Feasibility of Using Four-Post Road Simulator for Modal Analysis of a Truck Frame

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NOMENCLATURE

$\mathrm{H}(\omega)$	Frequency Response Function (FRF)
$\bar{\mathrm{H}}(\omega)$	Pseudo Frequency Response Function (FRF) using pressure/displacement input
$\mathrm{X}(\omega)$	Displacement measurement
$\ddot{\mathrm{X}}(\omega)$	Acceleration measurement
$\mathrm{F}(\omega)$	Force measurement
$\mathrm{P}(\omega)$	Pressure measurement
EMA	Experimental Modal Analysis
OMA	Operational Modal Analysis
PTD	PolyReference Time Domain
MAC	Modal Assurance Criterion
SSI	Stochastic Subspace Identification
RFP	Rational Fraction Polynomial
EMA ground 1	EMA using impact test with structure on the ground (before raising structure on four post road simulator)
EMA ground 2	EMA using impact test with structure on the ground (after raising structure on four post road simulator)
EMA 4 P	EMA using impact test with structure on the four post road simulator
EMA pressure	EMA using pressure as input signal
EMA displacement	EMA using displacement as input signal

ABSTRACT

This paper continues the evaluation of the feasibility of using a four-post road simulator to replicate operating conditions and perform response-only modal analysis on a truck frame. Work presented in this paper is continuation of a previous study that investigated the use of OMA methods for the same structure. This previous study helped in performing preliminary groundwork which included proof of concepts and suitability of existing techniques. Different excitation methods were employed and modal estimates from these tests were successfully validated against estimates from conventional EMA tests. However, the excitation methods employed in previous study were not true operational conditions. An attempt is made in this paper to employ more realistic operational conditions to excite the structure by using a four-post road simulator. Testing the truck frame using four-post road simulator poses several challenges due to factors such as effect of suspension systems, effect of dynamic boundary conditions, etc. which are close to situations encountered in real life. Considering that the excitations are vertical and restricted to the four wheelpan locations in this case, they do not satisfy the general OMA

requirements of spatially well distributed excitations, and might pose challenges in modal parameter estimation. It should also be noted that other excitation conditions such as wind, driveline unbalance and engine vibration are not considered.

In addition to utilizing OMA techniques, two other methods are considered. These methods compute transfer functions between measured outputs and measured input pressure and displacements (from the four-post road simulator), thus providing a function similar to a Frequency Response Function (FRF). These functions differ from conventional FRFs in terms of their numerical characteristics and form, and their applicability for modal parameter estimation is evaluated. Results obtained from the various tests are compared and validated with those from conventional EMA impact tests. From these results, the four-post road simulator is found to be marginally suitable for conducting modal tests involving OMA methods on the automotive structure. The displacement based pseudo FRF method also offer promising results, while shortcomings of the pressure based method are discussed.

1 INTRODUCTION

Four-post road simulators have traditionally been used for applications such as transmissibility studies, durability, buzz, squeak and rattle (BSR) evaluations and non-linear analyses. These tests have been an integral part of vehicle design and performance analysis. Since four-post road simulators have numerous applications, it is of significant interest to assess their feasibility for modal analysis purposes, which is a commonly used method for finding the dynamic characteristics of automotive structures. Generally, modal parameters are estimated using traditional Experimental Modal Analysis (EMA) [2] techniques which require excitation of structure with artificial known forces directly on the structure rather than through the tires and suspension. One shortcoming of this approach is that excitation forces in case of EMA do not represent true operational conditions to which the vehicle is often subjected.

Operational Modal Analysis (OMA) is an emerging field of research which involves estimation of modal parameters based on measured responses only. This technique does not require the measurement of input excitation forces, which are assumed to be random, uncorrelated in nature. The unmeasured excitation forces in the general case of OMA are often the naturally acting ambient forces which are representative of actual operating conditions. Application of OMA to automotive structures [4-5] has been an area of interest over the past few years. Purpose of this paper is to assess the feasability of utilizing four-post road simulators for simulating and collecting response data in order to determine dynamic characteristics of automotive structures using OMA techniques. For ease of writing, the four-post road simulator will be further referred as ‘simulator’ in this paper.

In a previous paper [6], preliminary studies were conducted in this regard, where different methods to excite the structure were studied, for conventional and operational modal analysis. This study showed that even though results from OMA techniques suffer on account of some of the OMA assumptions not being true, they are still usable and mostly coherent with estimates from conventional EMA methods. The excitation scenarios used, however, did not reflect the true operating conditions for the automotive structure (a truck frame with engine and gearbox mounted and supported on the suspension system). This was due to the fact that in this study the structure always rested on the ground. Thus the boundary conditions remained rigid and static throughout the tests. In real life, however, boundary conditions are dynamic due to vehicle-ground interactions. Further, excitations due to this interaction reach the frame and other parts of the vehicle through the tires and after being filtered through the suspension. These factors were not significant during those preliminary tests as boundary conditions were not changing and excitation was directly provided through the frame.

This paper attempts to use a more realistic operational excitation method by using a four-post road simulator to simulate road-induced vibrations and dynamic boundary conditions. The focus of this study is to get more insight into the dynamics of the truck frame when boundary conditions are dynamic and effects of the suspension system are prominent. Excitations from the simulator are primarily in the vertical direction, unlike previous studies where it was possible to excite the structure in lateral and longitudinal directions using an impact hammer. This implies that modes which are predominantly lateral or longitudinal in nature might not be sufficiently excited. It can be argued that this reflects limitations of the approach, but, despite this being true to a certain extent; it also shows that these modes might not be of much significance when the vehicle is actually operating. It is worth noting that, although being closer to operational conditions than the tests conducted in previous study, this scenario is still not completely representing true operational conditions as other operational excitations are still not considered.

In addition to using OMA techniques for system identification purposes, the simulator is also used to collect two more datasets employing displacement and pressure measurements, instead of force, as inputs for computing functions similar to frequency response functions. These functions, being of the form Output/Input, are referred as pseudo FRFs in the paper. While force measurements require the use of external load cells, displacement and pressure measurements can be directly obtained from built-in transducers in the simulator configuration. But it is to be noted that displacement and pressure are measured at the bottom of the column of the simulator and not at the surface of the wheelpan. This might potentially introduce dynamics of the oil column and wheelpan into the measurements performed using the simulator. This condition is studied in detail as part of this paper. Though pseudo FRFs have Output/Input formulation similar to conventional force based FRFs, they have numerical characteristics that differ from conventional FRFs. These factors are expected to pose challenges to modal parameter estimation using traditional algorithms [7] and interpretation of their results.

The paper proceeds with a description of the test setup in Section 2, along with sensor and excitation source information. Section 3 discusses several tests performed on the truck frame, starting with a baseline test using impact hammer, use of the simulator for collecting response time histories for OMA purposes, and measurement of pseudo forms of FRF data with pressure and displacements as input signals. Results from all these tests are discussed and compared in Section 4 of the paper, and a final summary is presented in Section 5. Multiple avenues of research and development possible from this study are briefly presented in Section 6.

2 TEST SETUP

The same structure as used in the previous study [6], consisting of a truck frame with engine and gearbox mounted, is considered for tests conducted in this work. The frame is supported by independent double wishbone suspensions in the front and solid axle leaf springs at the rear. Similar to the previous study, the overall structure is studied based on constituent elements, viz. the frame, the suspensions, the gearbox and the engine. The tires are strapped to the wheelpans of the simulator for all tests, as shown in Figure 1, in order to ensure safety during the test. The simulator mentioned is an MTS 320 four-post road simulator [1] located at the Structural Dynamics Research Laboratory, University of Cincinnati. Figure 1 shows the experimental setup, with the simulator exciting the structure with broadband uncorrelated forcing functions at the four posts.

Figure 1: Test Structure with sensors mounted and strapped to the simulator

50 tri-axial accelerometers are distributed over the test structure in the same spatial configuration as the previous study. Figures 2(a), 2(b) and 2(c) show a few sensor locations on the front double wishbone, the rear leaf spring and the engine respectively. In addition to these, a single uni-axial accelerometer is placed on each wheelpan of the road simulator, adjacent to the tire. This is done in order to make sure that forces imparted through the simulator are indeed random, uncorrelated in nature. These uni-axial sensors also serve the purpose of calculating driving point pseudo FRFS which are desirable from algorithmic point of view while carrying out modal parameter estimation.

(a) **(b)** **(c)**

Figure 2: Front left wishbone, rear left leaf spring, and engine with mounted sensors

3 DATA ACQUISITION AND MODAL PARAMETER ESTIMATION FOR TESTS ON THE SIMULATOR

This section gives details of the various tests conducted on the structure and their corresponding modal parameters. The tests are listed as:

- Conventional FRF based analysis using impact test data
- Pseudo FRF based analysis using simulator excitations with pressure measurements as input signal
- Pseudo-FRF based analysis using simulator excitations with displacement measurements as input signal
- Operational Modal Analysis using response collected through simulator excitations

3.1 Conventional FRF based analysis using Impact Test Data (EMA 4 P).

A conventional impact hammer test is conducted with the test frame mounted on the simulator. This test is carried out to study the effect of change in boundary conditions when the structure is on the simulator instead of being on the ground. Results of this test are to be compared in Section 4.1.1 with the EMA test on ground, which was conducted as a part of studies presented in previous paper [6]. A separate impact test was conducted on the column of the simulator to identify the modes of the simulator itself. It was ascertained that the simulator did not have any modes in the frequency range of interest for this study. However, the fact remains that though each column of the simulator is bolted down to the floor, it might still not be as rigid as a concrete floor. Difference in boundary conditions can also result due to strapping of tires to the wheelpans of the simulator.

The data acquisition parameters for this impact hammer test are listed below:

- Sampling Frequency : 125 Hz
- Frequency Resolution : 0.125 Hz
- RMS averages : 3
- Window : Uniform
- Excitation degrees of freedom : 14
- Response degrees of freedom : 150
- Excitation signal : Force
- Response signal : Acceleration

Figure 3 shows the consistency diagram using the Polyreference Time Domain algorithm [3] for 0 - 30 Hz frequency range.

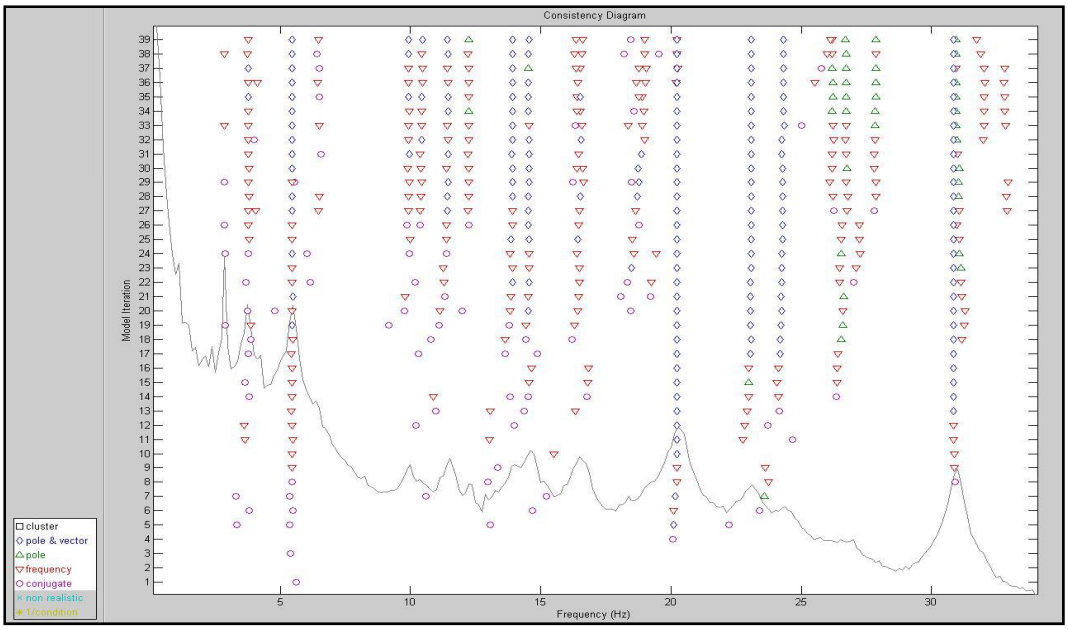

Figure 3: Consistency diagram for an estimate for the EMA impact test on the simulator

A total of eighteen modes are obtained in the frequency range of interest. A high degree of linear independence can be observed from the auto-MAC [2] plot shown below in Figure 4. Modal frequencies and damping estimates are listed in Table 2 in Section 5. Estimates from this test are to be used as a comparison for all tests performed on the simulator. It is important to note that though this test is done with the structure on the simulator, it still has static boundary conditions. This would differ from the forthcoming tests involving simulator excitations due to dynamic boundary conditions caused by motion of wheelpans of the simulator.

