SAND2022-14600C

Operational Analysis of a Structure with Intermittent Impact

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ABSTRACT
Modal characterization of a structure is necessary to inform predictive simulation models. Unfortunately, cost and schedule limitations tend to prioritize other dynamic tests, which can lead to inadequate or nonexistent modal testing. To utilize the dynamic test data that is acquired, analysts can extract operational deflection shapes (ODS) which can then be used as a substitute for modal data in model updating and structure characterization. However, extremely high levels of excitation during vibration testing may introduce nonlinear behavior that distorts the ODS prediction. This paper investigates the reliability of using ODS as a replacement for traditional modal testing on an academic structure designed to respond with intermittent impact. This paper calculates ODS from responses at several input excitation levels, and the influence of nonlinear impact on the resulting operating modes is discussed.

Keywords: ODS, Model Updating, Modal Analysis, Dynamic Environments Testing

INTRODUCTION

Knowing the modal parameters of a structure is an important step towards understanding how that structure responds to dynamic excitation, as well as a key step in model validation. Traditional modal testing, however, is not always possible. As a substitute, operational techniques such as operational modal analysis (OMA) and operational deflection shapes (ODS) can be used to estimate the mode shapes and frequencies of a structure without traditional modal testing. Contrary to modal testing, structural response is measured during the operation of the structure, such as while a machine is running or in response to wind blowing past a building. Operational techniques are historically used to predict mode shapes of structures that cannot be tested with traditional modal techniques, such as extremely large structures like buildings or bridges [1]. The techniques can provide accurate predictions of the structure's modes, but only if the structure is properly excited. Operational techniques will miss modes if certain directions or subcomponents of a structure are not adequately excited. Similarly, if the excitation spectra does not include a certain frequency range, any mode shapes in that frequency range will not be captured. This is very likely in operational measurements where the excitation of the structure is uncontrolled.

More recently, operational techniques have been used to gain insight about structures that could be modal tested when modal data does not exist. Test schedules may not prioritize modal testing, but other test data such as shaker table vibration data are acquired. Performing operational techniques on the existing test data can give analysts some indication of the mode shapes and frequencies, which can help validate models. In addition, the excitation input spectra of vibration testing is known. Although the actual inputs to the structure in a vibration test are not measured, there is some insight about the structural excitation that is captured and can be used to help identify frequency ranges or components that may have mode shapes missed in the operational shape estimation due to inadequate excitation.

As operational techniques are used more often and for smaller complex structures, an investigation of potential pitfalls is needed. Nonlinear structural response in combination with varying levels of dynamic excitation have the potential to affect the ODS predictions. Excitation level has previously been shown to affect operational frequencies and shapes of a bridge and bell tower [2]. However, the modal parameters of those structures were not measured or analytically predicted, so the difference between the operational shapes and frequencies and the actual shapes and frequencies was not quantified.

This paper investigates the influence of excitation levels on the predicted ODS of an academic structure with and without components with observable nonlinear contact in order to understand the reliability of predicted ODS from high-amplitude vibration test data.

THEORY

Modal analysis is a powerful tool to characterize the frequencies, damping, and mode shapes of a structure. In traditional modal analysis, these characteristics are measured in a controlled test environment where all forces acting on the structure are measured. The response of the structure to a given load can be predicted once these characteristics of a structure are known. However, excitation forces required for traditional modal analysis may not be measured, either due to unknown forces acting on a structure in an operational setting or due to testing constraints. Operational modal analysis (OMA) and operational deflection shapes (ODS) are two similar approaches that can estimate the mode shapes of the structure without performing traditional modal analysis. The key differences between traditional modal, OMA, and ODS is based on the forces that are measured in the test and how the modal parameters are estimated [3].

