An investigation into NVC characteristics of vehicle behaviour using modal analysis

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Abstract. NVC characterizations of vehicle behavior is one essential part of the development targets in automotive industries. Therefore understanding dynamic behavior of each structural part of the vehicle is a major requirement in improving the NVC characteristics of a vehicle. The main focus of this research is to investigate structural dynamic behavior of a passenger car using modal analysis part by part technique and apply this method to derive the interior noise sources. In the first part of this work computational modal analysis part by part tests were carried out to identify the dynamic parameters of the passenger car. Finite elements models of the different parts of the car are constructed using VPG 3.2 software. Ls-Dyna pre and post processing was used to identify and analyze the dynamic behavior of each car components panels. These tests had successfully produced natural frequencies and their associated mode shapes of such panels like trunk, hood, roof and door panels. In the second part of this research, experimental modal analysis part by part is performed on the selected car panels to extract modal parameters namely frequencies and mode shapes. The study establishes the step-by-step procedures to carry out experimental modal analysis on the car structures, using single input excitation and multi-output responses (SIMO) technique. To ensure the validity of the results obtained by the previous method an inverse method was done by fixing the response and moving the excitation and the results found were absolutely the same. Finally, comparison between results obtained from both analyses showed good similarity in both frequencies and mode shapes. Conclusion drawn from this part of study was that modal analysis part-by-part can be strongly used to establish the dynamic characteristics of the whole car. Furthermore, the developed method is also can be used to show the relationship between structural vibration of the car panels and the passengers’ noise comfort inside the cabin.

1. Introduction
The size of requirement for industrial vehicles excelling in comfort and reliability has become very large for the last few decades. Due to the demands of high speed operation and the use of light weight and flexible structures in modern machinery, static measurements of stress/strain properties are not sufficient. One of the main areas that should be considered when investigating a vehicle is its NVC characteristics. This leads to the improvement of its vibro-acoustic quality as we know that structural vibration has a strong relationship with the interior noise. Therefore, due to the important of NVC characteristics of vehicles it is become a competitive target for car selling. The more refined the car is, the more selling potential it will be. In improving noise, vibration and comfort (NVC) characteristics of a vehicle, it is a must to understand the dynamic behaviour and characteristics of each structural part of the vehicle. This issue has been lacking in the open scientific literature. The only available information is modal analysis of the body-in-white and chassis but the individual structural components for the whole car still not studied yet. It has been researched and studied for decades how computational and experimental modal analysis methods are used to solve noise and
vibration problems in automotive and also the importance of the combination between them in order to provide a comprehensive understanding of the dynamic characteristics of vehicles prior and after manufacturing which is leading to improve their noise, vibration and comfort. This method has been used successfully for simple structure. However, for complex structure like cars still under investigation, because of no good linearity between the excitation force and the response force which for sure affect the quality of the results. Therefore, to solve this problem we have developed a new method. This method is based on modal analysis part by part and was validated by a comparison between experimental and computational modal analysis. It helps us to identify easily the individual structural component of the vehicle that can be radiate high vibration or contribute to the interior noise and then getting control of it. In this work, we are addressing this point by detailed study for a specific passenger car model as a case study using modal analysis part by part.

Knowing dynamic characteristics of a vehicle components such as natural frequencies, mode shapes and damping ratios are essential for vehicle comfort, structural safety evaluation, longer fatigue life, and structural health monitoring. By knowing the dynamic behaviour of a structure under its natural frequencies, we could determine the location of damaged parts of the structure before it starts causing losses of life, because damaged structure tends to exhibit excessive deformation under the same magnitude of excitation. Also by knowing the dynamic characteristics of a structure, engineers can design better structures.

Weight, stiffness and damping are the important factors that have a strong effect on the natural frequencies and mode shapes of a structure, which they can determine where these natural frequencies and mode shapes will exist. For that reason, we have to look into these frequencies and know how they affect the vibration behavior of the structure when a force excited it. This can help engineers to design good structures with improved vibration. Therefore, most of the developments of the structural vibration analysis tools in the new era of engineering came about due to the need of solving vibration problems. Modal analysis in both methods (experimental and computational) is one of the useful tools that provide an understanding of structural dynamic characteristics, operating condition and performance criteria that makes design of optimal dynamic behavior or solving structural dynamics problems in the existing designs.