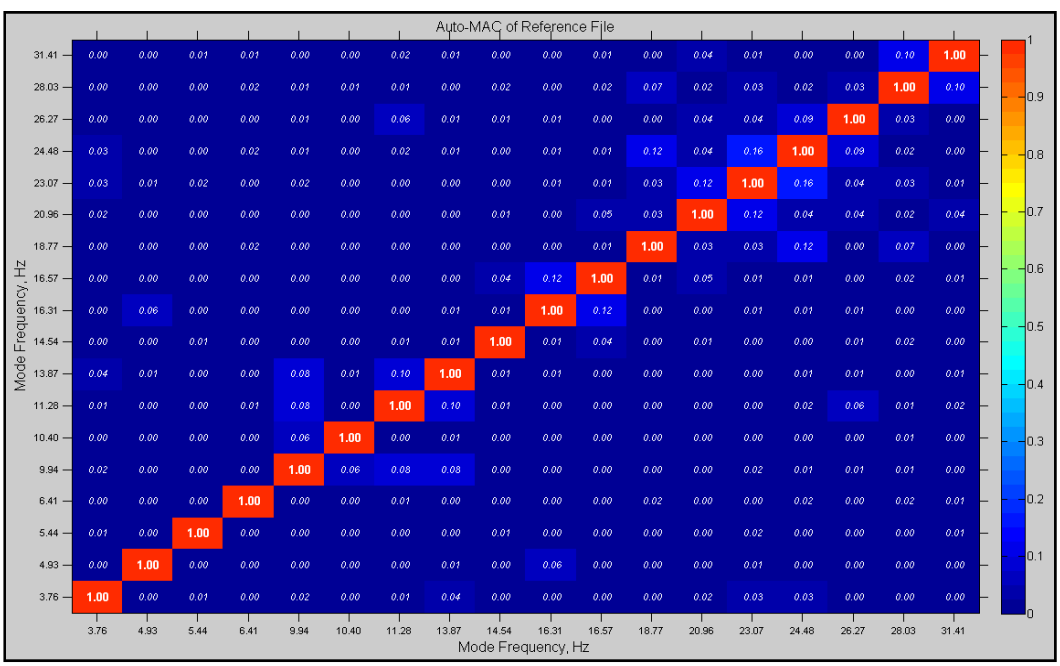

Figure 4: Auto-MAC plot for EMA estimates of impact hammer test on simulator

3.2 Pseudo FRF based analysis using simulator excitations with pressure measurements as input signal (EMA Pressure)

This test is conducted using the simulator for exciting the structure. The test is unique and significant in many ways. One of the main reasons to attempt this measurement is to analyze the possibility of using road simulator facilities available in labs and industries to perform modal test on automotive structures. This would enable obtaining modal frequencies and mode shapes while using the very same facility that is used for various other vehicle development tests. This has been attempted in the past with poor results.

As no load cell is used in this test, it is not possible to measure the force input to the system (load cells can be purchased for each wheelpan at considerable expense). Unlike conventional FRF measurements which measure force as input, this set of FRF-type measurements are obtained using the pressure going to each post of the simulator as input, measured using transducers built into the simulator system. The pseudo FRF is constructed as shown below,

$$\bar{H}_{xp}(\omega) = \frac{\ddot{X}(\omega)}{P(\omega)} \tag{1}$$

where $P(\omega)$ is the pressure entering the column of each cylinder (P_1, P_2, P_3, P_4) of the simulator. It is to be noted that this is not the pressure acting at the surface of the wheelpan but at the bottom of the column where the pressure transducer is located. There is no direct method to compute the forces from the pressure measurements because of the inertial effect caused by the weight of the wheelpan, the moving components of the column of the simulator and the dynamics of the wheelpan. These are the primary reasons for the pseudo FRFs, constructed using pressure measurements, to have numerical characteristics that differ from conventional FRFs.

The following data acquisition parameters have been used for this test:

- Sampling Frequency : 125 Hz
- Frequency Resolution : 0.0625 Hz
- 50 RMS averages with 3 cyclic averages [8]
- Window : Hanning
- Excitation degrees of freedom : 4
- Response degrees of freedom : 154 (150 on the structure, 4 on the wheelpans)
- Excitation signal : Pressure
- Response signal : Acceleration

Figure 5 below shows the consistency diagram for one of the estimates using the PTD algorithm for the EMA pressure dataset. Good estimates, with near complete modal information in terms of frequency, damping and mode shapes, are represented by blue diamonds. It can be seen from the diagram that only a few modes have been estimated consistently across increasing model orders. Multiple estimates using narrow frequency bands, with varying combinations of reference channels, are essential in order to extract maximum possible number of modes of the structure.

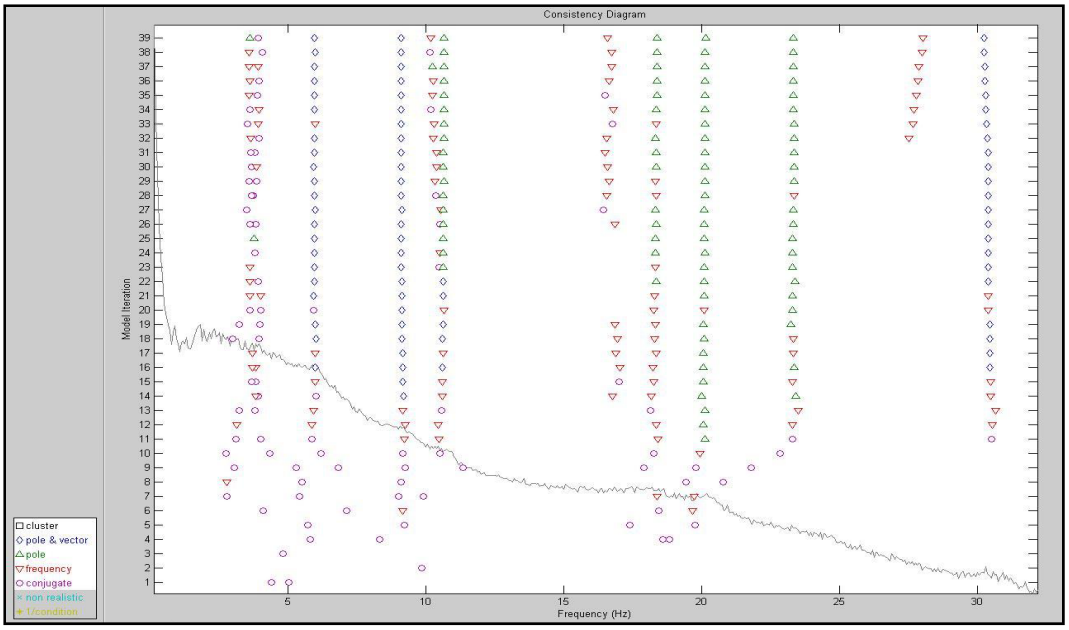

Figure 5: Consistency diagram for an estimate for EMA pressure test

Poor estimates from this method can be explained by studying the driving point pseudo FRF for this test as shown in Figure 6. As mentioned earlier in Section 3.1, the column of the simulator is known to have no modes of its own in the frequency range of interest. Hence a FRF constructed using an input signal from the bottom of the column and output from the top of the wheelpan should ideally resemble a smooth curve with no sharp peaks. This would mean that the input to the structure through the tires is the same as the input being measured at the bottom of the column. In contrast to this, a pseudo FRF curve computed using pressure measurement from the bottom of the simulator column as input and accelerometer response on the top of the wheelpan as output appears to have peaks in the frequency range of interest. These peaks are not representative of the dynamics of the simulator column itself, but are mere inconsistencies involved in the use of pressure signals for computing pseudo FRFs. This is one of the probable reasons for the poor estimation of modal parameters, as can be seen in the consistency diagram in Figure 5.

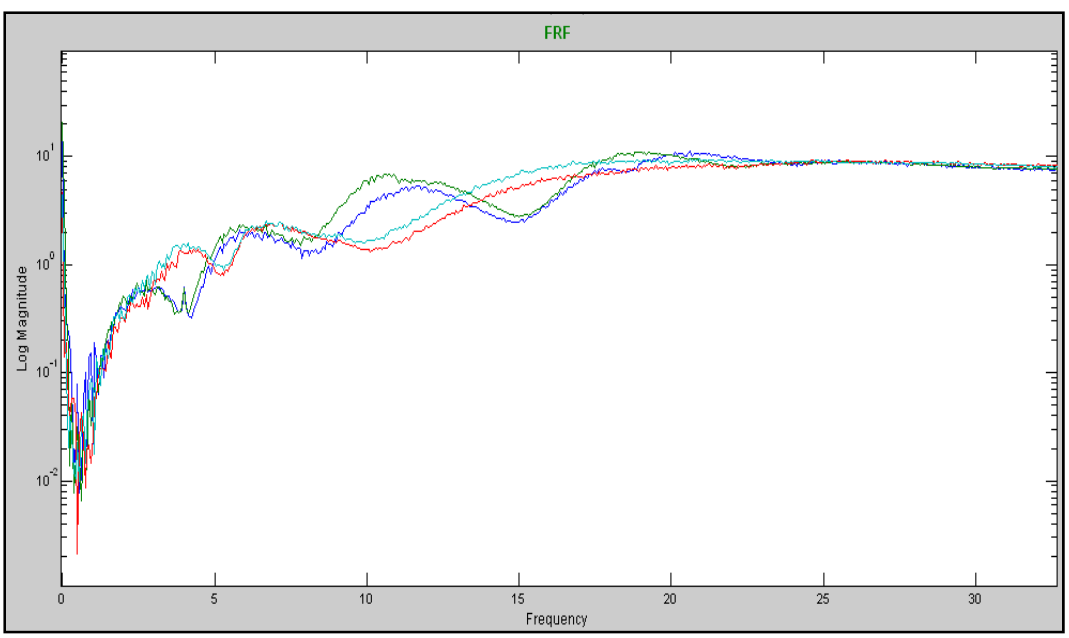

Figure 6: Driving point pseudo FRFs for responses on the wheelpans of the simulator

The auto-MAC plot in Figure 7 shows thirteen modes obtained in the frequency range of 0-30 Hz. As this is not a true FRF measurement, it is difficult to extract some of the modes of the system. It can also be noted that a few modes seem to have relatively high auto-MAC values in the off-diagonal elements. On the basis of this analysis, the results from pressure based pseudo FRFs are not very encouraging. However, more insights will be available when results are compared with those from other datasets in Section 4.3. Additionally, alternate methods for validation of results from this test have been attempted and described in Section 4.6. A complete list of modal frequencies and damping values are provided in Table 2 in Section 5.