In modal analysis, an excitation force and location are chosen to excite the full structure up to a frequency of interest. The relationship between the response of the structure and the excitation is calculated as a frequency response function (FRF). The H_1 formulation of the FRF is given in Equation 1, where G_{xf} is the cross power spectrum between the response spectrum and the force spectrum, and G_{ff} is the auto power spectrum of the force. An FRF is calculated between each response measurement channel and each excitation channel, creating an FRF matrix.

$$\boldsymbol{FRF} = \frac{\boldsymbol{G_{xf}}}{\boldsymbol{G_{ff}}} \qquad \textbf{1}$$

The FRF matrix is then used to estimate the poles of the system, which are also estimates of the modal frequencies of the system. The partial fraction form of the FRF is given in Equation 2, where $H(j\omega)$ is the FRF at a chosen frequency line given by the summation of m modes of the system, A_k is the residue matrix for the k^{th} mode, and p_k is the pole for the k^{th} mode. When the chosen frequency line is at the pole of a mode the term for that mode becomes very large, dominating the overall response of the structure. For systems with well-spaced modes, the response of the system at that frequency line will resemble a mode shape of the structure [4].

$$[\boldsymbol{H}(\boldsymbol{j\omega})] = \sum_{\boldsymbol{k=1}}^{\boldsymbol{m}} \frac{[\boldsymbol{A_k}]}{(\boldsymbol{j\omega} - \boldsymbol{p_k})} + \frac{[\boldsymbol{A_k^*}]}{(\boldsymbol{j\omega} - \boldsymbol{p_k^*})} \qquad \textbf{2}$$

One common method to estimate poles, and the method used in this paper, is the complex mode indicator function (CMIF). The CMIF is a plot of the singular values of the FRF matrix at each frequency line. The singular value decomposition equation is given in Equation 3, where A is a matrix, Σ is the diagonal matrix of singular values of A, U and V are singular vector matrices of A, and $\dagger$ indicates the Hermitian transpose [5]. Once the poles of the system have been estimated, the modal parameters of the structure can be estimated using curvefitting techniques, such as polyMAX, to extract the modal parameters of the structure.

$$\boldsymbol{A} = U\Sigma V^{\dagger} \qquad \mathbf{3}$$

OMA attempts to use response data without measured force data to estimate the mode shapes of a structure. Instead, the structure is measured during an operating condition with the assumption that the excitation of the structure is broadband. A true FRF cannot be computed because the excitation force is not measured. However, a reference measurement can be selected, and transmissibility functions can then be calculated between the chosen reference measurement and all other measurements. An example transmissibility function, TF, is given in Equation 4, where G_{xr} is the cross correlation spectra between the response spectra and the reference spectra, and G_{rr} is the auto power spectra of the reference.

$$\boldsymbol{TF} = \frac{G_{xr}}{G_{rr}} \qquad \mathbf{4}$$

The same steps as modal analysis are followed using the TF instead of the FRF: poles are estimated and the TF is curvefit to estimate the modal parameters of the structure. However, there is no guarantee that the excitation is broadband or that the structure is fully excited. If so, mode shapes that are not excited will not be present in the data. The choice of reference point may also distort the predicted modal parameters, for example if the reference point is located at a node of a mode.

ODS also attempts to use response data without measured force data by selecting a reference point and computing the TF. However, ODS does not attempt to curvefit the TF. Instead, ODS assumes that the response of the structure at an estimated pole of the system is dominated by the motion of that single mode. Because ODS does not require curvefitting to estimate the mode shapes, it is a simpler post-processing technique. Instead, the CMIF of the TF is calculated, and peaks in the CMIF are used to predict modal frequencies of the system. The response of the structure at a given frequency is the imaginary component of the TF at that frequency line, so the estimated mode shapes are easily obtained by selecting response motion at the predicted modal frequencies. As in OMA, the excitation is assumed to be distributed and broadband which can lead to missed modes if these assumptions are not met. In addition, the modes are assumed to be adequately spaced such that the response at a predicted modal frequency is dominated by only one mode. If there are two closely spaced modes in the system, ODS is likely to predict a single mode with a shape that is the combination of the two modes.

Operational shapes will be compared to the predicted true modes of the structure by computing the modal assurance criterion (MAC), given in Equation 5 where e_i and e_j are the chosen shape vectors being compared.

$$\boldsymbol{MAC_{ij}} = \frac{\left[\{e_i\}^T\{e_j\}\right]^2}{\left[\{e_i\}^T\{e_i\}\{e_j\}^T\{e_j\}\right]} \qquad \mathbf{5}$$

MAC values can range from a value of one, which indicates that the two vectors are correlated, to a value of zero, which indicates that the vectors are uncorrelated. The MAC is traditionally calculated between all modes of two data sets, such as between all measured modes and all FEA predicted modes, and presented as a color plot. Values on the diagonal of the color plot are expected to be near one, indicating that the modes between two sets are the same shapes and in the same order. Values off diagonal are expected to be near zero, indicating that the modes of the system are orthogonal to each other.