2. Computational modal analysis in automotive

By the early 70’s, Finite Element Analysis (FEA) was limited to expensive mainframe computers generally owned by the aeronautics, automotive, defence, and nuclear industries. Since the rapid decline in the cost of computers and the phenomenal increase in computing power, FEA has been developed to an incredible precision. Present day supercomputers are now able to produce accurate results for all kinds of parameters. There is a lot of researches conducted in the last decade provide a comprehensive source of information. This information can be used as a guideline to make this work more successful. The following are samples of researches performed on structural dynamic analysis of vehicles using computational modal analysis method.

A computational modal analysis was used to investigate the dynamic characteristic of a truck chassis design to determine the relationship between road excitations and the vibration and strength of the truck chassis. The responses of the truck chassis which include stress distribution and displacement under various loading condition were also investigated. There are two types of structural analysis carried out; normal mode analysis to determine the natural frequencies and mode shapes, and the linear static stress analysis to look into the stress distribution and deformation pattern of the chassis under static load. Different boundary conditions were used for each analysis. It was found that the road excitation is the main disturbance to the truck chassis as the chassis natural frequencies lie within the road excitation frequency range. Furthermore, the determination of the suitable mounting locations of components such as engine and suspension system was possible from the mode shape (Teo & Roslan 2007). Another work was also carried out by (Kassahun 2008) on static and dynamic analysis of a commercial vehicle structure (van body). The method used was finite element modelling and analysis for which the inputs were obtained from quarter car model analysis. The dynamic analysis includes
only modal and random response analysis of the structures. In the dynamic analysis, damping properties of the structure were not considered, only the damping due to shock absorber of the vehicle was included in the analysis. The analysis was carried out for the main load bearing structure of the vehicle, i.e., chassis side frames and cross members, steel frames of the van body structure, and cab. The responses to static loads and random excitation caused by road roughness were determined. Components and, particularly areas which are much affected by the different loads were identified. A further method was reported and validated by (Abrams 2008) using an engine as a structural member of a formula of Society of Automotive Engineers (SAE) race car chassis. A modal analysis was used with a beam mesh of the chassis to provide relative comparison; with and without structural support from an engine. It was also used to validate the assumption that chassis stiffness would increase an increase of chassis stiffness. Results confirmed that providing structural support at engine mounting points of the chassis increased the stiffness and changed the first elastic mode of vibration from bending to torsion. Additionally, a static analysis was used to measure chassis deflection, and established a decrease in deflection when structural support was provided at the engine mounting points of the chassis. This report’s conclusion indicated that using the engine as structural member significantly improves the stiffness of the chassis.

In a different study (Zhanwang & Chen 2000), finite element model for Mini-car Changan Star 6350 using SDRC/I-DEAS was developed. The model was imported to MSC. Patran to generate the mesh and the dynamic analysis of the model was accomplished using MSC.Nastran. The analysis included modal analysis of the load bearing structures, the chassis and the body structure frame, in which the first 100 natural frequencies were determined in torsion and bending modes, and transient dynamic analysis of the structures due to road loads that were introduced in the form of acceleration excitations. Results show good consistency with experimental results, therefore, ability of Mini car to meet dynamic design requirements was confirmed. The implementation of FE methods for dynamic analysis of heavy trucks was presented by (Johansson & Gustavsson 2000). The proposed approach involves the development of FE model of the vehicle and its subsystems, i.e. frame, superstructure, engine, cab, axles and chassis suspensions, and steering system. There was a focus on the analysis of the road, wheel, brake, and driveline induced vibrations, static load cases and fatigue life prediction. An example was presented which shows the lateral and vertical acceleration of the cab structure due to periodic excitation of axle due to wheel run out. It was concluded that, FE-based complete vehicle analysis has the advantage to cover broad range of analyses with one type of model.