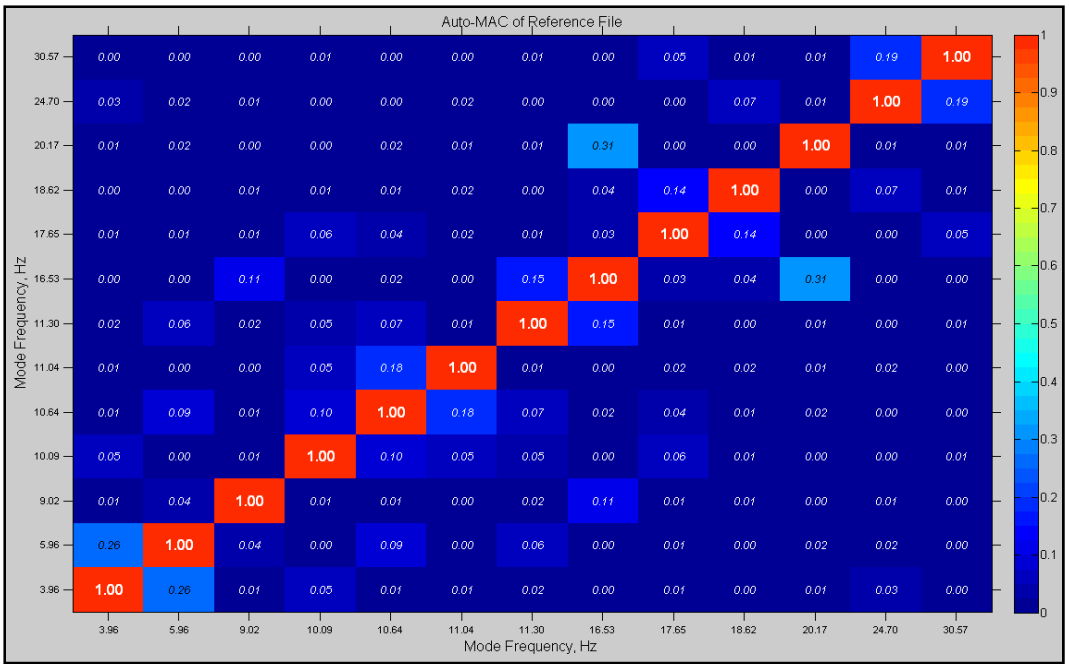

Figure 7: Auto-MAC plot of EMA test using pressure measurement as inputs

3.3 Pseudo-FRF based analysis using simulator excitations with displacement measurements as input signal (EMA displacement)

The EMA displacement based test uses input displacements from the LVDT sensors and response accelerations to construct pseudo FRFs for modal parameter estimation. The theory behind this method is discussed below.

$$\bar{H}_{st}(\omega) = \frac{\ddot{X}_s(\omega)}{X_t(\omega)} = \frac{H_{st}(\omega)}{H_{tt}(\omega)} \tag{2}$$

where

$$H_{st}(\omega) = \frac{\ddot{X}_s(\omega)}{F_t(\omega)} \tag{3}$$

and

$$H_{tt}(\omega) = \frac{X_t(\omega)}{F_t(\omega)} \tag{4}$$

where $\ddot{X}_s(\omega)$ denotes response accelerations on the structure and $X_t(\omega)$ represents the input displacements measured by LVDT sensors built into the simulator. Considering the fact that $F_t(\omega)$, representing force inputs, cannot be measured for this test, $H_{st}(\omega)$ and $H_{tt}(\omega)$ are not available for direct computation of $\bar{H}_{st}(\omega)$. $\ddot{X}_s(\omega)$ and $X_t(\omega)$ measurements are used instead, as represented in Equation 2. $\bar{H}_{st}(\omega)$ will give correct modal parameters

as long as $H_{tt}(\omega)$ and $F_t(\omega)$ are relatively constant and smooth over the frequency range of interest. For the MTS 320 four-post road simulator, this should be true from 0-50 Hz.

The data aquisition parameters for this test have been summarized below:

- Sampling Frequency : 125 Hz
- Frequency Resolution : 0.0625 Hz
- 50 RMS averages with 3 cyclic averages
- Window : Hanning
- Excitation degrees of freedom: 4
- Response degrees of freedom: 154 (150 on the structure, 4 on the wheelpan)
- Excitation Signal : Displacement
- Response Signal : Acceleration

To estimate driving point pseudo FRFs for this test, responses are measured on the wheelpans of the simulator rather than on the test structure. These driving point pseudo FRFs are essential to ascertain the dynamics of the column to be rigid within frequency range of concern for this study. Further, the presence of these measurements in the FRF matrix is essential for the algorithm to estimate modal parameters. It is important to note that these measurements do not provide any useful information regarding the test structure itself nor do they allow estimation of modal scaling as in the case of conventional driving point FRF measurements.

Driving point pseudo FRFs for this test, shown in Figure 8, are smooth curves and contain no poles in the frequency range of interest. This is in agreement with the theory discussed in Section 3.2. Hence this test is expected to estimate better modal parameters than the pressure based pseudo FRF test, which does not reflect this characteristic.

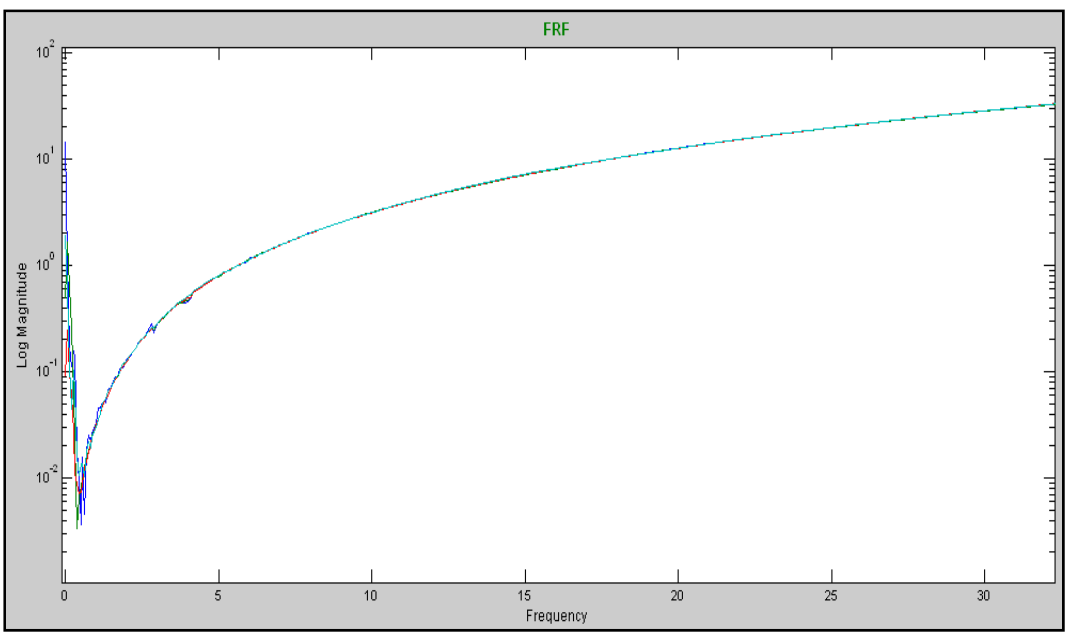

Figure 8: Pseudo FRF curves for EMA displacement test

To further validate the usability of this method for modal parameter estimation using the simulator, autopower spectra for the four input displacement measurements (Inputs 901z – 904z) and four response measurements (Response 901z – 904z) on the wheelpans of the simulator are studied (Figure 9). As random signals are used to excite the four columns of the simulator, the autopower spectra for the input measurements are expected to be flat with no prominent peaks. Further, the fact that the simulator column acts rigid in the frequency range of interest for this test implies that the response at the wheelpans should also be random in nature. Hence the autopower plots for the response points on the wheelpans would also be flat. These expectations are closely met by the autopower spectra of input displacements and wheelpan responses as shown in Figure 9, and confirm the

initial claim that the dynamics of the simulator do not affect measurements made in this test. This reiterates the applicability of the simulator for modal studies using displacements as input signals. Time histories for the pressure based EMA test discussed in Section 3.2 are not recorded for this study and hence a similar evaluation of autopower spectra for pressure inputs has not been discussed. However, as discussed in the last section, pressure based driving point pseudo FRFs do not exhibit these characteristics and provide an explanation for poor estimates using the same.

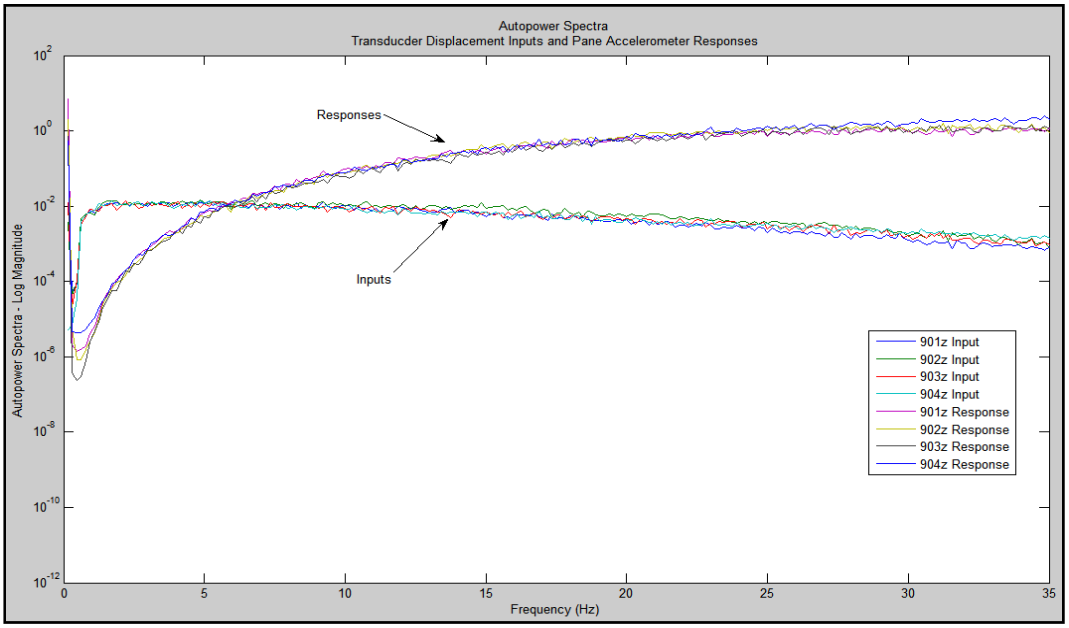

Figure 9: Autopower spectra for LVDT displacement inputs and wheelpan accelerometer responses

The PTD algorithm has been used for estimation, and a consistency diagram for one estimate is shown in Figure 10. It can be observed that this consistency diagram is significantly better than the consistency diagram for the EMA pressure test, indicating good parameter estimation.

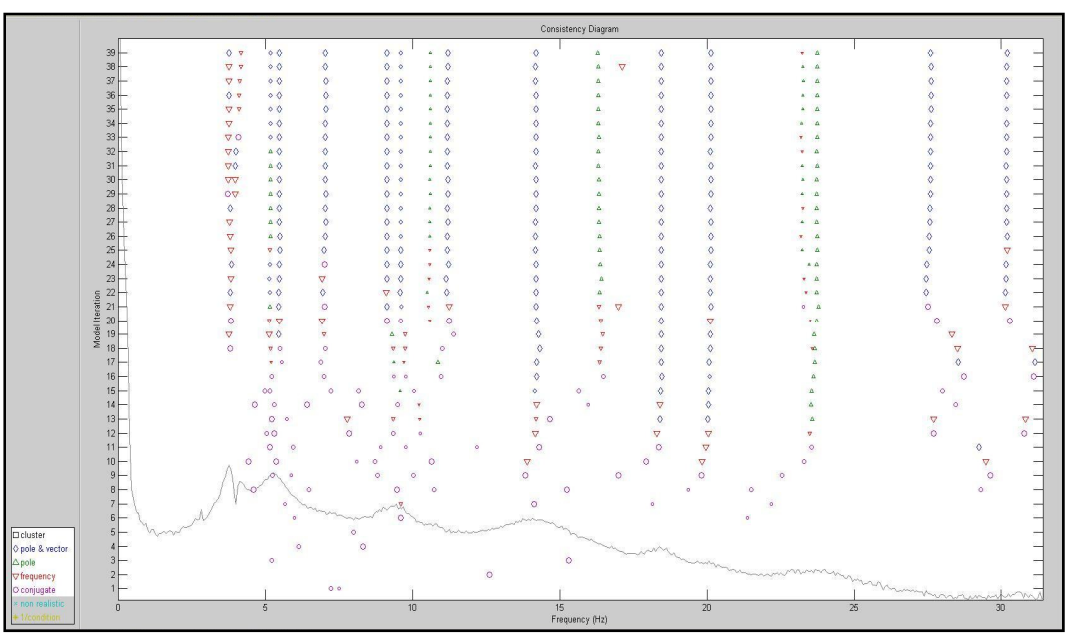

Figure 10: Consistency Diagram for the EMA displacement test

Seventeen modes are determined in the 0-30 Hz range, and have been summarized in Table 2. The auto-MAC plot for these modes has been shown in Figure 11, and the linear independence of the modes can be observed.