STRCUTURE CONFIGURATIONS

An academic structure shown in Fig. 1 was designed to respond to dynamic excitation with intermittent impact, thus inducing nonlinear contact. This structure was comprised of five primary components: platforms, springs (essentially columns), blocks, L-brackets, and the impact stack. The impact stack is shown in Fig. 2 and consists of an impact hammer tip, a force gauge, an accelerometer cap, and an accelerometer. The L-brackets and impact stack will be collectively referred to as the impact assembly. The intermittent impact in the structure resulted from the impact stack colliding with the lower L-bracket.

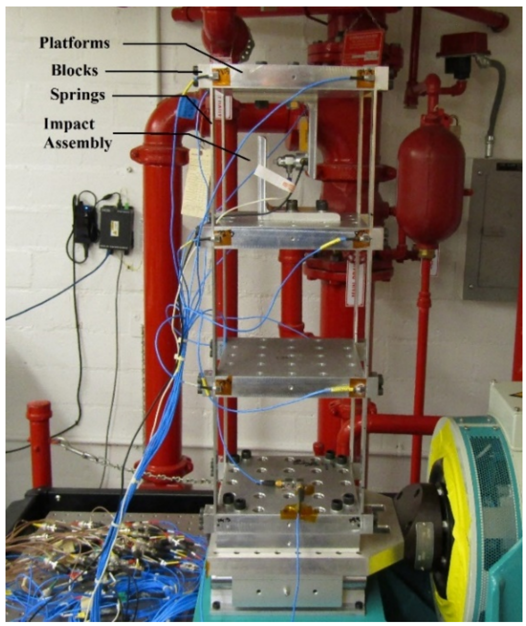

Fig. 1 Structure used to study the effect of nonlinear impact on ODS

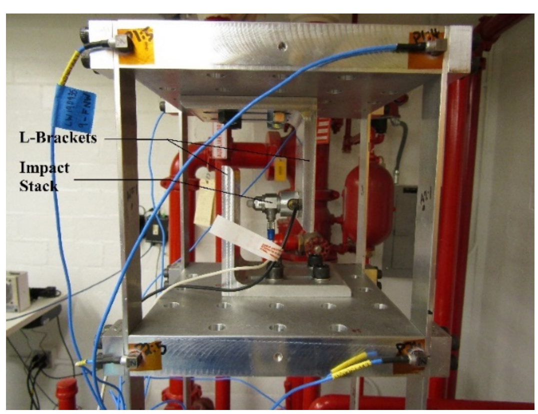

Fig. 2 Impact assembly used to induce intermittent impact in the structure

Throughout this paper, several variations of the structure are referenced. Table 1 identifies all of the variations of the structure discussed in the paper. In addition, Table 1 identifies if modal test data was acquired in those configurations.

Table 1 All configurations of the structure modeled and tested

-	Structure Without the Impact Assembly in Free-Free	Structure With the Impact Assembly in Free-Free	Structure Without the Impact Assembly on the Vibration Shaker	Structure With the Impact Assembly on the Vibration Shaker
FEA	✓	✓	✓	✓
Modal Test	✓	✓	X	X
Vibration Test	X	X	✓	✓

APPROACH

To assess the effect of excitation level on the calculated ODS, the structure was tested at six excitation levels: 0.57 Grms, 0.95 Grms, 1.06 Grms, 1.5 Grms, 2.12 Grms, and 3 Grms. Each test was a random vibration test (5 to 1000 Hz) performed on the same shaker table and instrumented using accelerometers placed at points on the platforms and impact assembly. Internal impact occurred at each excitation level, however higher excitation levels experienced more frequent impact, resulting in increased levels of nonlinear behavior. The measurement locations were chosen to capture the dominant motion of the structure. Fig. 1 shows the structure in the vibration test configuration.

To assess the effect of excitation level on ODS consistency independent of impact, the structure was also tested at the same levels with the impact assembly removed, as seen in Fig. 3. Without the impact assembly, the structure's response was linear.