An optimization of light weight bus side structures was reported (Lan et al. 2004). The optimization targeted the reduction of the number as well as weight of different parts in the whole body structure. For this purpose, two scenarios, with and without supporting structures between longitudinal waist beams of the side frame were considered. The FE-model for the structure without the supporting structure was developed using the computer package UG and mesh generated in MSC.Patran, while the FE Analysis was completed using ANSYS. All the results obtained were compared with experimental results. The study was stated that the structure with support enables significant alleviation of body weight and satisfactory performance requirements were achieved. Furthermore, a comparative study was conducted between analytical and numerical methods for modelling automotive dissipative silencers with mean flow (Kirby 2009). The results presented in this study demonstrate excellent agreement between analytic and numerical models provided a sufficient number of propagating acoustic modes were retained. However, the numerical mode matching method was shown to be the fastest method. Moreover, the hybrid finite element method was demonstrated to be as fast as the analytic technique. The possible causes of the failure of an exhaust system for heavy duty commercial trucks and the possible solutions were examined (Vascncell0se et al. 2003). The researchers describe the use of computer simulation supported by experimental data on the design process of the exhaust system, with regard to structural behaviour. A finite element model was generated including the complete vehicle and the exhaust system. Static and dynamic analyses were performed in MSC.Nastran software simulating different loading conditions and system configurations. The results obtained assured the structural integrity of the modified exhaust system.
when implemented on the vehicle and also contribute to a better understanding of this system behaviour and its structural strength. Generally, numerical models validated with experimental data are a powerful tool during the development phases of a new vehicle, reducing project time and costs.

A numerical modelling approach for investigating the onset of squeal in a drum brake system was proposed few years back (Jinchun et al. 2006). The brake system model was based on the modal information extracted from finite element models for individual brake components. The component models of drum and shoes were coupled by the shoe lining material which was modelled as springs located at the centroids of discretized drum and shoe interface elements. The vibrational characteristics of the developed multi degree of freedom coupled brake system model were determined by a complex eigen-value analysis. The study showed that both the frequency separation between two system modes due to static coupling and their associated mode shapes played an important role in mode merging. In another research work (Jae-Yeol Park & Singh 2010). The effect of spectrally varying mount properties (including stiffness and damping) was examined analytically on the dynamics of power train motions under torque excitation. New analytical formulations were verified by comparing the frequency responses with numerical results obtained by the direct inversion method (based on Voigt type mount model and finally, an axiom for torque roll axis decoupling was provided by employing direct and adjoint eigenvalue problems.

Finally, an interesting approach for a Reduced Beam and Joint Modelling approach to optimize and analyse global vehicle body dynamics, specifically bending and torsion modes was developed currently (Donders et al. 2009). This is a concept of Computer-Aided Engineering (CAE) modelling approach based on predecessor FE models, which can be used to improve the fundamental body noise, vibration and harshness behaviour (global modes) by means of beam and joint modifications. Modifications in the body beam-like sections and in the joints were analysed using body reduced modal model. This small-sized model was used to quickly and accurately optimize the low-frequency vehicle performance. The work conclude that static Beam Force Analysis strategy can be used to provide complementary insight in the vehicle body design in an early design stage, which can be employed to improve the static body performance.

As can be seen the above researches focus on the finite element analysis of vehicles and their different components, i.e. engine, steering system, superstructure, cab and axles and chassis suspensions. Different computer packages such as ANSYS, ADAMS, DADS, MSC.patran/MSC.Natran and IDEAS were used to develop the virtual models of vehicles and their components in order to perform the required analysis the method still recognized as an acceptable tool for designing simple structural design. However, the confidence in the results for complex structures still questionable because of several simplifications in the model. Therefore, certainty in the computational results most often depends on the experience of the analyst who builds the finite element model.