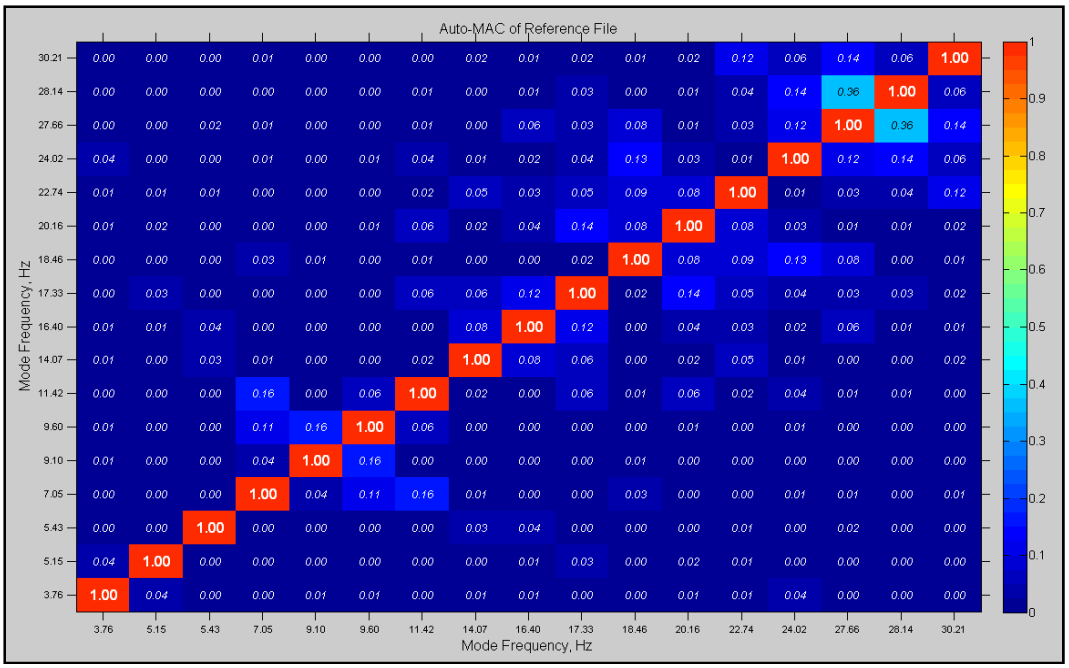

Figure 11: Auto-MAC plot for EMA displacement test

3.4 Operational Modal Analysis using four-post road simulator excitations

Tests conducted for OMA in the previous study [6] involved the use of random impact excitations in one case and shaker excitations in the other. Shaker excitations violated the OMA requirement of uniform spatial distribution, and limit the ability to identify all modes of the system. Random impact excitations, though conforming to the OMA assumptions and providing better results, were not representations of true operational conditions. This was due to the fact that both excitation methods were used directly on the frame and hence did not account for excitations coming from the ground through the suspension system. The use of the simulator in this study is expected to more closely replicate operational conditions such as the effects of vehicle ground interactions, and the role of the suspension system in isolating the vehicle structure from these excitations.

It is important to note that this test is not fully operational in the exact sense, as the effects of rotational and wind excitations are absent in the lab environment. This condition is favorable for this study, as OMA requirements of broadband excitational forces could be violated by the presence of sub-component interactions such as engine harmonics [5][12]. The simulated forces thus satisfy the spectral requirements of excitation for OMA. Given that these excitations are limited to forces coming through the four wheels and also being mitigated by the suspension system's mechanical filtering, the OMA requirements of spatially well distributed forces are partly violated. The effects of these violations on the quality of the estimates would be discussed in detail in Section 4.

Time histories recorded from 150 response channels are processed to obtain power spectra for use in OMA methods [9]. The data acquisition and processing parameters have been listed below:

- Sampling Frequency : 160 Hz
- Duration of data acquisition : 12 minutes (114892 time points)
- Number of excitation locations : 4
- Response degrees of freedom : 150
- Cyclic Averaging over 3 ensembles with 66.6% overlap processing employed for noise reduction
- Number of averages after cyclic averaging and overlap processing : 97

- Hanning window employed for reduction of leakage errors
- Response Signal : Acceleration

The power spectra data computed from the recorded response time histories are processed using two different algorithms; the Stochastic Subspace Identification (SSI) algorithm [10] and the PTD, both being time domain algorithms. There are inherent differences in the way the algorithms work, and hence there is a marked difference in the quality of parameters estimated by the two methods. The results from each algorithm have been listed below and compared in detail in Section 4.5.1.

3.4.1 OMA results using SSI-Data algorithm (OMA SSI)

The data dependent SSI algorithm (SSI-Data) [10], as implemented in B&K Type 7760, is first used to extract modal parameters for this test. This is an established OMA technique which has been shown to work well for typical OMA applications. It also differs from most other algorithms in the sense that it works directly on measured output responses without processing the time histories to calculate correlation functions (used by most time domain OMA algorithms including PTD). With the use of the SSI-Data algorithm, choosing five projection channels, fourteen modes are found to be present between 0-30 Hz. The MAC plot for the estimates obtained has been shown in Figure 12. Engine and lateral modes are not well excited given the nature of the test, and appear to have relatively high off-diagonal MAC values, as observed between the 16.4 - 7.64 Hz modes and the 22.85 - 24.32 Hz modes. The rest of the modes show a high degree of linear independence.

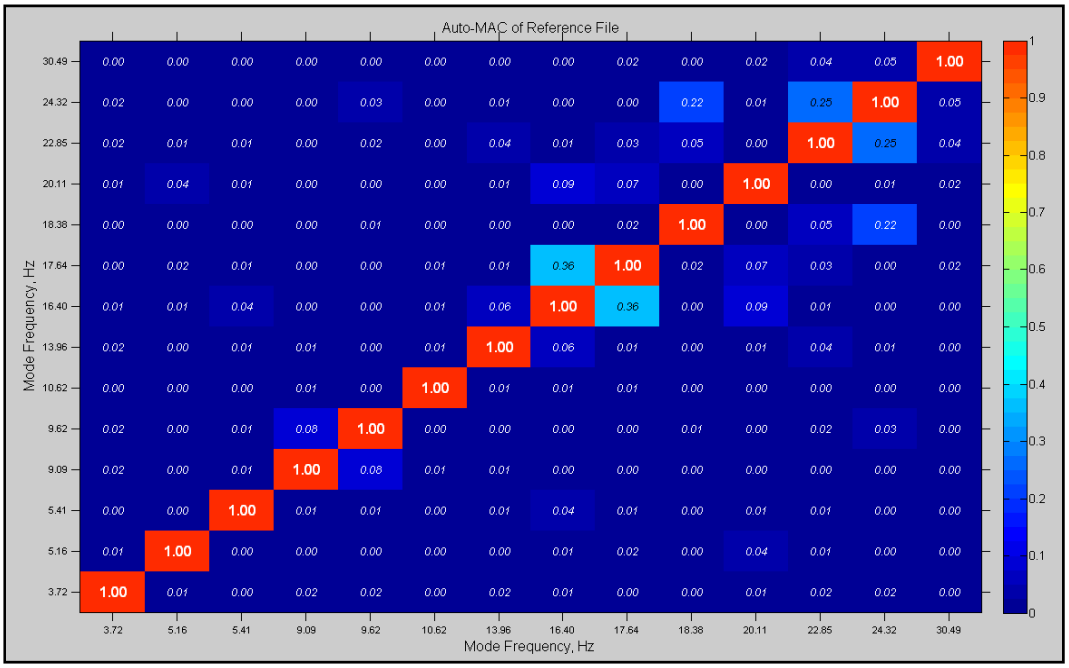

Figure 12: Auto-MAC plot for the OMA estimates using the SSI-Data algorithm

3.4.2 OMA results using PTD algorithm (OMA PTD)

In addition to SSI-Data, the PTD algorithm is used to process the response data to extract modal parameters. PTD is a high order algorithm and generally a few response channels are selected as reference responses for PTD to work satisfactorily. In order to choose the channels with maximum spectral information as references, the methodology used in the previous study is again employed [6]. Various combinations of response channels are used as references and their results are combined to choose the best estimates. Starting with two and choosing up to twenty reference responses, modes are obtained by consolidating estimates from various combinations of response channels used as references. A consistency diagram for one of the estimates using PTD algorithm on the response power spectra has been shown in Figure 13.

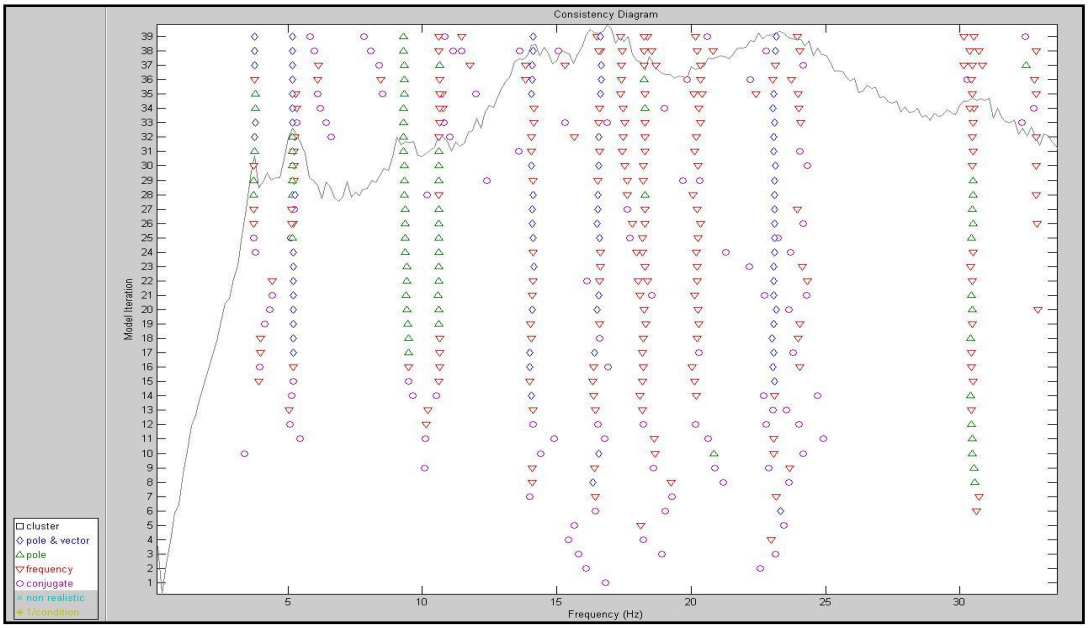

Figure 13: Consistency Diagram for an OMA estimate using the PTD algorithm

Fourteen modes are found in the 0-30 Hz frequency range. The auto-MAC plot for the estimates obtained has been shown in Figure 14. There is a clutter of off-diagonal MAC values observed in the higher frequency range. These are again the closely lying lateral modes, predominantly in the 20-30Hz range. One possible explanation for this could be that the combinations of reference responses chosen for parameter estimation did not have sufficient information pertaining to these lateral modes.

