided by the linear finite element model, but keeping its initial complexity level. The adjusted model was then used in an optimization analysis to reduce the structure's weight, leading to interesting cost savings and important reductions in the use of natural resources and carbon emissions.

Keywords: finite-element model; frequency response function; vibration testing; vehicle structures; correlation analyses



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1. Introduction

The level of correlation between the behaviour of a physical system and the results obtained from a finite element (FE) model is essential in many engineering areas. The accuracy of results from finite element analysis (FEA) increases with the correlation levels, thus reducing the demand for prototypes and providing high savings in time and resources for engineering projects. This statement is especially relevant in the automotive industry [1–3], where the use of FE models is necessary and determinant to accelerate product development cycles and increase the cost and time efficiency of new vehicle projects.

From this perspective, the validation of analytical simulation models became a decisive factor in developing new products, especially since the use of advanced computational resources became economically viable. In 1992, Baker [4] compared different methods to correlate analysis predictions with test data, including modal matrices and comparisons of frequency response functions. In 1999, Brughmans et al. [5] studied FEA model updating techniques for correlation improvement in order to support the implementation of a state-of-the-art NVH CAE design optimization environment. In 2005 Schedlinski et al. [6] presented an FEA model update technique based on a sheet metal gauge update. They used the Modal Assurance Criterion (MAC) to compare the modal consistency of the original and updated models. Splendi et al. [7] presented in 2013 a technique to tune the behaviour of damping patches in FE models to improve their correlation with experimental tests. They used the MAC to verify the modal consistency and compared the point mobility FRF main peak amplitudes between the experimental and numerical results. In 2017, Rotondella et al. [8] proposed a novel welding representation for metallic joints, using the

MAC to measure the numerical–experimental correlation and validate the newly proposed modelling technique.

Despite this continuous improvement of FEA [2,9–12], the accuracy level of linear models remains limited by the lack of nonlinear phenomena representation, such as friction and contact reaction forces. In addition, the complexity of nonlinear models needed to represent those phenomena limits its use to specific boundary conditions. Otherwise, it would be unfeasible to solve them considering the computational resources currently available. For this reason, linear models are traditionally mainly used in these situations.

In the case of a vehicle body subjected to a vibrational analysis, the structure is typically composed of a set of thin-wall stamped steel sheets connected by spot welds at their edges. As a consequence of these constructive properties, nonlinear contact between flanges of adjacent sheets will always occur. Considering the high complexity given by the deep detail level required, we decided to build and analyse the simulation models using the FE technique, taking advantage of its flexibility in design representation and its ability to simulate distinct mechanical elements. On the other hand, to perform vibration analyses, we should analyse these FE models using the eigenvalue approach [13,14] at least up to 100 Hz, to capture the highest energy modes of the structure, leading to an undesirable combination of a large nonlinear model with a very demanding computational effort.

In this work, we aim to present a new modelling methodology to increase the accuracy of linear FEA, thus avoiding exceeding the computational resources currently available. To this end, we compared the correlation levels obtained with both linear and nonlinear models of the whole vehicle structure.

Taking a vehicle body structure, we performed a set of bench tests to capture its modal behaviour and its frequency response functions (FRFs). We also built a reference FE model of the same vehicle body structure to simulate an identical analysis to the field tests performed on the bench test (Figure 1). We compared the results obtained from the reference FE model of the body structure with those obtained from the bench tests and computed the correlation coefficients.

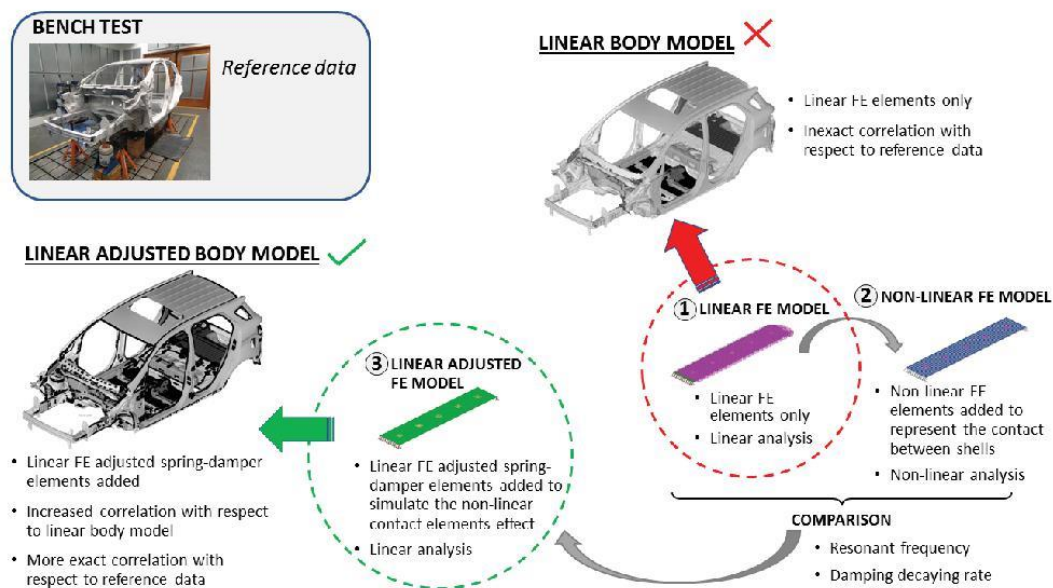


Figure 1. Scheme followed to develop the adjusted linear FE model of the vehicle body structure.

After that, we built three FE test models to represent a small part of the structure welded flange. We first built an FE test model using only linear parameters, thus avoiding nonlinear behaviours in the contact area between the flanges of adjacent panels. We then built a second nonlinear FE test model using the first model as a base, adding to the first model the nonlinear contacts of the flanged areas and computed the difference of behaviour between both models. We also built a third linear FE test model using the first model as a base but now adding linear spring-damper elements to simulate the nonlinear behaviour at the contact areas. The elastic constants of these spring-damper elements were adjusted using the differences found between the responses of the first and second FE models.

We finally built an adjusted FE model of the whole vehicle body structure, adding to the reference FE model of the body structure the linear spring-damper elements previously adjusted with the help of the FE test models. We then performed with this adjusted model the same analyses carried out with the bench test and the reference FE model, comparing the adjusted model results with the bench test results, and computing the correlation coefficients.

The results of the adjusted FE model of the body structure showed an improvement of 10.6% in the correlation coefficient, compared to the results of the initial reference model. The modal behaviour coefficients and the visual comparison of the FRF curves also show a higher correlation given by the adjusted model, confirming the validation of the adjustment methodology proposed here.