3. Experimental Modal analysis in automotive

One common reason for experimental modal analysis is the verification or correction of the results of computational modal analysis approach. Below are some reviews on the application of experimental modal analysis in automotive.

In a case study work, (Verboven et al. 2006) presented a new developments for the nonparametric processing of modal test data. The application of the presented methods was studied for the case of a typical modal test of body-in white structure. An application of multi sine excitation technique permits improvement of the data quality, as well as the detection, qualification and quantification of nonlinear distortions during Frequency Response Function (FRF) measurements. Also, the study suggested that, to make the presented techniques available for multi-input modal testing, attention is paid to the design of optimal multi-input excitations by maximizing the Fisher information matrix as well as minimizing the crest factor of the applied excitation.

An investigation of the applicability of a non-parametric exponential time-window in the pre-processing of data acquired with arbitrary excitation to improve the modal parameters of an engine
A successful implementation of modal control to arbitrary curved panels using experimentally evaluated mode shapes was demonstrated (Hurlebaus et al. 2008). The identification of the modal control parameters from a measured data base was of interest. The test object was a modern roaster car body. The floor panel has been selected for the experimental setup. Since, for such structures, the evaluation of modal parameters from numerical calculations of local modes is complicated because the results strongly depend on proper boundary conditions of the truncated structure. Thus, the modal data was identified using experimental modal analysis. The modal controller was implemented on a digital controller board, and experimental tests with the floor panel and centre panel of a car body were carried out to validate the proposed concept. The work ended to a reduction of structural vibrations, which also leads to a reduction in acoustic radiation.

According to the previous research studies, much work has been done on vehicle components such as engine, exhaust system, cabin structure, chassis suspension, bearing structure, power train system and drum brake system using either experimental modal analysis or computational modal analysis or both. In addition there is also some work has been done on the whole car but with using the experimental modal analysis only or computational modal analysis only. The researchers discussed how computational and experimental modal analysis methods are used to solve noise and vibration problems and came out with the importance of the combination between them in order to provide a comprehensive understanding of the dynamic characteristics of vehicles prior and after manufacturing which leading to improve the noise and vibration comfort (NVC). In this study we develop an accurate method to analyse the dynamic characteristics of whole car using modal analysis part by part in order to solve the noise and vibration problems of a passenger car in easy way and less errors. The confidence in the results for complex structures still questionable because of the several simplifications and assumptions applied on the structure. The main assumption involved in modal analysis is that the structural system is linear, i.e., structural displacements are directly proportional to applied loads.

4. Methodology
The main objective of this research is to implement an accurate and reliable method to investigate the dynamic characteristics of a complex structure, hence assures easier and faster implementation. In the first part of this work computational modal analysis part-by-part is utilized to determine and analyse the dynamic characteristics of a specific vehicle structure. The second part carried out experiment modal analysis part by part of the same vehicle. The method was validated by a comparison between experimental and computational modal analysis results. Modal analysis was utilized as a tool to find the dynamic properties of the passenger car namely natural frequency, mode shape and damping. In this paper only few vehicle panels are considered namely, trunk, hood, roof and door panels.

4.1. Computational Modal Analysis part by part
Computational modal analysis part by part was conducted using FEA technique. It is a computer simulation technique used in engineering analysis. In general, there are three phases in any computer-aided engineering task are pre-processing, analysis Solver and Post-Processing. The simulation results of any Finite element analysis depend on many variables. The most important ones are:

a) CAD Model,  
b) Boundary Conditions and Initial Loading  
c) Assumptions made
The Model used for this investigation is Dodge Neon car from NCAC (National Crash Analysis Centre).

The car model used for this study was designed for crash test and has several other parameters attached to it that is not related to modal analysis. Therefore we have to remove unwanted parts and components that pose a problem to the modal analysis and make the problem complicated. Two softwares have been used in this investigation namely VPG 3.2 and Ls-Dyna. VPG 3.2 software is very capable software with a lot of libraries that contain vehicle related car panels and Ls-Dyna has been used as its processor and solver. The car panels (trunk, hood, roof and door) were extracted from the original FEA model of the whole car then performed modal analysis test. Modal analysis is performed in two stages, firstly the modes are computed and then output to a binary database, secondly this database been used as input to supply modes for the transient dynamic analysis. The modes are computed using implicit analysis.