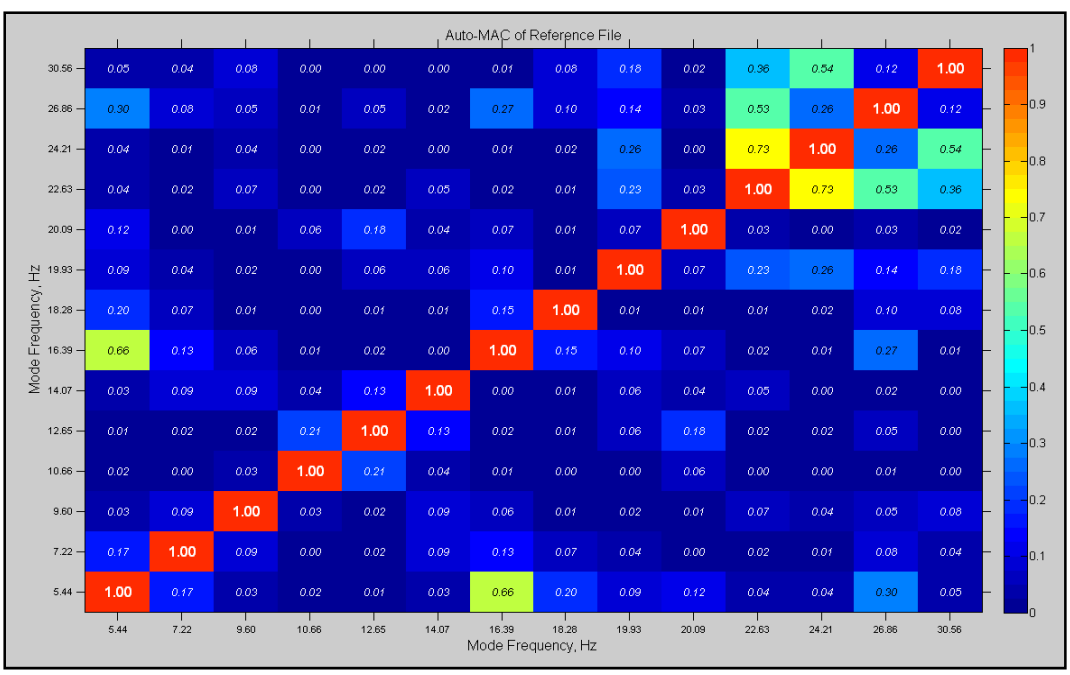

Figure 14: Auto-MAC plot for the OMA estimates using PTD

It is observed that results from PTD are very sensitive to selection of reference response channels. The fact that OMA assumptions of spatially well distributed input excitations are also not true in this case adds more complexity. While such issues are not encountered in EMA, they are critical in case of OMA as there are no definite guidelines about choosing reference responses, except that these responses should be able to extract all

modes of interest. One rule of thumb in this regard is to check the auto spectra of selected references and see if they have all modal information, i.e., peaks corresponding to most system modes. This is often a tedious task, particularly when a large number of sensors are involved and the system being analyzed is complex. Thus it is pertinent to find more effective ways of choosing suitable responses as references, especially in case of high order algorithms like PTD, Rational Fraction Polynomial (RFP) [11], etc. One way of utilizing maximum reference data would be the use of the SVD method to develop virtual reference responses for optimal parameter estimation.

4 COMPARISON BETWEEN ESTIMATES

4.1 EMA impact tests on the ground and the simulator

The first comparison is made between EMA impact test results with the structure raised on the simulator and results from an earlier test conducted on the ground [6]. This is done to validate the modes obtained by the impact test on the simulator. There is also another EMA impact hammer test conducted on the ground after bringing down the truck frame at the end of all experiments. There are various factors like spring stiffening, wheel camber angle change, lateral deflection of tires, etc., that can affect the test structure once it is lifted on to the simulator for tests, excited with the simulator and finally brought down to the ground at the end. These three tests and comparisons among them will help in better understanding of the changes the system undergoes across various tests. This will also help in ascertaining the time-invariant aspect of the structure, considering that all tests were conducted over a span of one month.

4.1.1 EMA impact 4 P vs EMA impact ground 1

From the cross-MAC plot shown in Figure 15, it can be observed that most of the modes obtained by the EMA 4 P test have good cross-MAC values with those obtained by the EMA ground 1 test. Some of the low cross-MAC values can be attributed to the fact that boundary conditions differ between the two tests, as mentioned earlier in Section 3.1. This might also be a reason for the slight shifts in frequencies across the two tests. Additionally, a mode at 28.03 Hz which could not be earlier estimated in the ground 1 test has been determined in the EMA 4 P test. Overall, there are not significant changes in the dynamic characteristics of the structure when it is on ground and when it is raised on the simulator.

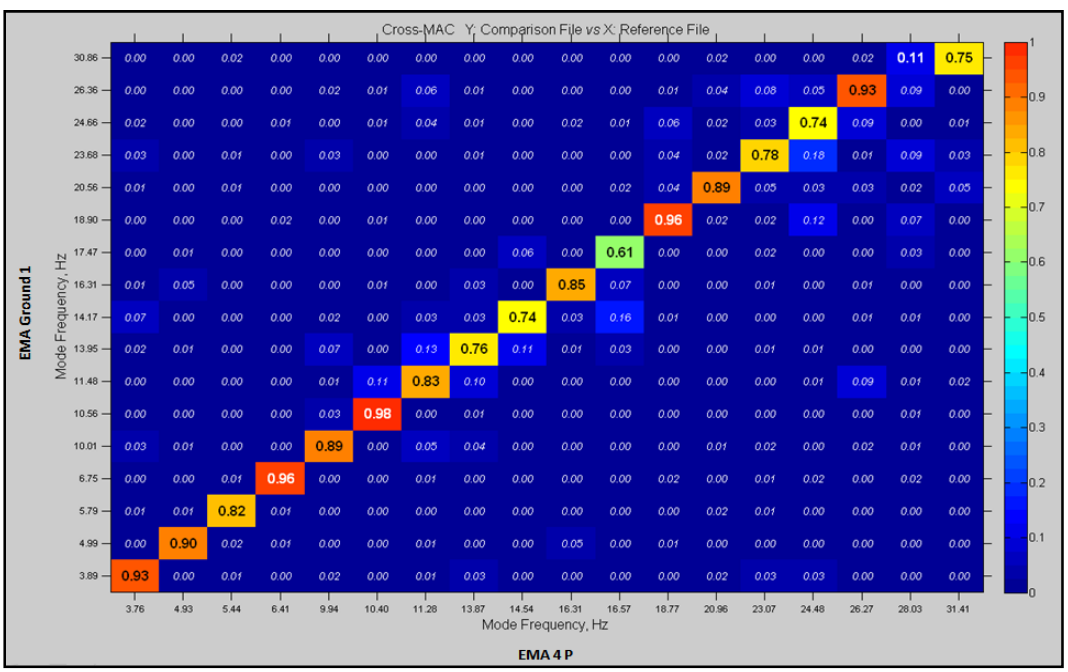

Figure 15: Cross-MAC of EMA impact 4 P vs EMA impact ground 1 estimates

4.1.2 EMA impact 4 P vs EMA impact ground 2

In this case also, most modes obtained in EMA 4P have been determined in the EMA ground 2 test and the corresponding cross-MAC values are high (refer Figure 16). The low frequency mode at 3.76 Hz is missing from the ground 2 estimates, probably due to the low energy of impact going into the test structure. It has also been observed from the previous study [6] that it is difficult to determine the closely lying engine modes at 13.95 -14.17 Hz, given the nature of the modes and limitations on the observability of engine modes. The higher frequency lateral modes at 23.07 Hz, 26.27 Hz and 28.03 Hz are also difficult to fully excite consistently in each test, as will be seen in further comparisons.

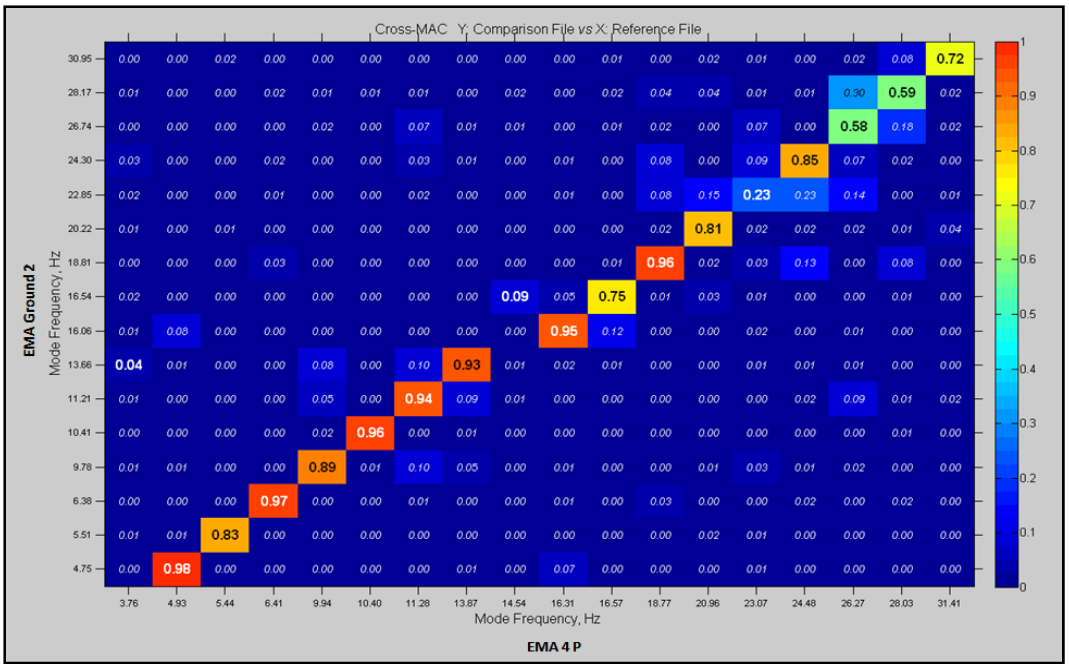

Figure 16: Cross-MAC of EMA impact 4 P vs EMA ground 2 estimates

4.1.3 EMA impact ground 1 vs EMA impact ground 2

As seen in the cross-MAC plot shown in Figure 17, most modes are determined consistently in the EMA tests with the vehicle on the ground before and after the use of the simulator, barring the modes at 3.89 Hz, 14.17 Hz, and some higher frequency modes beyond 20 Hz hitherto discussed. It is to be noted that the mode at 17.47 Hz estimated by the EMA ground 1 test has shifted to around 16.54 Hz. These factors might indicate a change in the dynamics of the system itself after running it on the simulator. As mentioned before, this could be due to suspension stiffening, wheel camber angle change when placing the vehicle on the ground, etc. However, based on these comparisons it can be concluded that the system dynamics are not affected much before and after the tests. This indicates the time-invariant nature of the system and also the fact that changing boundary conditions (structure on ground and structure on simulator with tires strapped) do not have a significant effect on its overall dynamics.