During the state-of-the-art review, we detected a knowledge gap regarding the representation of nonlinear contacts in large finite element models subjected to frequency domain analysis. To overcome this gap, the objective of this work is to find a new approach to model spot-welded flanged joints into linear finite element models of thin-wall vehicle body structures, using a nonlinear model behaviour as a reference. In this way, we could improve the accuracy of the modal and frequency response function results without increasing the complexity of the FE model analysis.

2. Materials and Methods

2.1. Tests Performed on the Bench Test. Reference Data

2.1.1. Test Definition

Based on the vehicle manufacturer's experience, we performed an experimental modal analysis of the physical structure of the vehicle body on a bench test (Figure 2, left), as explained in the following paragraphs. From this test, we generated a reference set of results to support the correlation analysis of the FE models.

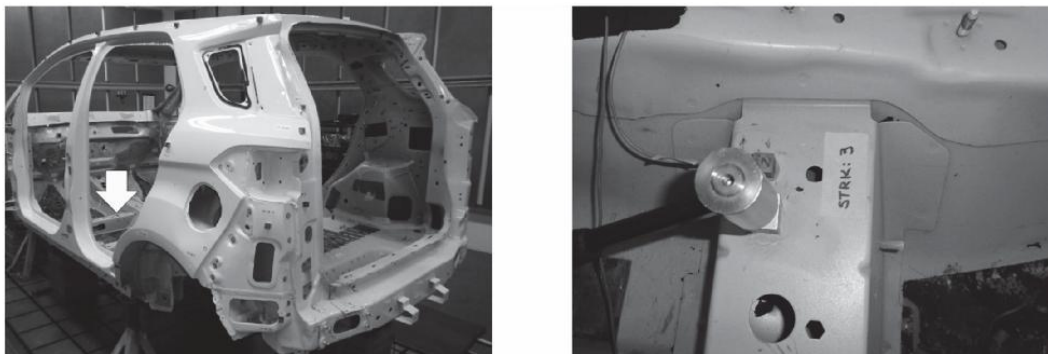


Figure 2. Experimental test body (left) and impact hammer load on the front seat attachment (right).

We isolated the vehicle structure from the other parts of the vehicle and supported it on a set of four air springs. Then, we distributed 152 triaxial accelerometers on the structural members and large panels, forming a mesh.

We used two electrodynamic shakers (Tira GmbH, Schalkau, Germany) to excite the body structure with a time-domain burst random noise. We also configured the shape of the power spectral density (PSD) of the input signal to concentrate the higher amount of energy within the frequency range from 10 to 60 Hz, where we can usually find the main eigenmodes.

We acquired the response of the vehicle body structure through a mesh of accelerometers (PCB Piezotronics, Depew, NY, USA). We identified the main body structure eigenmodes and gathered their eigenfrequencies and damping ratios. We performed three sequential runs with fifty sample records per run to cover all excitation and response points. In this way, the data analyses were more accurate and faster. We then processed these data to compute the eigenfrequencies and eigenmodes of the vehicle body structure.

We also used an impact hammer (PCB Piezotronics, Depew, NY, USA) to perform an impact test by exciting, one at a time, the four shock tower attachment points of the vehicle body structure. We applied a 1.0 N pulse at each point on the x, y, and z directions (dir) separately. We measured the magnitude and phase of the accelerations at the driver seat attachment point (Figure 2, right) in the x, y and z-dir.

We performed an average of 10 impacts per direction to smooth the transfer functions and improve data accuracy even more. We computed the fast Fourier transform (FFT) to convert the acquired data from the time to frequency domains. In this way, we obtained the frequency response function (FRF) for each point and direction.

In total, we acquired 72 FRF data sets (4 excitation points $\times$ 3 excitation directions $\times$ 3 response point directions $\times$ 2 complex roots—magnitude and phase). Although we used the entire data set for the calculations, we just plotted the amplitude of the root mean square (RMS) of the four excitation points to reduce the number of graphics shown.

2.1.2. Eigenfrequencies

Table 1 gathers the eigenfrequencies and damping ratios of the main eigenmodes of the body structure found in the bench test, from 10 to 60 Hz.

Table 1. Eigenfrequencies and damping ratios of the vehicle body structure from the bench test.

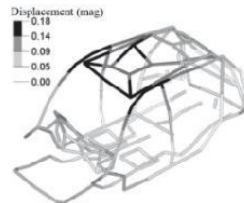
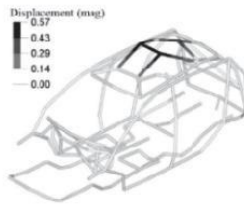
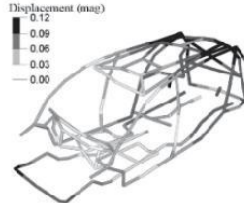
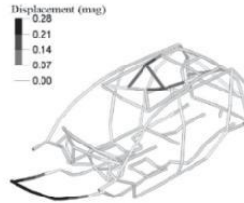
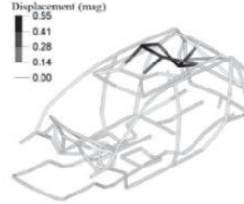
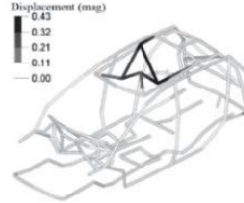
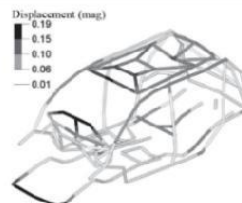
Mode	Frequency [Hz]	Damping c/co []	Mode Shape	Mode Picture
1	33.38	0.00513	Windshield Torsion	
2	33.38	0.00390	1st Moonroof Bending	
3	39.23	0.00567	Global Torsion	
4	44.71	0.00228	Global Lateral Bending	
5	46.34	0.00642	2nd Moonroof Bending	
6	49.34	0.00344	Moonroof Torsion	

Table 1. Cont.

Mode	Frequency [Hz]	Damping c/co []	Mode Shape	Mode Picture
7	53.73	0.00453	Global Vertical Bending	

2.2. Reference FE Model of the Vehicle Body Structure

2.2.1. Model Definition

We built a reference FE model of the vehicle body structure (Figure 3, left). We used the Hyperworks software pack to support the FE model building (Hypermesh), the FE analysis solving (Optistruct), the post-processing FRF graphics (Hypergraphics), visualization (Hyperview), MAC (Hyperview NVH module) and FRAC (Compose) calculations. We also used MS Excel to plot additional graphics other than FRFs.