4.1.1. Results and discussion

The results of computational modal analysis are presented in table1, table2 and table3. The first mode of vibration occurs at 48.63 Hz, the horizontal part appears to be not vibrating and the vertical part looks like flying freely with high amplitude at the bottom edge. The second mode of vibration occurs at 81.13Hz, the vertical part of the trunk appears to be oscillating in a second bending mode as we can see clearly that the panel exhibit two nodes and three antinodes, it also shows significant vibration at the two corners of the horizontal part of the trunk. At the third mode the panel shows third bending mode at both vertical and horizontal parts of the panel with high amplitude.

The fourth mode was at 105.43 Hz, both parts of the trunk showing the third bending mode with significant amplitude at the horizontal part. In the fifth mode both parts of the panel showing first bending mode. The sixth mode was at 133.81 shows significant vibration at the horizontal part.

Roof panel showed elastic bending for all modes. First and second modes are closed to each other (75.4 Hz and 76.4 Hz) are showing same second bending mode at the rear part of the roof. Most probably the rear beam is the main contributor to the second bending mode. The third and fourth modes have similar behaviour and also their frequencies are close (97.81 Hz and 100.5 Hz). They show first bending mode at the left and right sides of the roof. The fifth mode occurs at 102.9 Hz. At this mode the roof shows a third bending mode at the rear frame of the roof with high amplitude. The last mode was at natural frequency 104.5 Hz. It shows second bending mode at the front of the roof.

The hood panel showed the first mode at 58.83 Hz, it is a torsional mode, each two opposite corners are vibrating in phase. At the second mode the hood panel shows second bending mode, and occurs at 76.08 Hz. The third mode occurs at 86.8 Hz, it appears to be a second bending mode. There are two antinodes, one on the right side and the other one on the left side. The fourth mode of vibration occurs at 107.78 Hz. It shows high vibration on the left and right sides, and looks like flying mode. The fifth mode occurs at 117.6 Hz, and appears to be a third vertical bending mode. The last mode of vibration occurs at 125.7 Hz and only the left side of the hood has significant vibrating.

The fist mode of the door panel exhibits first bending mode at the bottom half of the door panel at 55.71 Hz. The second mode occur at 74.01 Hz. It exhibits second bending mode at the bottom part of the door panel with low amplitude. At the third mode the door panel does not show significant mode. The fourth mode occurs at 94.88 Hz. It shows first bending mode at the centre of the door. This mode is resulted from the vibration of horizontal crossing beam as shown in the figure. At the fifth mode, the door does not show significant bending mode. The last mode was at 105.45 Hz. At this mode the door panel shows second bending mode at the bottom half.
Table 1: Comparison between computational and experimental mode shapes for Trunk and Roof panels.

| Mode | Trunk panel |  | Roof Panel |
|------|-------------|---|------------|
|      | Computational | Experimental | Computational | Experimental |
| 1    | ![Image](image1.png) | ![Image](image2.png) | ![Image](image3.png) | ![Image](image4.png) |
| 2    | ![Image](image5.png) | ![Image](image6.png) | ![Image](image7.png) | ![Image](image8.png) |
| 3    | ![Image](image9.png) | ![Image](image10.png) | ![Image](image11.png) | ![Image](image12.png) |
| 4    | ![Image](image13.png) | ![Image](image14.png) | ![Image](image15.png) | ![Image](image16.png) |
| 5    | ![Image](image17.png) | ![Image](image18.png) | ![Image](image19.png) | ![Image](image20.png) |
Table 2: Comparison between computational and experimental mode shapes for Hood and Door panels.