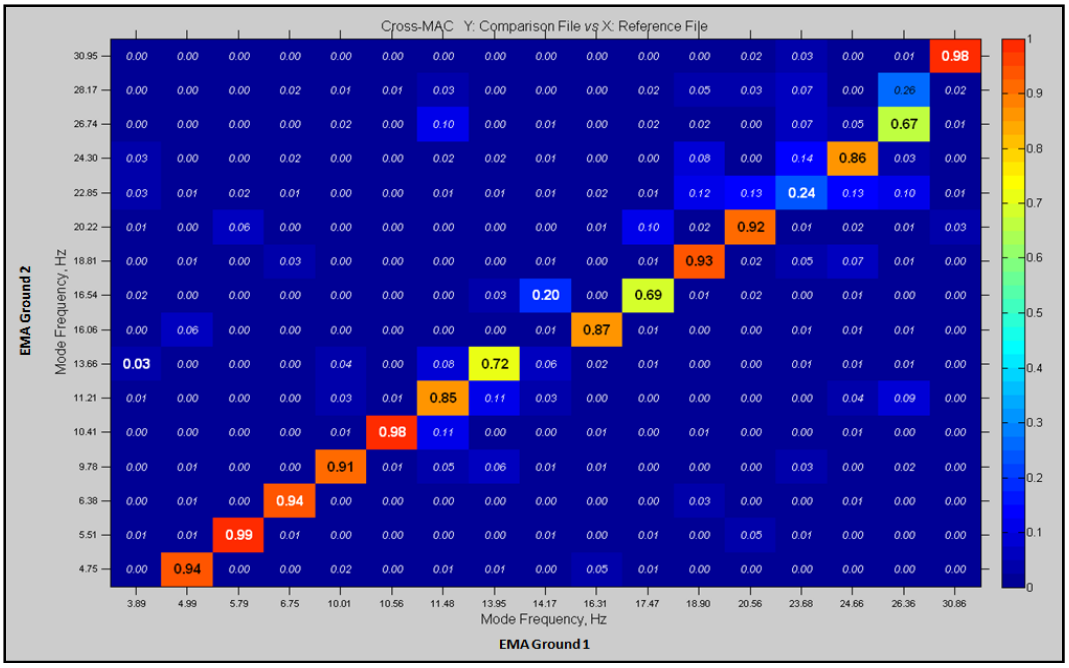

Figure 17: Cross-MAC of EMA impact ground 1 vs EMA impact ground 2 estimates

4.2 EMA impact 4 P vs EMA displacement

The EMA displacement test differs from the EMA 4 P test fundamentally in the nature of excitations. Unlike the direct excitation of the frame in the EMA 4 P case, the EMA displacement test involves the frame being excited by forces coming through the suspension system. The engine modes around 14 Hz are again poorly excited, as also the lateral modes beyond 20 Hz. Despite these limitations, some modes match well as can be seen from Figure 18. It has been observed that some modal vectors for this test have a high level of complexity, and hence result in low cross-MAC values for those modes. A visual inspection, however, shows a high level of visual similarity between these mode shapes and the expected shapes from conventional tests, and there is a need to look for validation methods beyond conventional MAC computations in order to fully understand this issue. The effects of these complexities in modal vectors on the MAC values have been discussed in relevant detail in Section 4.5.

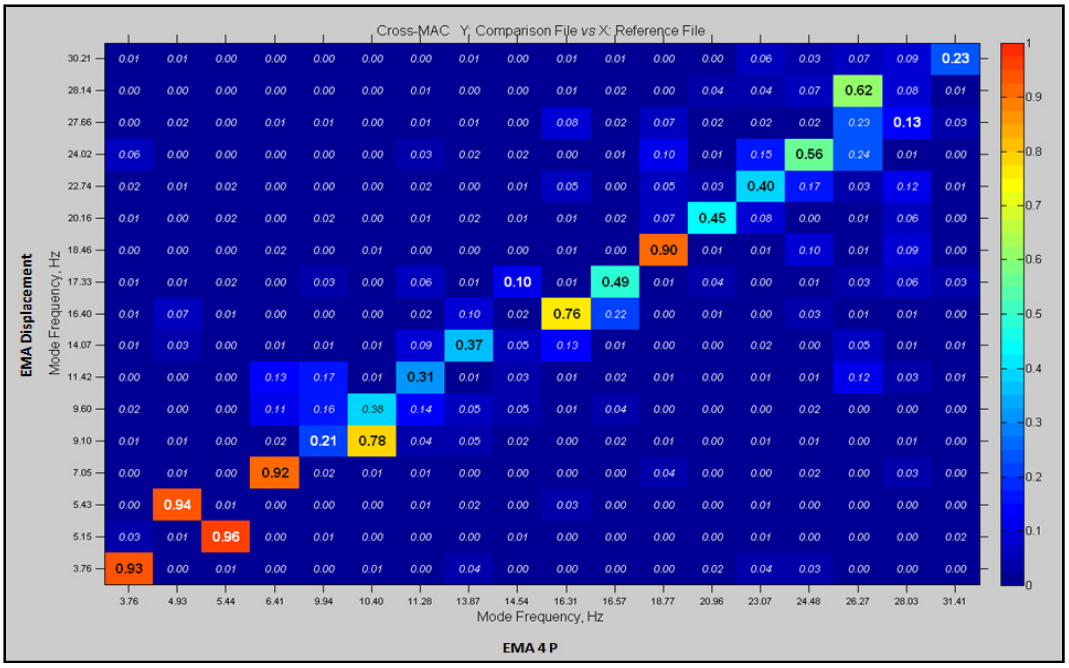

Figure 18: Cross-MAC of EMA impact 4 P vs EMA displacement estimates

4.3 EMA impact 4 P vs EMA pressure

A cross-MAC plot (Figure 19) between estimates from the EMA pressure test and the EMA 4 P test indicates that most of the modes do not have a high cross-MAC value. As explained earlier by means of pressure based driving point pseudo FRFs, parameters obtained using this method are inconsistent and not very accurate. This does not instill high level of confidence in the results obtained using pressure based EMA. However, visual inspection of some mode shape animations indicates a similar underlying pattern in both estimates (impact 4P and pressure based), with the EMA pressure modes tending to display more complexity in motion. An approach based on separating the components of the modal vectors and recalculating MAC values has been made in Section 4.5. This approach results in increasing the MAC values by a certain level thus being more indicative of the similarity between mode shapes. However it can be said at this point of time that pseudo FRFs based on displacements offer more potential than those based on pressure measurements.

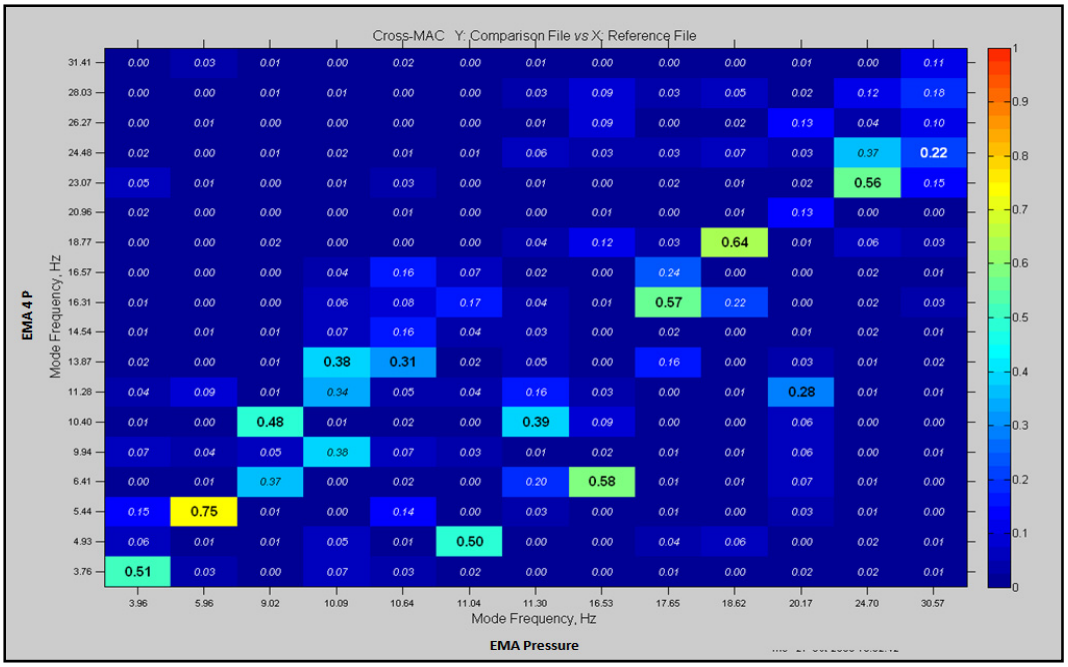

Figure 19: Cross-MAC of EMA pressure vs EMA impact 4 P estimates

4.4 Operational modal analysis results and comparisons

SSI-Data is an established OMA technique known to work well for typical OMA applications. Results discussed in Section 3.4 also indicate estimates from OMA SSI to be better than those from OMA PTD, as it does not involve variability on account of responses chosen as references. Hence, modal parameters extracted using SSI-Data instill more confidence in comparison to PTD estimates and will be used for comparison with estimates from the EMA 4 P and EMA displacement.

4.4.1 OMA SSI vs OMA PTD

Response data collected for the OMA studies are processed using two time domain algorithms, PTD and SSI-Data. Some modes determined by the use of the PTD algorithm for the OMA test match with those by the SSI-Data algorithm, as shown in Figure 20. The high order PTD algorithm is highly sensitive to the choice of reference channels, especially when OMA requirements for the excitations are not fully met and hence present inherent challenges in the estimation of modes. As mentioned earlier, it will be interesting to find an optimal method for determining the reference channels for use with this algorithm.

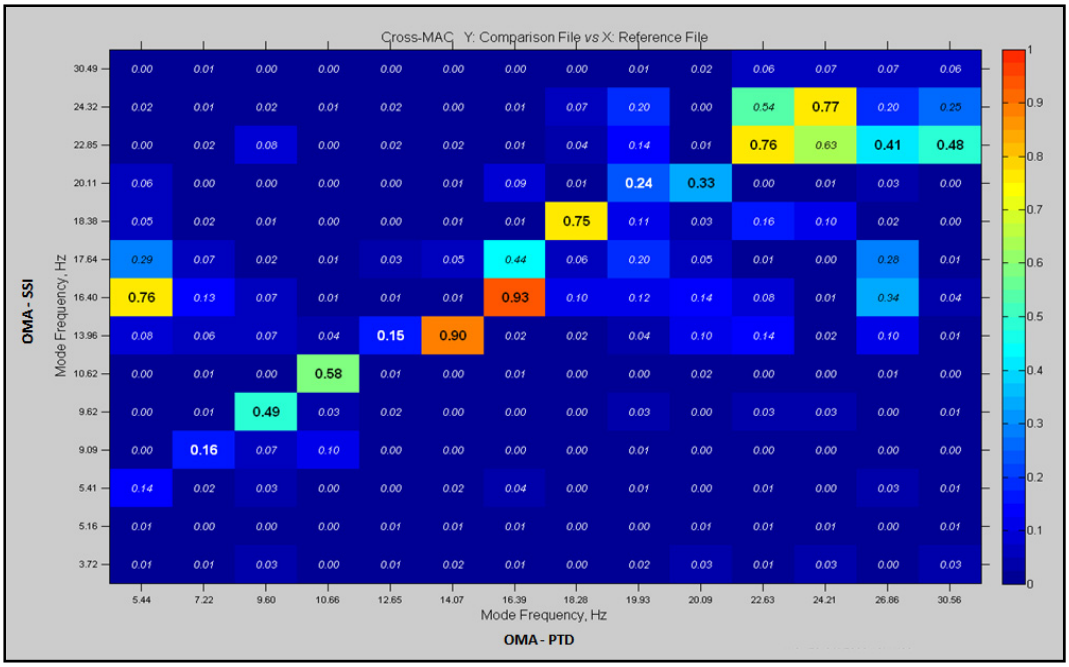

Figure 20: Cross-MAC of OMA PTD vs OMA SSI

4.4.2 EMA impact 4 P vs OMA SSI

One important aspect of the structure being tested using the simulator is the variation in boundary conditions. The conventional EMA tests had the vehicle being tested on the ground or on the simulator when the boundaries were rigid. Subsequent tests have the posts of the simulator exciting the structure and involve dynamic boundary conditions. This in turn induces moderate to significant shifts in modal frequencies of the structure.