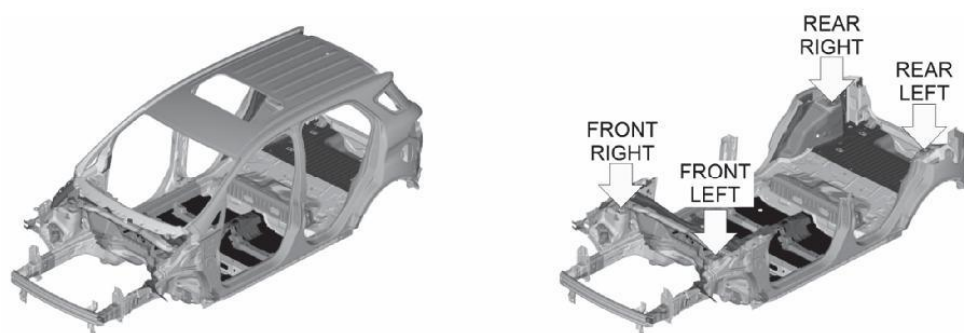


Figure 3. Reference FE model of the vehicle body structure (left) and body structure excitation points (right).

The FE model was meshed based on the vehicle manufacturer's experience, as summarized below. We used a 5 mm meshing size with a minimum size of 2 mm for refinement at complex design regions like the edges, flanges, beads, and attachment points. We used TRIA3 and QUAD4 2D shell elements to mesh stamped parts such as panels and structural beams; HEXA8 3D + RBE3 1D elements to represent spot welds and adhesive connections; RBE2 1D elements to represent bolts and nuts, and CBUSH 1D spring-damper elements for the adjusted model tuning. We used linear material properties to represent the stamped parts and the welds, made of steel (Young's modulus = 210.0 GPa, Poisson's ratio = 0.30 and density = 7900 Kg/m³), as well as the polymeric adhesive material (Young's modulus = 1.9 GPa, Poisson's ratio = 0.42 and density = 1190 Kg/m³).

We performed a free-free modal analysis (Normal Modes solver solution) with the reference FE model to get the same mode shapes [15] as those obtained in the bench test. We also simulated the hammer impact test (Frequency Response Modal solver solution) using a frequency response analysis to find the FRFs from the four shock tower attachments and considering the six degrees of freedom [16] (Figure 3, right), like in the bench test. We extracted the eigenvalues using the Lanczos' method, considering a structural critical damping ratio of 0.008 [17,18] from 0 to 100 Hz.

We used the Modal Assurance Criterion (MAC) and the Frequency Response Assurance Criterion (FRAC) metrics to measure the differences between the results obtained with the bench test and the reference FE model.

The MAC [19–21] is a normalized single-value metric that estimates the consistency between eigenvectors from different sources. We computed it to assess the accuracy of the modal behaviour of the reference FE model. We contrasted the eigenvectors found from the reference FE model with those found in the bench test, focusing on the frequency range of concern, from 10 to 60 Hz. We computed the MAC as follows [21]:

$$MAC_{(a,x)} = \frac{\left| \sum_{j=1}^{N_f} \{\varphi_a\}_j \{\varphi_x\}_j \right|}{\left(\sum_{j=1}^{N_f} \{\varphi_a\}_j^2 \right) \left(\sum_{j=1}^{N_f} \{\varphi_x\}_j^2 \right)} \quad (1)$$

where the eigenvectors $\{\varphi_a\}$, extracted from the reference FE model, were compared with the reference eigenvectors $\{\varphi_x\}$, extracted from the bench test data. N_f refers to the mode number, in ascending order.

The FRAC [20–23] is a frequency-dependent normalized single-value metric that estimates the correlation between two FRFs with the same excitation and response points. We computed it to assess the accuracy of the transfer functions found with the reference FE model. We contrasted the FRFs found from the reference FE model with those found in the bench test, restricting to the frequency range of concern, from 25 to 60 Hz, where the FRF peaks are most representative. We computed the FRAC as follows [20]:

$$FRAC_{(a,x)} = \frac{\left| \sum_{j=1}^{N_f} (\mathbf{H}_a(\omega_j))^H \cdot \mathbf{H}_x(\omega_j) \right|^2}{\left[\sum_{j=1}^{N_f} (\mathbf{H}_a(\omega_j))^H \cdot \mathbf{H}_a(\omega_j) \right] \left[\sum_{j=1}^{N_f} (\mathbf{H}_x(\omega_j))^H \cdot \mathbf{H}_x(\omega_j) \right]} \quad (2)$$

where the FRFs $\mathbf{H}_a$, extracted from the reference FE model, were compared with the reference FRFs $\mathbf{H}_x$, extracted from the bench test data. Both $\mathbf{H}_a$ and $\mathbf{H}_x$ are complex functions. The superscript $\mathbf{H}$ refers to the Hermitian, which is the transpose of the complex conjugate. N_f and ω refer to the mode number and to the frequency value in ascending order.

2.2.2. MAC Matrices

Table 2 gathers the MAC between the reference FE model of the vehicle body structure and the experimental test modes, the first obtained from the modal analysis simulation of the reference FE model and the second from the hammer impact bench test. From these results, we found that the main diagonal terms of the MAC matrix are higher than 0.9, which means that the modal results of the FEA are consistent with the experimental results [21].

Table 2. MAC results for the reference FE vehicle body structure model.

		BENCH TEST EIGENFREQUENCIES [Hz]						
		33.4	33.4	39.2	44.7	46.3	49.3	53.7
REFERENCE FE MODEL EIGENFREQUENCIES [Hz]	31.3	0.97	0	0	0	0	0	0
	36.2	0.02	0.98	0	0	0.07	0	0
	37.5	0	0.01	0.98	0	0	0	0
	43.2	0	0	0.01	0.96	0	0.01	0
	47.3	0.01	0	0	0.04	0.01	0.91	0
	50.2	0	0.09	0	0	0.95	0.02	0
	53.0	0	0	0	0	0.06	0	0.93

We also noticed that the fifth and sixth modes are switched. We considered this switch to be an undesirable effect of the small number of accelerometers used in the bench test to

acquire the local modes in the moonroof opening region. Even so, the main diagonal terms of the MAC matrix have high coefficients (>0.9) for all global modes. In any case, the effect of these switched modes affects only the moonroof area. Thus, we concluded that, despite this permutation of modes, the results are still consistent.

2.2.3. FRAC Results

Table 3 shows the results of the FRAC analysis between the reference FE model and the experimental test of the vehicle body structure.

Table 3. FRAC results for the reference FE model of the vehicle body structure.