| Mode | Hood panel | Door Panel |
|------|------------|------------|
|      | Computational | Experimental | Computational | Experimental |
| 1    | ![Hood Panel Mode 1 Computational](image1) | ![Hood Panel Mode 1 Experimental](image2) | ![Door Panel Mode 1 Computational](image3) | ![Door Panel Mode 1 Experimental](image4) |
| 2    | ![Hood Panel Mode 2 Computational](image5) | ![Hood Panel Mode 2 Experimental](image6) | ![Door Panel Mode 2 Computational](image7) | ![Door Panel Mode 2 Experimental](image8) |
| 3    | ![Hood Panel Mode 3 Computational](image9) | ![Hood Panel Mode 3 Experimental](image10) | ![Door Panel Mode 3 Computational](image11) | ![Door Panel Mode 3 Experimental](image12) |
| 4    | ![Hood Panel Mode 4 Computational](image13) | ![Hood Panel Mode 4 Experimental](image14) | ![Door Panel Mode 4 Computational](image15) | ![Door Panel Mode 4 Experimental](image16) |
Table 3: Comparison between experimental and computational frequencies of panels (in Hz).

| PANEL | MODE | 1    | 2    | 3    | 4    | 5    | 6    |
|-------|------|------|------|------|------|------|------|
| Trunk | Exp. Freq | 42.2 | 86.5 | 94.7 | 110  | 120  | 139  |
|       | Comp. Freq | 48.63| 81.13| 95.4 | 105.43| 120.68| 133.81|
| Roof  | Exp. Freq | 60.2 | 89.8 | 105  | 129  | 148  | 163  |
|       | Comp. Freq | 75.4 | 76.4 | 97.8 | 100.5| 102.93| 104.56|
| Hood  | Exp. Freq | 60.4 | 76.2 | 88.8 | 105  | 121  | 142  |
|       | Comp. Freq | 58.83| 76  | 86.8 | 107.78| 117.59| 125.68|
| Door  | Exp. Freq | 54.2 | 72.7 | 76.1 | 100  | 114  | 121  |
|       | Comp. Freq | 55.7 | 74  | 86.5 | 94.88| 99.37| 105.45|

4.2. Experimental Modal Analysis part by part and comparison with computational Modal Analysis

Experimental modal analysis was performed to determine a complete dynamic description of each car component (trunk, hood, roof and door). An impact hammer was used to excite the structure at a fixed point and six accelerometers were moved over each point on the structure. The force and accelerometers signals were fed into two three channels analysers to produce FRFs. Data was collected in B&K Pulse Labshop EMA software then transferred to ME’scope VES software for further analysis.

The quality assessment of the measurement of the frequency response functions is very important, because this leads to the quality of the modal parameters. Therefore some checks are important to ensure that the quality of the measured frequency response functions is sufficient to evaluate the dynamic parameters of the structure. The first check is the coherence function, it is a frequency dependent indicator that shows which part of the output signal is coming from the real input signal and which part is coming from the additional measurement noise. If the value of the coherence function is near one then a strong linearity exists between the input and output and the influence of the noise is negligible. If the value of the coherence function is far away from one then no linearity exist between
the input and output and the spectrum is dominated by the measurement noise. Figure1 shows the coherence measured in the experiment.

![Figure 1. Coherence function](image)

The second check was made by evaluating the reciprocity and reproducibility of several measured frequency response functions. Theoretically the reciprocity and reproducibility should be the same for all measured FRFs. The technique is to measure frequency response functions of a couple of points on the same structure for example front car door panel. First we excite point 5 and take the response at point 6 and vice versa, then excite point 3 and take the response at point 8 and vice versa. The results showed that the frequency response functions taken from the first couple of points are the same as shown in Figures 2c and 2d. The second couple of points also shows the same frequency response functions Figures 2a and 2b. Therefore, we conclude that the reciprocity and reproducibility of the measured frequency response functions is reasonably good.