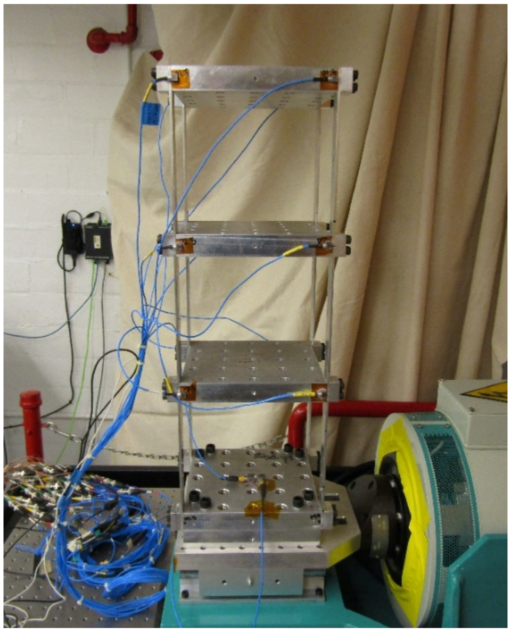

Fig. 3 Structure without the impact assembly on the vibration shaker

To assess the reliability of ODS with intermittent impact, a combination of traditional modal testing and finite element modeling was used to compare the ODS with the predicted true mode shapes of the structure. Modal testing in the same boundary condition as the vibration tests (i.e. structure fastened to the vibration shaker) was not possible due to equipment unavailability, so no test modes are available to compare to the ODS calculated shapes.

Instead, a calibrated finite element model was used to predict the mode shapes of the structure in the vibration test configuration to compare to the ODS. To aid in model calibration, a traditional modal test of the structure was performed in a free-free boundary condition. This test was performed using an impact hammer to excite the structure, and data was acquired using a laser Doppler vibrometer (LDV) to capture the structural response across more of the structure than would be possible using accelerometers. Impact between the impact assembly and the L-bracket was avoided during the modal testing by removing the hammer tip component from the impact stack because the unmeasured impact of the structure would have introduced forcing not accounted for in the calculation of modal parameters.

Model Calibration:

A model of the structure with the impact assembly on the vibration shaker was built to compare predicted modes to the ODS that will be discussed in a later section. The model was built in Cubit [6], and all simulation results were run using the Sierra SD finite element codes [7]. Fig. 4 shows the model with and without the impact assembly.

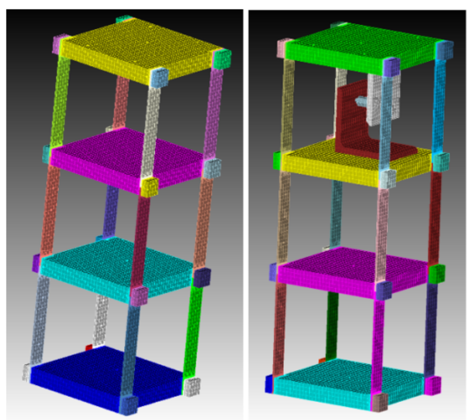

Fig. 4 Finite element model of structure without (left) and with (right) impact assembly

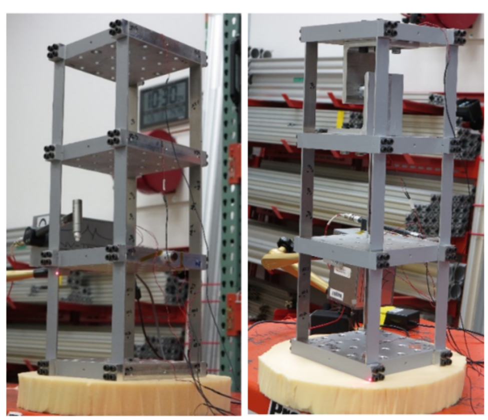

Fig. 5 Free-free modal test setup for structure without (left) and with (right) impact assembly

First, the measured free-free modal test data was used to calibrate the model of the structure in free-free conditions, initially without and then with the impact assembly. The modal test setup for each configuration is shown above in Fig. 5. The MAC plots between the modal test and calibrated model for the free-free structures are shown in Fig. 6.