Impact Point/Dir			FRAC	Impact Point/Dir			FRAC	Impact Point/Dir			FRAC
REFERENCE FE MODEL RESPONSE POINT: X-DIR	front left/x		0.9560	REFERENCE FE MODEL RESPONSE POINT: Y-DIR	front left/x		0.2315	REFERENCE FE MODEL RESPONSE POINT: Z-DIR	front left/x		0.8497
	front left/y		0.2270		front left/y		0.1654		front left/y		0.1428
	front left/z		0.4164		front left/z		0.2343		front left/z		0.6700
	front right/x		0.9320		front right/x		0.1547		front right/x		0.5928
	front right/y		0.1732		front right/y		0.1720		front right/y		0.1246
	front right/z		0.3994		front right/ z		0.2403		front right/ z		0.4808
	rear left/x		0.9647		rear left/x		0.2690		rear left/x		0.2206
	rear left/y		0.8365		rear left/y		0.2645		rear left/y		0.4020
	rear left/z		0.7581		rear left/z		0.1026		rear left/z		0.4220
	rear right/x		0.9682		rear right/x		0.2488		rear right/x		0.3059
rear right/y		0.8887	rear right/y		0.2379	rear right/y		0.3238			
rear right/z		0.6374	rear right/z		0.1110	rear right/z		0.5542			
Average x-dir			0.6798	Average y-dir			0.2027	Average z-dir			0.4241
-	-	-	-	-	-	-	Overall average			0.4355	

2.3. FE Test Models

2.3.1. Model Definition

From the MAC and FRAC results, we detected a gap between the results obtained with the bench test and the reference FE model of the vehicle body structure. Since the reference FE model already includes all linear structural components and connections, we could not increase its accuracy just by considering the linear boundary conditions. Therefore, the FE model should also include other nonlinear elements. Since the contacts between the welded flanges of the vehicle body parts were present in the whole structure, we would also include them in the FE model of the body structure.

However, the change from linear to nonlinear boundary conditions, given by the addition of nonlinear contact elements on the whole structure, will exponentially increase the required calculation efforts. Therefore, it would be unfeasible to analyse this nonlinear FE model using the current computational resources available.

Taking into account this limitation, we built a linear FE test model (Figure 4, left), replicating at a small scale the design of the flanges of the vehicle body structure, according to the following parameters:

- Plate size [mm]: $250 \times 50 \times 1.0$;
- Gap between upper and lower plates: 1.0 mm;
- Spot welds: 5 spots located every 50.0 mm; and
- System support condition: simply supported.

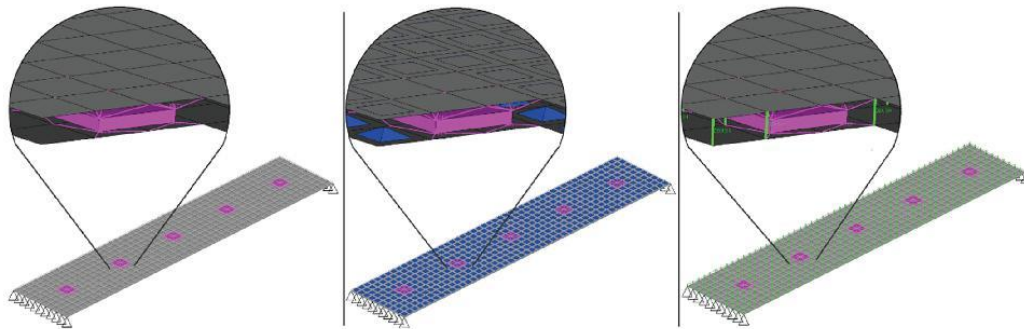


Figure 4. FE test models: without contact elements (left), with nonlinear contact elements (center) and with linear adjusted contact elements (right).

We also built a nonlinear FE test model (Figure 4, centre) by adding nonlinear contact elements to the linear FE test model in the surroundings of each spot weld. We used a contact static friction coefficient $\{\varphi_s\} = 0.25$ [24–26]. We also built a third FE test model with linear contact elements (Figure 4, right), adjusting their stiffness and damping coefficients by comparing the results of the other two FE test models.

2.3.2. Comparison of the First Eigenfrequencies

We performed a first set of analyses to assess the differences in the first eigenfrequency between the linear and nonlinear FE test models. To do this, we applied a load of 0.01 N to all nodes of the upper plate, exciting the system with a time-dependent sinusoidal enforced motion during 1 s, with a frequency ranging from 10 to 60 Hz, with a 1 Hz step.

Since the plate deflection shows a bending shape, we plotted the FFT of the deflection of the central node of the upper plate. From the FFT curves, we identified the first eigenfrequency of each model.

For the FE test model with only linear boundary conditions (Transient Direct solver solution), without contact representation, we found the first eigenfrequency at 66.4 Hz, with no variation of the results (<0.001 Hz) over the excitation frequency range (Figure 5).

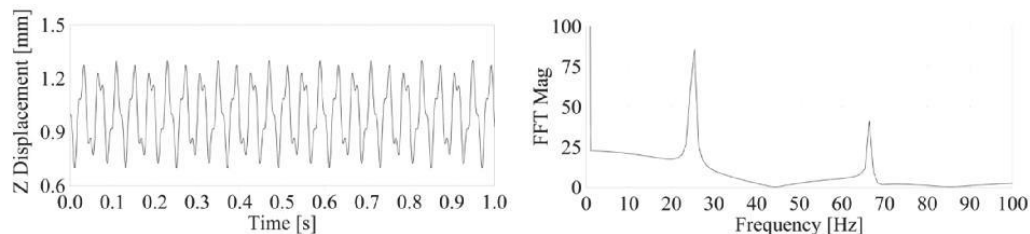


Figure 5. Linear FE model time response (left) and FFT response (right).

From the FE test model with nonlinear boundary conditions (Transient Non-Linear solver solution) and including nonlinear contacts, we found the first eigenfrequency at 70.3 Hz, with no other relevant results variation (<0.001 Hz) over the excitation frequency range (Figure 6). We used the difference between the first results of the two first FE test models (3.9 Hz) to adjust the stiffness of the spring-damper contact elements included in the third linear FE test model.

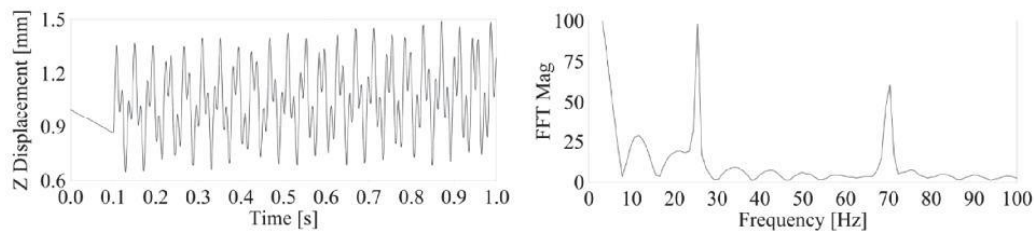


Figure 6. Nonlinear FE model time response (left) and time-to-frequency domain conversion of FFT response (right), at 25 Hz.