![Figure 2. Frequency response function](image)
5. General discussion

Comparison of computational frequencies and mode shapes with those obtained from the experimental modal analysis results indicated quite good agreement at majority modes for Trunk and Hood panels, whereas the Roof and Door panels showed lower agreement as we can see in table 3. This is because of the complexity of Door and Roof panels structures compared to Trunk and Hood panels structures. Moreover the experimental panels and the computational panels indicated some components attached to the experimental panels which were not accounted in the computational panels due the difficulty to include them explicitly in the computational model. These masses affect the overall frequency response function.

Table 3 shows the comparison between experimental and computational frequencies of all panels. Trunk panel shows fair agreement between the two methods except at the first mode as we can see in figure 3a. The roof panel does not show good agreement as the percentage different is quite high, it varies from 7% to 44%. We have also noticed that the last modes showed disagreement as seen in figure 3b. The third comparison was conducted on the Hood panel. Figure 3c shows good agreement at all modes as the percentage is less than 12%. Door panel does not show reasonable agreement especially at modes 3, 5 and 6 as the percentage different is 13%, 14% and 14% respectively. Table 3 and figure 3d.

![Figure 3: Comparison between experimental and computational panel frequencies](image)

Figure 3: Comparison between experimental and computational panel frequencies

For a reasonable comparison, computational and experimental modal analysis require comparison mode shapes as well see table 1 and 2. The first mode shape of the trunk panel does not show any vibration behaviour similarity. This is reasonable because their frequencies are far away from each other (42.2 Hz and 48.63). The second, third and fourth mode shapes are exhibit some similarities in their behaviour. All of them showing vibration in the horizontal and vertical part of the trunk panel. The fifth and sixth modes do not show any significant similarity. Mode shapes of roof panel are shown in table 1. At the first mode both showing vibration at the front and right side of the panel. The
third mode shape shows a fair similarity between the experimental and computational methods as both have significant vibration at both sides of the panel. The fifth mode also shows good similarity in the vibration behaviour at the rear part of the roof panel. The remaining modes do not show any interesting similarity. Hood panel shows reasonable similarity at modes 2, 4, and 5. At mode 2 both methods show significant vibration at the right sight of the panel. Mode 4 showing high vibration on left and right sides for both methods. The other modes do not show significant similarity. For door panel only two modes have similar behaviour, mode 1 and 3, the others do not show significant similarity.

Results from the simulation model using modal analysis part by part are in quite good agreement with the experimental modal analysis results within the investigated panels. Slight discrepancies may be accounted for geometric and material approximations made during the simulation of the FE model. It has been noticed from the comparison results that the vibration amplitude of the experimental mode shapes is significantly less from those of computational modal analysis. This is obvious because of the extra damping treatment applied on the surfaces of the actual car panels to reduce the strength of the noise and vibration. The numerous damping mechanisms at work in a real structure cannot be easily modelled, so damping is not included in most FEA models. However, mode shapes can still be obtained from model without damping.

In conclusion the results obtained from all panels showed a good agreement between experimental and computational modal analysis at the frequency level. However the mode shapes failed to have good agreement. In addition, we have noticed that the computational mode shapes exhibit high amplitude, whereas the experimental mode shapes exhibit low amplitude. Probably this difference occurred because of the damping applied to the surface area of the panels to reduce the vibration amplitude which can affect the excitation and response functions.

6. Conclusion

Generally, we can conclude that modal analysis part by part is a valid method and can be implemented to determine the dynamic characteristics of any complex structure. The results showed fair agreement between experimental and computational modal analysis. However, still some discrepancies between the experimental and computational results. This could be because of: Firstly, boundary conditions: inaccurate boundary conditions can have influence on some or all vibration modes. Therefore, to overcome these problems we keep try other boundary conditions, which could be a further work for this research. Secondy, linearity between input and output noticed from the coherence signals obtained from experimental modal analysis tests for all panels. The results showed that the coherence at low frequencies was far away from one which resulted in large error at the first modes. Thirdly, the correspondence between computational model and the real model as the available FE model was not exactly the same as the real model. This can have great effect on the results and must be considered in future research.

The proposed approach can be considered as an effective tool to determine the dynamic characteristics of vehicle components that can be expected to be the source of the interior noise comfort.

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