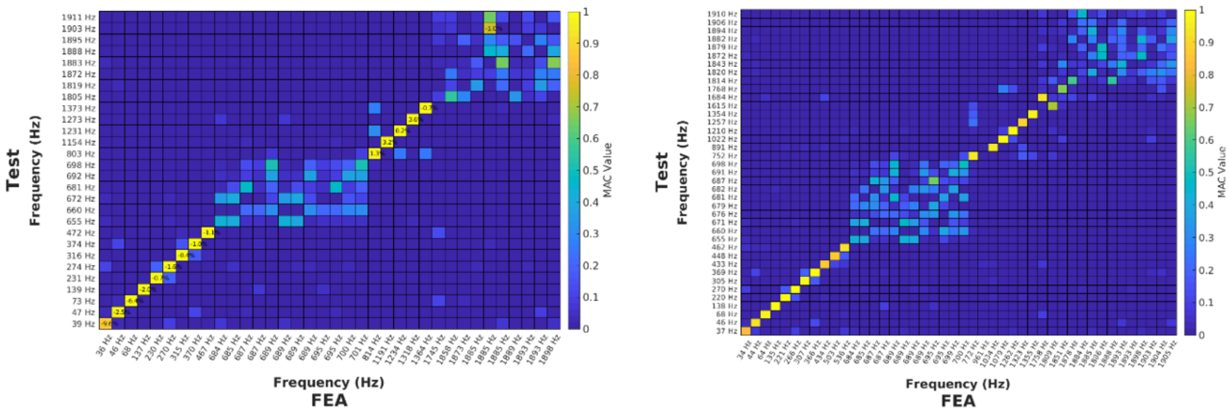

Fig. 6 MAC comparing model without impact assembly (left) and model with impact assembly (right) to free-free modal test

The high MAC values along the diagonals of these two MAC plots indicate that the mode shapes are very consistent between model and test. Additionally, the natural frequencies are within a 5 Hz difference between the finite element model and modal test data, further indicating the model and test are closely matched. The middle and top-right sections of the MAC tables where there are no highly correlated mode shapes are closely spaced, repeated root modes of the springs in the structure. The lack of shape correlation can be explained by observing a typical shape comparison in Fig. 7, which shows the first of the spring bending modes. Only two of the six measured springs show significant response for the selected mode, despite the model predicting that all modes will respond. The model predicts all of the springs in the structure will respond symmetrically at each level. This is unlikely to be captured in test data for three reasons: bolt tightness variations, boundary condition effects, and challenges resolving closely spaced modes from test data. Bolt tightness variations will result in non-symmetric spring responses, which can lead to poor matches between test and data unless every spring connection is individually tuned. The foam boundary condition used to test the structure in "free-free" was not included in the model, so some differences in shape are expected. This is represented in Fig. 8, where you can see the base of the structure in contact with the foam in the test data has less displacement than the top of the structure that is not in contact with the foam. And finally, resolving closely spaced repeated root modes with a single excitation is a known challenge in modal testing [4]. These modes clustered in a small frequency band are each a different combination of which springs are bending and their phase.

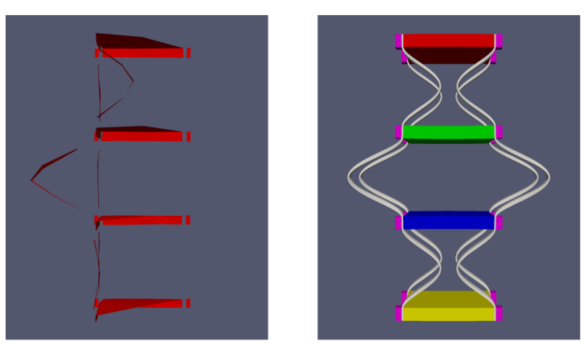

Fig. 7 Free-free modal test (left) and free-free model (right) first spring bending mode

Fig. 8 and Fig. 9 show the mode shapes obtained from the model for the structure in free-free with and without the impact assembly as compared to the corresponding mode shapes from the free-free modal testing. The primary discrepancy of note is the lack of symmetry in the 39 and 36 Hz mode shapes in Fig. 8, likely resulting from the non-zero stiffness of the foam the structure was sitting on in the free-free modal testing. Ultimately, these models were considered to be adequately calibrated to the test data and were then updated to have a fixed-base boundary condition. The modes from these fixed-base models were then used to compare to the shapes obtained from ODS. The model fixed base boundary condition is more severe than the actual shaker boundary condition shown in Fig. 1, so some differences are expected between the model predicted modes and the ODS obtained shapes and frequencies.