Since the results for the first eigenfrequency remain constant over the frequency range of interest (from 10 to 60 Hz), we used a linear interpolation approach. We took the first eigenfrequency of the linear FE reduced model, $F_a = 66.4$ Hz as a starting point. Since we did not include spring-damper elements in this model, we assumed its stiffness as $K_a = 0.0$ N/mm. Next, we added spring-damper elements to the linear FE test model using an arbitrary stiffness value of $K_b = 1000$ N/mm. We performed the same transient direct analyses as before and found the first eigenfrequency, F_b , at 71.3 Hz, with no results variation over the excitation frequency range. We then interpolated the first eigenfrequency of the nonlinear FE test model (70.3 Hz) between points a and b (Figure 7) and found the corresponding adjusted stiffness $K_{adj} = 801.0$ N/mm.

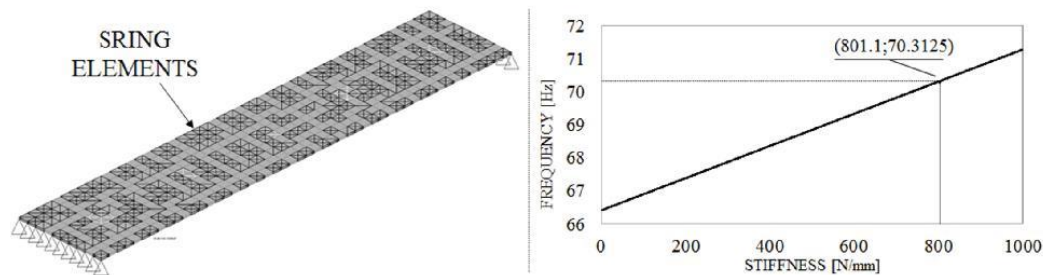


Figure 7. Adjusted linear FE reduced model (left) and spring element value interpolation (right).

We also validated the adjusted linear spring-damper elements ($K_{adj} = 801.0$ N/mm) by adding them to the linear FE test model. In this way, we found the first eigenfrequency at 71.3 Hz, the same value found with the nonlinear FE test model.

2.3.3. Comparison of the Damping Decay Curves

We performed a second set of analyses to assess the differences in the damping decay curves between the linear and nonlinear FE test models. To do this, we applied a pulse of 1 N on the central node of the upper plate for 0.01 s (Transient Modal solver solution). We took the time response of the node displacement over a 1 s span (Figure 8, left) and represented its envelope (Figure 8, right). In this way, we found the exponential damping decay curves for both the linear and nonlinear FE test models.

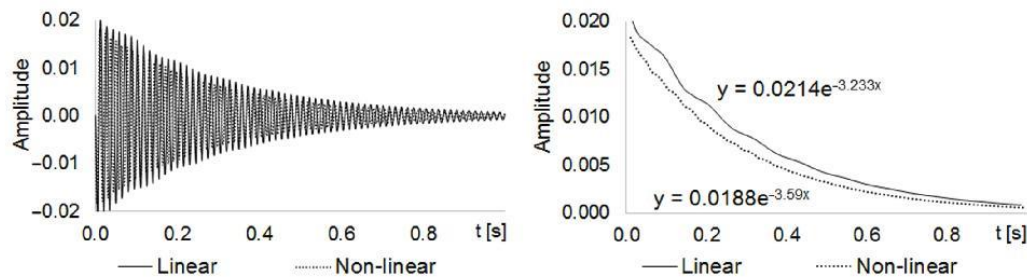


Figure 8. Linear and nonlinear z-dir displacement (left) and decaying rate (right) curves.

The exponents of the equations of these curves are their decaying rates, σ . We can also use this variable to calculate the damping ratio, ζ , of an undamped system [27], given by:

$$\zeta = \frac{1}{\sqrt{1 + \left(\frac{2\pi}{\sigma}\right)^2}} \quad (3)$$

being $\sigma \triangleq \ln \frac{y_1}{y_2}$ where y_1 and y_2 are the respective vibration amplitudes at two successive peaks of a decaying vibration, from which the decay rates $\sigma_{lin} = -3.233$ and $\sigma_{non-lin} = -3.590$ were calculated. It is also known that:

$$\sigma = \frac{B}{2m} \quad (4)$$

From Equation (4), assuming that both linear and nonlinear systems have the same mass, we found that $B_{non-linear} = 1.1104 \cdot B_{linear}$. We then used this relationship to adjust the damping of the linear FE spring-damper elements, to simulate the effect of the nonlinear contacts.

We also validated the adjusted linear spring-damper elements by adding them to the linear FE test model. Figure 9 shows the decay curves and equations found for the adjusted linear FE test model and the nonlinear FE test model.

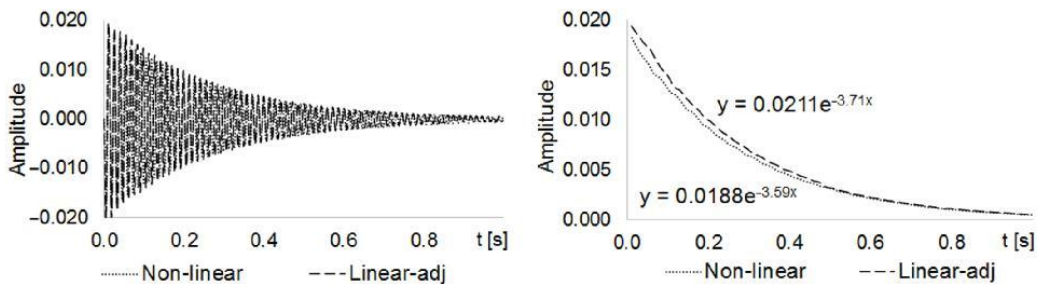


Figure 9. Nonlinear and linear adjusted z-dir displacement (left) and decaying rate (right) curves.

Comparing the decay rate of the non-adjusted ($\sigma_{lin} = -3.233$) and adjusted ($\sigma_{adj} = -3.715$) linear FE test models with the decay rate of the nonlinear FE test model ($\sigma_{non-lin} = -3.590$), we found that the difference between the decay rates of the linear and nonlinear models decreased from 11.04% to 3.34% when using adjustment. This comparison can also be visually confirmed by comparing the curves and decay rates shown in Figures 8 and 9.

3. Results

3.1. Adjusted FE Model of the Vehicle Body Structure

3.1.1. Model Definition

We then built an adjusted linear FE model of the vehicle body structure (Figure 10). To this end, we added to the reference FE model of the vehicle body structure a set of adjusted spring-damper elements along all the welded flanges of the vehicle structure. For this purpose, we used the adjusted stiffness and damping values, $K_{adj} = 801.0 \text{ N/mm}$ and $B_{adj} = 1.1104 \text{ N.s/mm}$, previously calculated from the FE test models. From this adjusted FE model of the vehicle body structure, we obtained the same mode shapes and FRFs [15] as for the bench test and the reference FE model of the vehicle body structure.