A significant observation has been the reordering of the sequence in which the low-frequency rigid-body modes appear in the estimates. Conventional tests with static boundary conditions have the front-axle pitching mode at 4.93 Hz preceding the yaw mode at 5.44 Hz. The tests with the simulator in action, however, consistently see the yaw mode preceding the front-axle pitching mode, as observed in the cross-MAC plot shown in Figure 21. This clearly indicates changes in the dynamics of the structure in operation – a phenomenon that could not be observed in tests where boundaries are static. Note that same behavior was observed in case of displacement based EMA tests which share the same excitation scenario as OMA tests (Figure 18).

Also, as emphasized several times, excitations generated by the simulator violate the OMA requirement of good spatial distribution over all degrees of freedom. The fact that the frame and engine are not directly excited, but through the suspension system further affects the energy that goes into the response degrees of freedom of those sub-structures. This explains the poor cross-MAC coefficients for some of the modes - for instance, the engine mode at 13.87 Hz. The simulator generates excitations primarily in the vertical direction, and hence most lateral modes such as the frame modes between 26-29 Hz are poorly excited as well. In view of these limitations, not all modes determined by the OMA test have high MAC values against EMA 4 P results. But the low-frequency rigid-body modes compare very well with corresponding modes from the EMA 4 P test, as also some of the dominant deformation modes at higher frequencies.

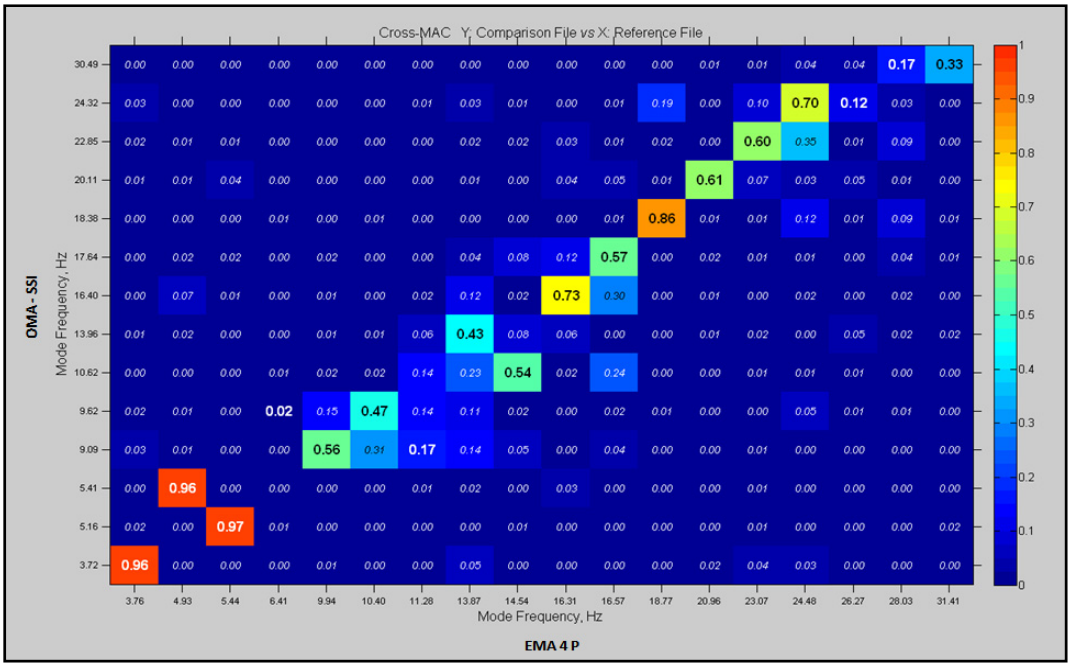

Figure 21: Cross-MAC of EMA impact 4 P vs OMA SSI estimates

4.4.3 OMA SSI vs EMA displacement

The EMA displacement test utilizes pseudo FRF computations, while the OMA test relies on response-only data for parameter estimation. However, these tests are very similar to each other with respect to the nature of the boundary conditions and the physical system setup and excitation scenario. The mode estimates by these two methods are found to be in strong agreement with each other, as shown by the cross-MAC plot in Figure 22. The low frequency rigid body modes match very well both in terms of frequencies and cross-MAC values. The mode at 10.62 Hz, being a transaxle mode, has not been excited well due to the nature of input forces. Unlike the displacement-based test which is closer to conventional FRF-based methods, OMA estimates using the SSI-Data algorithm are affected to a greater extent by violations in terms of the nature of excitations, and hence certain modes such as the rigid body mode around 7 Hz, and modes at 11.42 Hz and 28.14 Hz have not been estimated with the SSI-Data algorithm for the OMA test.

Modal vectors from both methods have a significantly complex nature for some of the modes, especially the deformation modes. While conventionally, a MAC value of 0.8 or higher is considered good as a rule of thumb, it is prudent, considering the nature of these tests, to relax the expected cross-MAC limit for a valid comparison. A visual inspection of the mode shapes obtained in these tests indicate a high level of similarity. The overall results from this comparison are extremely encouraging as they indicate positively that displacement readings from the simulator can be utilized for finding dynamic characteristics of an automotive structure while simulating true road conditions.

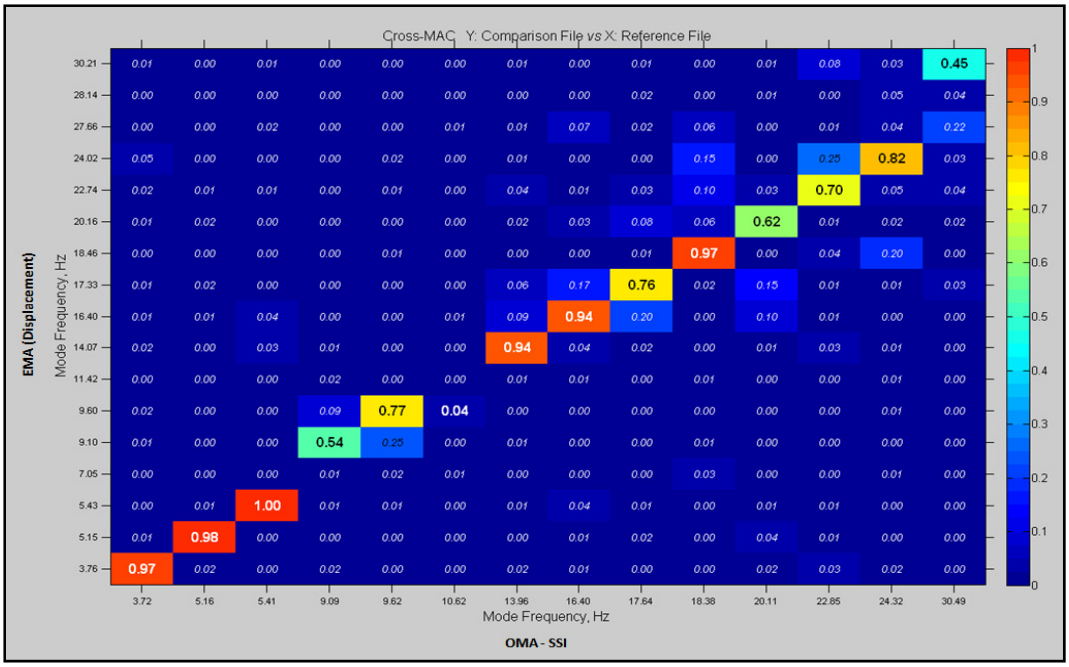

Figure 22: Cross-MAC of OMA SSI vs EMA displacement estimates

As discussed in Section 4.3, pressure based EMA test results do not have high cross-MAC values in comparison with EMA 4 P test results. The various issues involved with this particular method at present require a deeper understanding to enable its usage for successful parameter estimation. Comparison between modal estimates from the OMA SSI test and the EMA pressure test has predominantly low cross-MAC values across most of the modes. Hence this discussion is not presented further in the paper.

4.5 Enhanced MAC computations

Conventional validation methods have generally utilized computation of MAC coefficients for comparing modes obtained between multiple estimates. This is known to work well for traditional EMA tests where the measurement of force consistently normalizes the data. However, modal vectors estimated by OMA methods and the unconventional tests sometimes have a significant complex character and hence there is a need for an altered validation methodology to compare modes with greater reliability. Note that this complex character is not simply a complex scalar times an underlying similar mode shape; MAC is not sensitive to this complex scaling issue when computed properly.

For enhanced MAC computations, modal vectors obtained from various tests are studied in terms of their real and imaginary components. While normal modes obtained from conventional EMA tests have a dominant imaginary portion, modal vectors obtained from the OMA test and from the tests based on pseudo FRF measurements do not display similar characteristics. This method compares complex modal vectors from the EMA 4 P test with real and imaginary components of modal vectors from the other tests separately to recompute cross-MAC values. The modes chosen for comparison are the ones which have been observed to have a high similarity in their mode shapes when visually inspected, yet having a low cross-MAC coefficient.

Figure 23 displays traditional and recomputed cross-MAC values between estimates from the EMA 4 P test and from the EMA displacement test for certain modes. It can be observed that cross-MAC values between complex modal vectors from the EMA 4 P test and the dominant imaginary component of the modal vectors from the EMA displacement test have a marginal improvement over cross-MAC values computed between complex modal vectors from both tests. The use of imaginary components of modal vectors from both datasets further improves the MAC computations, and is perhaps more reflective of the observed visual similarity in mode shapes. It is to be noted that these enhanced MAC values might still be much lower than the high values expected for similar mode shapes.

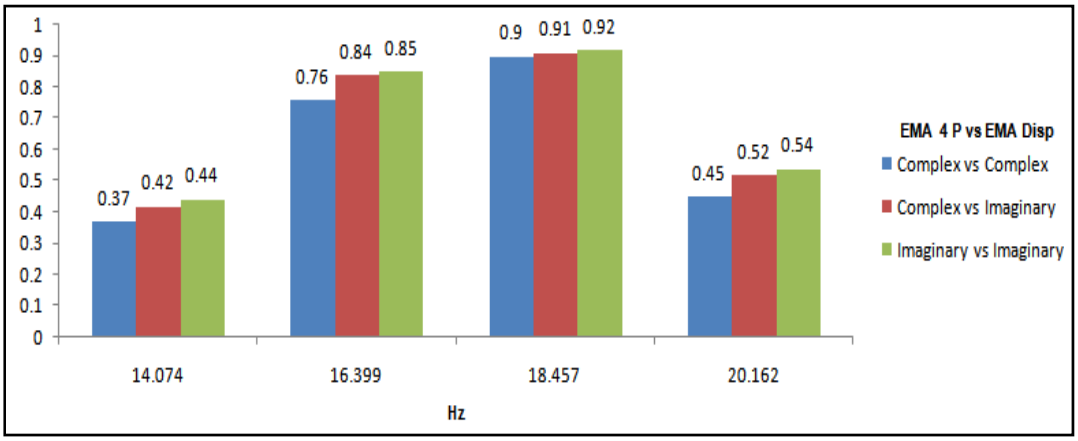

Figure 23: EMA 4 P versus EMA Displacement

In the case of the OMA estimates obtained using the SSI-Data algorithm, modal vectors are found to have a dominant real portion. Hence these real components are used in recalculation of cross-MAC values against complex vectors from the EMA 4 P test. As seen in Figure 24, there is a moderate improvement in the recalculated cross-MAC values in comparison with the traditional cross-MAC coefficients. Further, use of the imaginary portion of the OMA SSI modal vectors displays very low cross-MAC values, hinting towards the presence of normal modes. An exception to this observation is the mode at 9.647 Hz where the cross-MAC value computed with the imaginary component of the OMA modal vector is relatively higher, and is indicative of higher complexity in the particular mode.