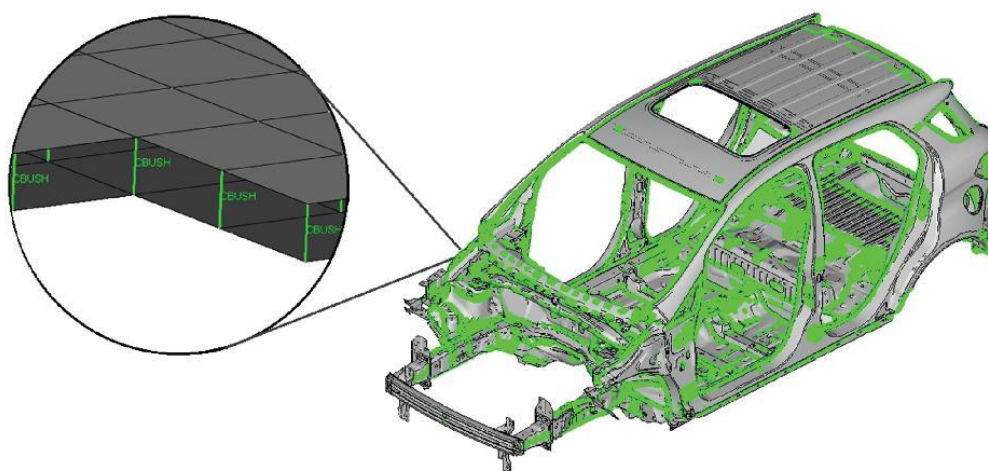


Figure 10. Adjusted linear FE model showing the added spring-damper elements.

Like for the bench test and the reference FE model of the vehicle body structure, we also simulated the impact hammer test from 0 to 100 Hz. To do so, we performed a frequency response analysis to capture the FRFs. We extracted the real part of the eigenvalues using the Lanczos' method, considering the average structural damping ratio of 0.0044815, as found during the bench test.

Table 4 and Figure 11 show the eigenfrequencies and FRFs of the adjusted FE model of the vehicle body structure. They also gather those found from the bench test and the reference FE model of the vehicle body structure.

Table 4. Eigenfrequencies of the bench test and the reference and adjusted FE models of the vehicle body structure.

Mode	Modal Shape	Bench Test [Hz]	Reference FE Model [Hz]	Adjusted FE Model [Hz]
1	Windshield Torsion	33.38	31.33	33.40
2	1st Moonroof Bending	33.38	36.22	37.10
3	Global Torsion	39.23	37.52	40.01
4	Global Lateral Bending	44.71	43.19	44.87
5	2nd Moonroof Bending	46.34	47.32	49.65
6	Moonroof Torsion	49.34	50.17	52.43
7	Global Vertical Bending	53.73	53.02	55.89

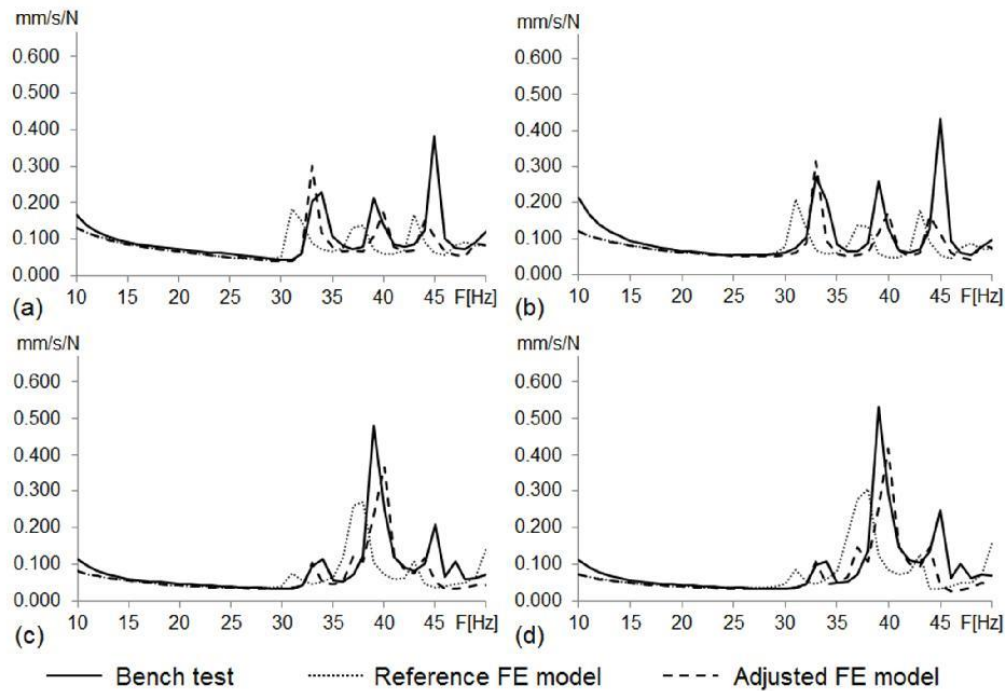


Figure 11. RMS curves from body structure bench test, reference and adjusted FE model FRFs (driver seat attachment vibration responses to unitary excitation). (a) Load at front left shock tower; (b) at front right shock tower; (c) at rear left shock tower; (d) at rear right shock tower.

We assessed the accuracy of the new adjusted model using the MAC and FRAC metrics to compare the modal behaviour and the FRFs obtained with this model in comparison with the bench test.

Tables 5 and 6 show the MAC and FRAC metrics of the adjusted FE model of the vehicle body structure, together with those of the reference FE model of the vehicle body structure. We calculated them by contrasting the results of the reference and adjusted FE models of the vehicle body structure with the bench test results.

Table 5. MAC results of the adjusted FE model of the vehicle body structure.

		BENCH TEST EIGENFREQUENCIES [Hz]						
		33.4	33.4	39.2	44.7	46.3	49.3	53.7
REFERENCE FE MODEL EIGENFREQUENCIES [Hz]	33.4	0.97	0	0	0	0	0	0
	37.1	0.02	0.99	0	0	0.10	0	0
	40.0	0	0	0.99	0	0	0	0
	44.9	0	0	0	0.98	0	0	0
	49.7	0.01	0	0	0.03	0.01	0.93	0
	52.4	0	0.07	0	0	0.97	0.02	0
	55.9	0	0	0	0	0.02	0	0.96

Table 6. FRAC results for the adjusted FE model of the vehicle body structure.