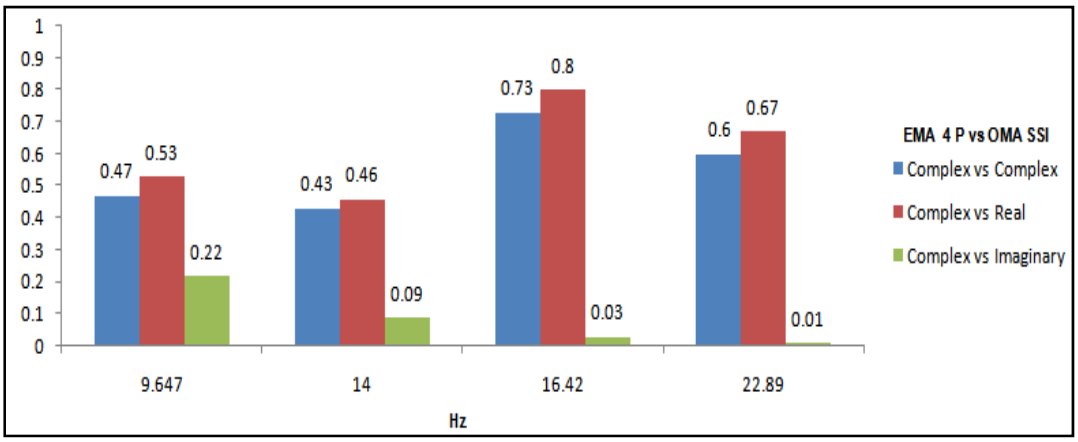

Figure 24: EMA 4P versus OMA SSI

Modal vectors from the EMA 4 P test are compared with modal vectors from the EMA pressure test as shown in Figure 25. Modal vectors from the EMA pressure test are observed to have either a dominant real or a dominant imaginary component for each mode, without exhibiting a specific pattern across the modes. Keeping this in mind, cross-MAC values are computed using both real and imaginary components of the modal vectors for this dataset. It can be observed from the figure that there is a marked improvement in the cross-MAC coefficients between the complex modal vectors from the EMA 4 P test and the dominant component of the modal vectors from the EMA pressure test. It is also relevant to note that cross-MAC values computed using the non-dominant component of the modal vectors from the EMA pressure test are very low as expected. This observation supports the earlier claim that in consideration of the numerical characteristics of the pseudo FRFs measured for the EMA pressure test, enhanced validation methods would be essential for a more reliable comparison of these results with traditional estimates.

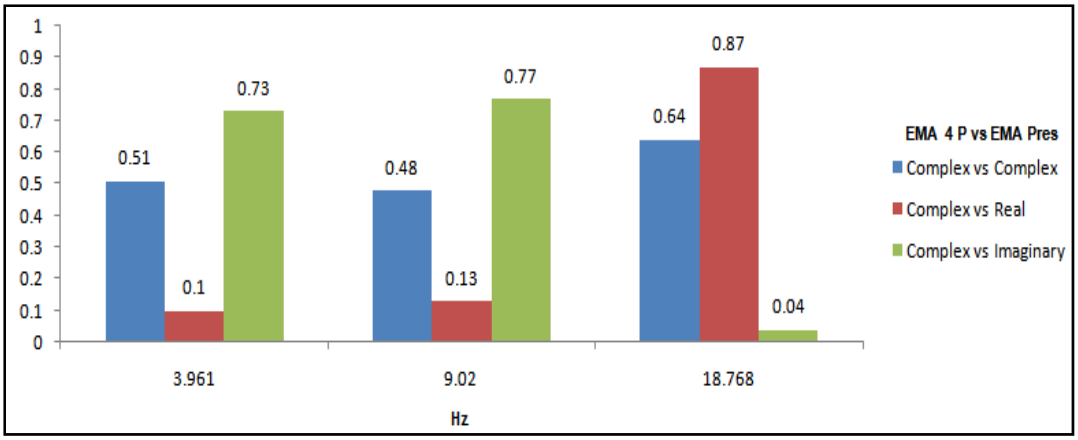

Figure 25: EMA 4P versus EMA Pressure

Tests conducted based on response-only data and the two pseudo FRFs significantly differ from conventional EMA tests in terms of their numerical characterisitics, and consequently, their modal vectors have higher levels of complexities. As has been demonstrated in this study, calculation of MAC coefficients after examination of these complexities, or real normalizing the modal vectors, might be a better validation methodology than conventional MAC computations.

5 SUMMARY OF RESULTS

Estimates from the displacement-based EMA test and OMA SSI test have been the most promising results across this study. Visual inspection of mode shapes also confirm similarities across these tests for the rigid-body modes and the dominant deformation modes. Estimates from the displacement-based EMA test are used for the description of prominent mode shapes in Table 1. Prominent rigid-body modes have been found below 10 Hz. The first torsion mode appears at 11.42 Hz and the first frame bending mode is observed at 18.46 Hz. These mode shape descriptions are consistent with corresponding modes across all tests.

Mode	Description of mode shape	
3.76	Rocking in the lateral direction	Rigid Body Modes
5.16	Yawing	
5.43	Front axle pitching	
7.05	Rear axle pitching	
9.1	Rolling	
9.6	Transaxle bending	Deformation Modes
11.42	First torsion	
14.07	Engine rolling with transaxle lateral movement	
16.4	Engine pitching out of phase with frame	
18.46	First frame bending	
20.16	Complex mode with leaf springs bending	
22.74	Complex mode with lateral frame bending	
24.02	Engine lateral swaying with frame twisting	
30.21	Higher order lateral frame bending	

Table 1. Mode shape descriptions

Table 2 lists the complete set of modal frequencies and damping values for all the tests performed in this study. It can be observed that not all modes have been estimated across the various tests, reasons for which have been discussed earlier in Sections 3 and 4. Due to the pseudo-forms of the FRFs used for modal parameter estimation, reasons for high damping estimates observed in Table 2 are not known at present, and need further investigation.

EMA Ground 1		EMA 4 Poster		EMA Ground 2		OMA SSI		EMA (pressure)		EMA (displacement)		Avg. Freq.	Std. Dev.
Freq	*Damp*	*Freq*	*Damp*	*Freq*	*Damp*	*Freq*	*Damp*	*Freq*	*Damp*	*Freq*	*Damp*		
Hz	*%*	*Hz*	*%*	*Hz*	*%*	*Hz*	*%*	*Hz*	*%*	*Hz*	*%*	*Hz*	-
3.89	*1.28*	3.76	1.96	-	-	3.73	7.59	3.96	0.81	3.76	4.17	3.82	0.10
4.99	*1.57*	4.93	1.99	4.75	2.03	5.43	8.58	-	-	5.43	8.40	5.19	0.31
5.79	*1.81*	5.44	2.02	5.51	1.94	5.17	5.52	5.96	14.02	5.16	5.57	5.50	0.32
6.75	*2.45*	6.41	2.29	6.38	2.54	9.11	5.81	9.02	7.17	7.05	16.28	7.67	1.27
10.01	*2.28*	9.94	2.59	9.78	2.54	9.65	6.84	10.09	8.18	9.10	6.55	9.76	0.36
10.57	*1.76*	10.40	1.65	10.41	1.55	10.63	3.07	10.64	1.72	9.60	5.45	10.37	0.39
11.48	*3.17*	11.28	3.00	11.21	2.99	-	-	11.04	10.62	11.42	14.64	11.30	0.18
13.95	*2.32*	13.87	3.00	13.66	2.81	14.00	8.23	-	-	14.07	8.08	13.97	0.16
14.17	*1.92*	14.54	1.75			-	-	-	-	-	-	14.36	0.26
16.31	*2.08*	16.31	2.34	16.06	2.83	16.42	5.26	16.54	5.23	16.40	6.42	16.39	0.16
17.47	*1.36*	16.57	1.47	16.54	1.66	17.67	5.72	17.65	9.66	17.33	7.06	17.34	0.52
18.90	*1.23*	18.77	1.71	18.81	0.94	18.39	2.97	18.62	2.70	18.46	2.89	18.63	0.20
20.56	*1.29*	20.96	1.34	20.22	1.26	20.12	2.81	20.17	4.18	20.16	3.19	20.39	0.33
23.68	*1.79*	23.07	*1.82*	22.85	1.14	22.90	6.29	-	-	22.74	5.84	23.10	0.37
24.66	*1.36*	24.48	*1.91*	24.31	1.85	24.35	5.19	24.70	0.79	24.02	6.54	24.44	0.25
26.36	*1.49*	26.27	*1.78*	26.74	1.46	-	-	-	-	27.66	1.47	26.76	0.63
-	-	28.03	*1.65*	28.17	1.57	-	-	-	-	28.14	5.88	28.08	0.08
30.86	*0.76*	31.41	*1.17*	30.95	0.65	30.50	2.36	30.57	1.16	30.21	1.77	30.71	0.42

Table 2. Summary of modes obtained from all tests

6 CONCLUSIONS AND SCOPE FOR FUTURE WORK

In the previous study, OMA methods using simulated excitations were successfully evaluated against conventional EMA methods. However, the need was felt to evaluate OMA methods under more realistic operational conditions. This is achieved in this study with the use of a four-post road simulator for exciting the vehicle structure through the wheels and the suspension system.

As part of this study, conventional FRF-based impact tests have been conducted on the structure at specific stages of this exercise to observe variations in the system over the length of the study. The first of the three tests (EMA ground 1), performed with the vehicle on the ground, characterizes the dynamics of the system in terms of its modal parameters. The second test (EMA 4 P), with the structure mounted on and strapped to the simulator, studies the effects of variation in boundary conditions induced by the straps and the simulator posts on the modal parameters of the system. A third test is conducted at the end of the study (EMA ground 2) to identify changes in the system such as suspension stiffness variations and wheel camber angle changes. Between them, these conventional tests evaluate the time-invariance of the structure as it is subjected to a variety of excitations with differing boundary conditions and shifts in frequencies have been discussed.

Three tests have been conducted to explore the potential of the simulator for modal applications. The first test utilizes response data collected across the structure under random excitations from the simulator for parameter estimation using OMA methods. Excitations in this test, although limited spatially and by the action of the suspension system, are a closer representation of true operational conditions for the automotive structure. Modal estimates from this test have been found to be in substantial agreement with a conventional FRF-based impact test conducted on the structure. The OMA test further highlights variations in the modal signature of the vehicle when in operation due to the influence of dynamic boundary conditions. It is significant to note that such shifts in the pattern of modes are difficult to determine using conventional EMA tests with static boundary conditions. The OMA test using the simulator has thus been found to be fairly successful in estimating modal parameters of the structure, as also in understanding variations in the system dynamics under true operational conditions.

Further, two tests have been conducted with pressure and displacement measurements from the simulator instead of force in the synthesis of FRFs. These are subjected to parameter estimation using conventional EMA methods. Estimates from the displacement-based EMA test have been found to be in very good agreement with benchmark estimates from the EMA 4 P test. Estimates from this test are also in substantial agreement with estimates from the OMA test, and also indicate the reordering of rigid-body modes under operational conditions. Of significance is the similarity in the order of modes between the estimates from this test and those from the OMA test, and reiterates the influence of dynamic boundary conditions on the modal behaviour of the system. The displacement-based test thus makes a strong case for further research in the applicability of the four-post road simulator for modal analysis.

The pressure-based EMA test has inherent limitations in its suitability for experimental modal analysis using conventional methods. The numerical characteristics of the pressure-based pseudo-FRFs limit the extent to which pressure can be used as an alternative input measurement to force. The estimates obtained from this test, though not very encouraging considering conventional MAC validations, do see an improvement in the cross-MAC coefficients when the effects of modal vector complexities are factored into the validation process. In addition, visual inspection of mode shapes from this test with those from other tests shows a good level of similarity, and encourages further investigation into this method for modal analysis.

Considering the presence of complexities in the modal vectors determined by the tests on the simulator, future work will involve the development of better validation techniques that would take into account these complexities. The possibilities of improving estimates from the pressure-based test by compensating for the inertial effects of the moving components of the simulator column can be explored. For the OMA studies, a future test could be done with the vehicle running on a test track in true operational conditions, and estimates obtained would be compared with the results from the road simulator test. A more efficient method to identify the most suitable reference channels for use with time and frequency domain algorithms such as PTD, RFP, etc, which are sensitive to selection of references, will be evaluated based upon an initial SVD reformulation to virtual references.

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