Impact Point/Dir		FRAC	Impact Point/Dir		FRAC	Impact Point/Dir		FRAC
REFERENCE FE MODEL RESPONSE POINT: X-DIR	front left/x	0.9560	REFERENCE FE MODEL RESPONSE POINT: Y-DIR	front left/x	0.2315	REFERENCE FE MODEL RESPONSE POINT: Z-DIR	front left/x	0.8323
	front left/y	0.1859		front left/y	0.1795		front left/y	0.1909
	front left/z	0.4848		front left/z	0.3990		front left/z	0.7250
	front right/x	0.9846		front right/x	0.1181		front right/x	0.5154
	front right/y	0.1774		front right/y	0.1946		front right/y	0.2316
	front right/z	0.4986		front right/z	0.3854		front right/z	0.4522
	rear left/x	0.9647		rear left/x	0.3333		rear left/x	0.2151
	rear left/y	0.8627		rear left/y	0.3187		rear left/y	0.3266
	rear left/z	0.8263		rear left/z	0.3838		rear left/z	0.3873
	rear right/x	0.9657		rear right/x	0.2916		rear right/x	0.3961
	rear right/y	0.9432		rear right/y	0.3388		rear right/y	0.2362
	rear right/z	0.6384		rear right/z	0.3813		rear right/z	0.5864
Average x-dir		0.7082	Average y-dir		0.2923	Average z-dir		0.4246
-	-	-	-	-	-	Overall average		0.4751

3.1.2. Eigenfrequencies

Table 4 shows the eigenfrequencies of these same mode shapes in both the reference and adjusted FE models of the vehicle body structure, together with those of the bench test.

3.1.3. MAC Matrices

Table 5 gathers the MAC of the adjusted FE model of the vehicle body structure contrasted with the bench test. Despite the switch mentioned above between the fifth and sixth eigenmodes, the MAC coefficients are still higher than 0.9. Thus, the modal results also have consistent correspondence with the bench test data. The main diagonal terms on the MAC matrix of the adjusted FE model of the vehicle body structure are generally better (higher) than for the reference FE model, thus indicating an improved accuracy for the adjusted model.

3.1.4. FRAC Coefficients

From the FRAC coefficients shown in Table 6, we computed the overall average FRAC for the adjusted FE model of the vehicle body structure, $FRAC_{adj} = 0.4751$. It is 9.1% higher than the overall average FRAC for the reference FE model of the vehicle body structure, $FRAC_{ref} = 0.4355$ (Table 3). This confirms the correlation improvement given by the proposed adjustment, also noticed by comparison of the MAC and the FRF plots.

3.1.5. Frequency Response Functions (FRFs)

Figure 11 shows the root mean square (RMS) of the FRFs of the four excitation points for the bench test, the reference FE model of the vehicle body structure and the adjusted FE model of the vehicle body structure.

From these results, and based on the visual analysis of the plots, we found that the curves for the adjusted FE model generally fit better with the curves for the bench test than those for the reference FE model.

3.2. Optimization Analysis

We performed a weight optimization analysis of the vehicle body structure to evaluate the isolated effect of the proposed methodology. In this way, we could quantify the enhancement provided in terms of weight and cost savings. We considered the following restrictions to perform the optimization analysis:

- Disregard panels, since they were already set to the minimum gauge allowed;
- Disregard component brackets, since their main function is related to component tests;
- Avoid gauge changes smaller than 0.1 mm due to manufacturing restrictions;
- Avoid gauge changes higher than 15% of the initial gauge (if $t < 1.0$ mm) due to manufacturing restrictions for new tooling;
- Avoid gauge changes higher than 10% of the initial gauge (if $t \geq 1.0$ mm) due to manufacturing restrictions for new tooling;
- Disregard critical parts related to frontal and lateral impact tests, since their main function is related to crash analysis; and
- Disregard critical parts related to chassis and powertrain attachments, since their main function is related to fatigue durability analysis.

The optimization analysis provided a gauge reduction in four parts of the vehicle body structure (Figure 12), resulting in a weight saving of 0.566 Kg. As expected, the FRF peaks of the optimized model moved in the direction of the bench test curves (Figure 13) but without exceeding their peaks.

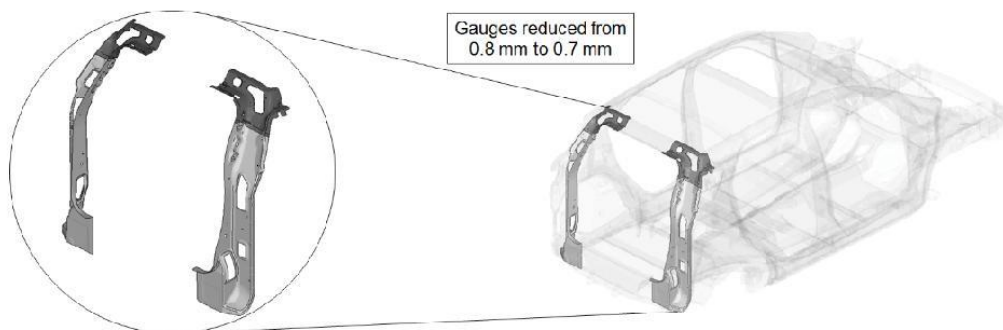


Figure 12. Optimized adjusted FE model of the vehicle body structure highlighting the parts with gauge reduction.

In addition to this, we also analysed the stresses on the down-gauged components (0.7 mm thickness on the optimized adjusted FE model). We compared them with the stresses on the same components with the original gauge (0.8 mm thickness on the adjusted FE model). To this end, we performed a linear static calculation using an input torque of 3000 N.m applied in the front shock towers by opposite vertical loads. Based on the vehicle manufacturer's experience in durability simulation test analysis, we partially constrained the rear shock towers to simulate a critical road test condition. We measured the maximum stress in the components considering the original and down-gauged conditions (Figure 14). Then we calculated the predicted lifetime [28] for each condition using the Neuber plasticity correction and the Smith–Topper–Watson mean stress correction, considering the properties of the SAE-1020 material. As a result, the original lifetime decreased from $5.681 \cdot 10^6$ to $4.120 \cdot 10^6$ cycles, but still complies with the material endurance limit [29].

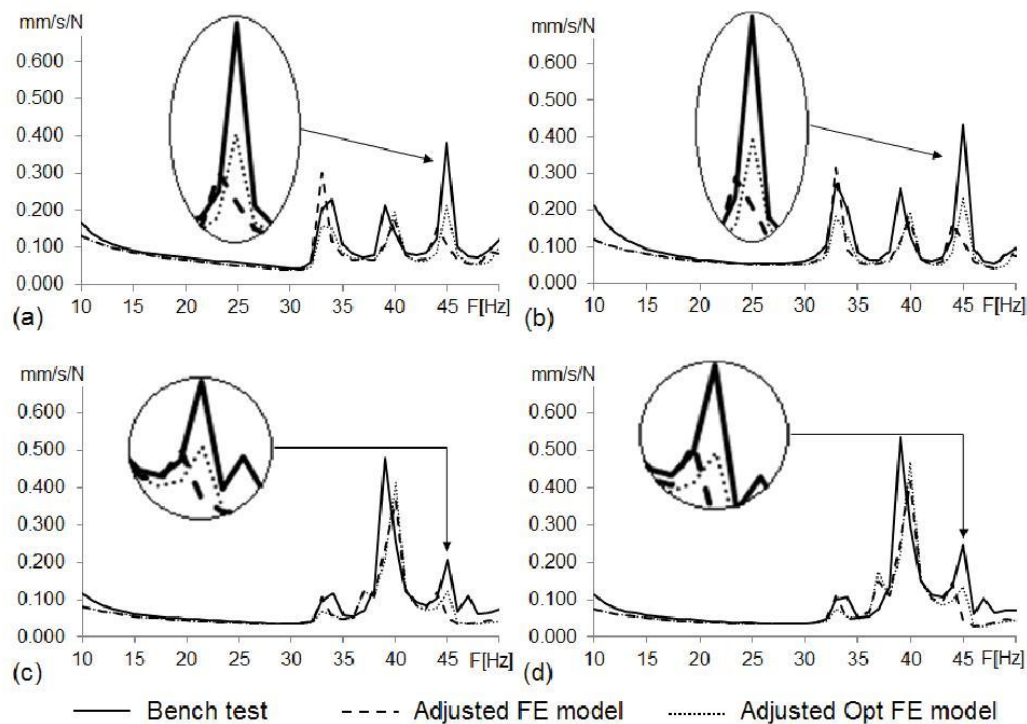


Figure 13. RMS curves from the body structure bench test, adjusted and optimized FE models FRFs (driver seat attachment vibration responses to unitary excitation). (a) Load at front left shock tower; (b) at front right shock tower; (c) at rear left shock tower; (d) at rear right shock tower.

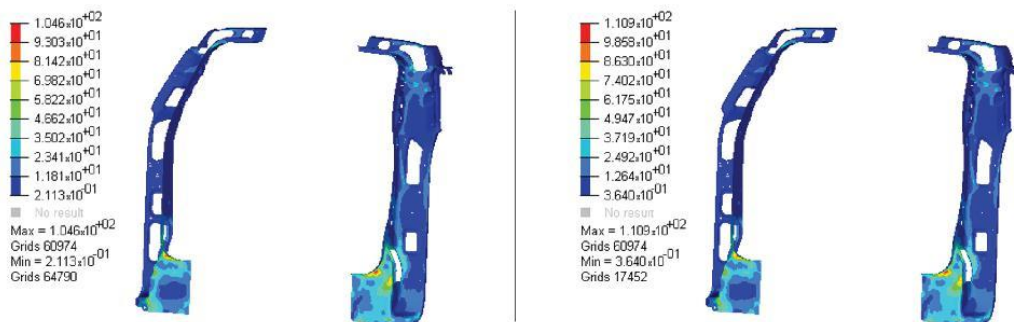


Figure 14. Maximum stress under torsional critical load on the adjusted FE model (left) and the adjusted optimized FE model (right).

The optimization analysis confirmed the improvement of the MAC and FRAC coefficients given by the adjusted model, thus corroborating the significant improvement of the correlation level provided by the proposed adjustment methodology.

To better illustrate the impact of the modelling approach proposed here, we also performed a weight-saving estimate as a result of this action, based on the following parameters:

- Sales in 2019 of the tested vehicle [30–32]: 282,371 units; and
- Weight savings per vehicle: 0.566 kg

As a result, we estimated material savings of 160 tons of steel per year for this vehicle model only. Considering that the life cycle of a vehicle can extend for approximately ten years [3] and the great diversity of vehicle models on the market, the total savings can reach very impactful numbers not only in economic terms, but also in environmental and social issues.

In this case, the reduction in the use of steel, which brings about important cost savings, also helps preserve natural resources. In addition, the weight reduction provided by this optimization improves the vehicle's fuel economy performance, bringing another positive impact on the preservation of natural resources and the reduction of carbon emissions.

4. Discussion

Improving the correlation level between FE simulations and field tests is a key factor in improving the time- and cost-efficiency of many engineering projects. In particular, it is essential in the automotive industry. A possible solution to achieve this goal would be to add nonlinear elements and boundary conditions to traditional linear FE simulation models.

However, this change from linear to nonlinear conditions would substantially increase the computational time and resources required to perform an analysis. Therefore, when large structures are involved, such as in the case of vibration analysis of the vehicle body structure, the use of nonlinear models becomes an unfeasible alternative.

In this work, we propose a new methodology to increase the accuracy of linear models without compromising their complexity and feasibility. For this, we compared the MAC coefficients, the FRF curves and the FRAC coefficients obtained for both the reference and the adjusted FE models of the vehicle body structure, getting the following findings:

- Both the reference and the adjusted models have a consistent correspondence with respect to the bench test data ($MAC > 0.9$). The adjusted model shows slightly better MAC correspondence than the reference model;
- The FRF curve of the adjusted model fits better with the bench test results than the FRF curve of the reference model, especially around the 33 and 39 Hz peaks; and
- The overall average FRAC coefficient of the adjusted model ($FRAC_{adj} = 0.4751$) showed an improvement of 9.1% compared to the overall average FRAC coefficient of the reference model ($FRAC_{ref} = 0.4355$).

In addition, the optimization performed based on the results of the adjusted FE model generated a significant improvement in the weight of the vehicle body structure weight.

The analysis of these results confirmed the effectiveness of the proposed methodology. It fills the knowledge gap, initially identified, through a novel process for adjusting the stiffness and damping of spot-welded joints on thin-wall structures using small FE test models and including them into linear FE models of the entire vehicle body structure.

However, considering the results obtained, the following research opportunities to expand the correlation improvements achieved here can be further exploited:

- Improve the accuracy of the mesh representation of welding spots by proposing new element formulations;
- Improve the accuracy of the mesh representation of bolted connections by proposing new element formulations;
- Improve the accuracy of the mesh representation of adhesive connection joints by proposing new element formulations;
- Refine the representation of the FE model by incorporating variations in sheet metal thickness from stamping manufacturing processes;

- Improve the accuracy of experimental bench test results by proposing new acquisition strategies for raw data measurements; and
- Improve the accuracy of experimental bench test results by proposing new data processing methods for both vibration and damping evaluation.

These considerations, among others, will be introduced in future research models applied to vehicle structures for their refinement, either for vibration and noise analysis or even for durability and fatigue studies, aiming to increase the accuracy of their simulations and consequently improve the correlation level between the analytical and test results.

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