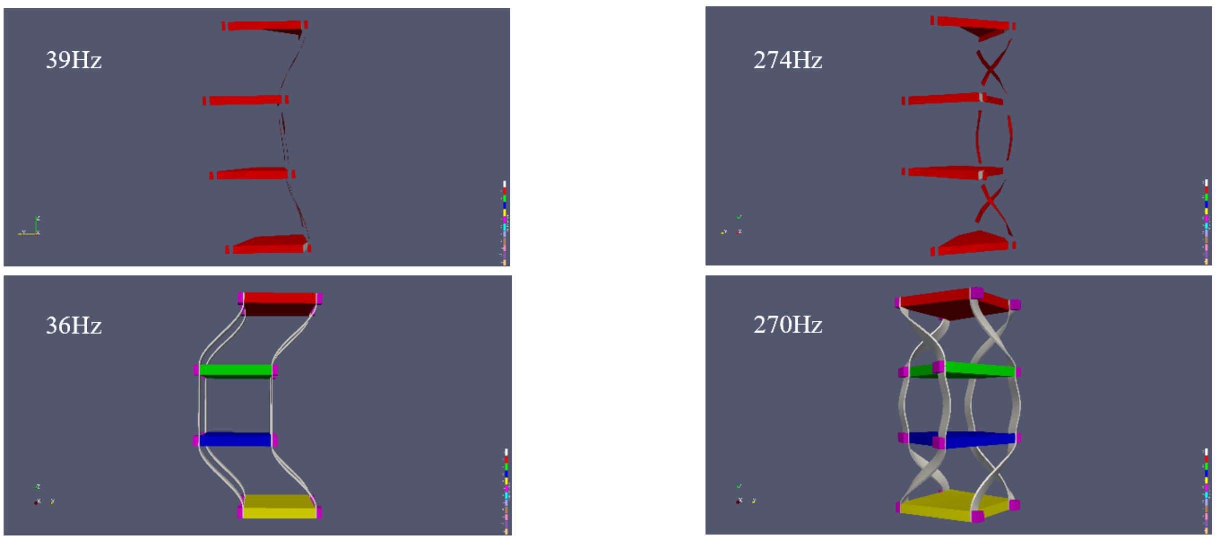

Fig. 8 Mode shape comparisons for structure without impact assembly

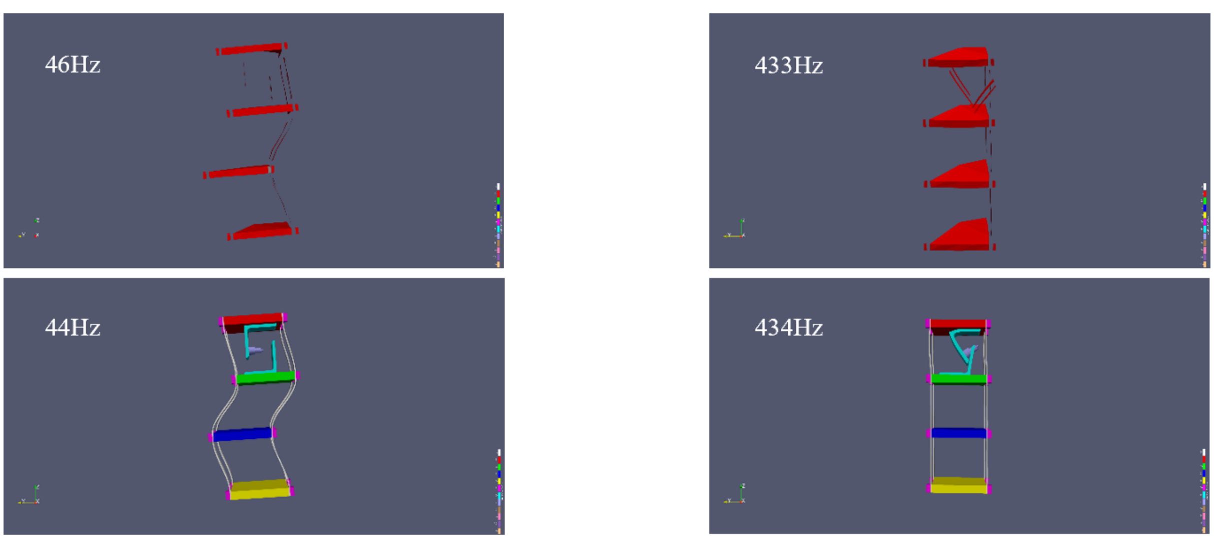

Fig. 9 Mode shape comparisons for structure with impact assembly

ODS:

ODS were obtained for each excitation level using the acceleration time history data from the vibration tests. The selected reference node for the ODS procedure was located at the base of the structure, closest to the shaker. It was chosen because it most closely represented the input motion which helped to better capture the relative motion of the rest of the structure. Resonant frequencies were estimated by computing the CMIF of the TFs calculated using the reference node, and then using the peak-picking function in Matlab to select peaks in the CMIF. Shape orthogonality was also enforced by computing a MAC of all shapes at each possible peak in the CMIF. If several peaks had a similar shape, the frequency with the highest CMIF value was chosen to be the estimated modal frequency. All other possible modal frequencies with that similar shape were not selected.

The primary method for comparing the mode shapes obtained from ODS and those obtained from the calibrated finite element model was also by using the MAC. Additionally, the MAC was used to compare the ODS obtained mode shapes at different excitation levels to more closely inspect the sensitivity of ODS to excitation level.

RESULTS

Once the ODS procedure was conducted for each excitation level, the mode shapes from each excitation level were compared to one another in a “bigMAC” plot. With and without impact, the modal consistency across excitation levels is high overall. For both testing configurations (with and without the impact assembly), the primary region of low consistency is in the low frequency range less than 20 Hz. Fig. 10 and Fig. 11 show these bigMAC plots for the structure with and without the impact assembly, respectively.

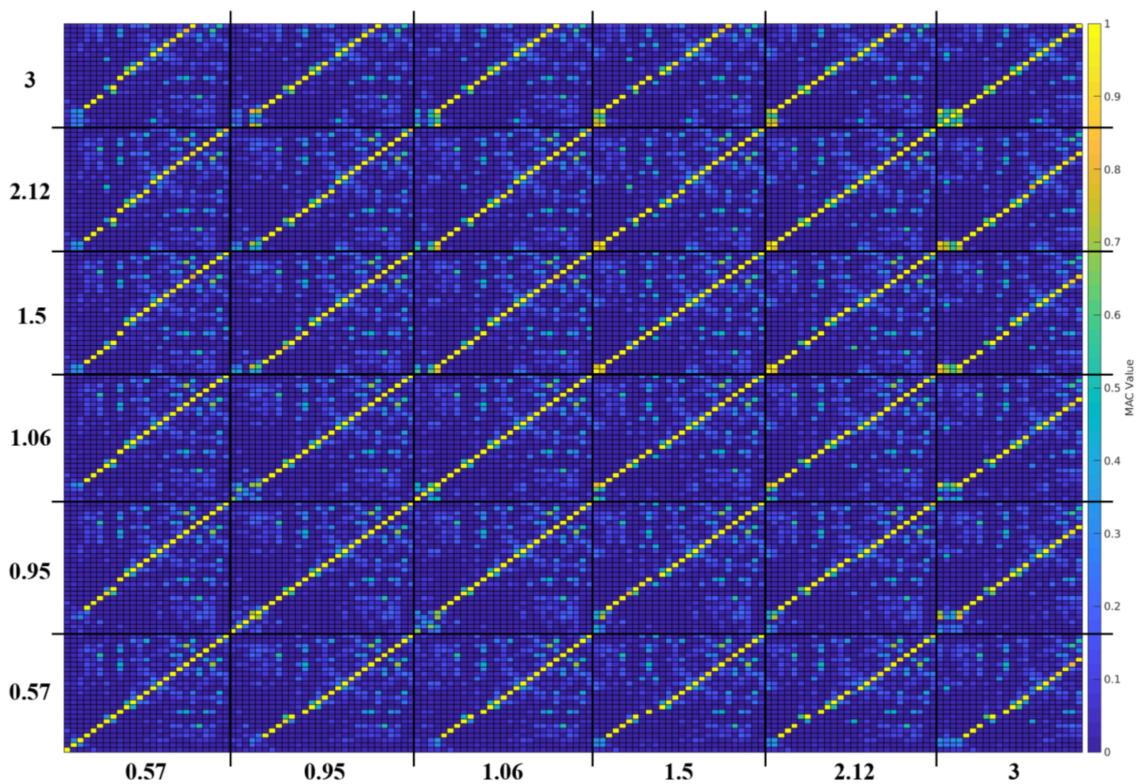

Fig. 10 MAC plot comparing consistency of ODS mode shapes across excitation level for structure with impact assembly

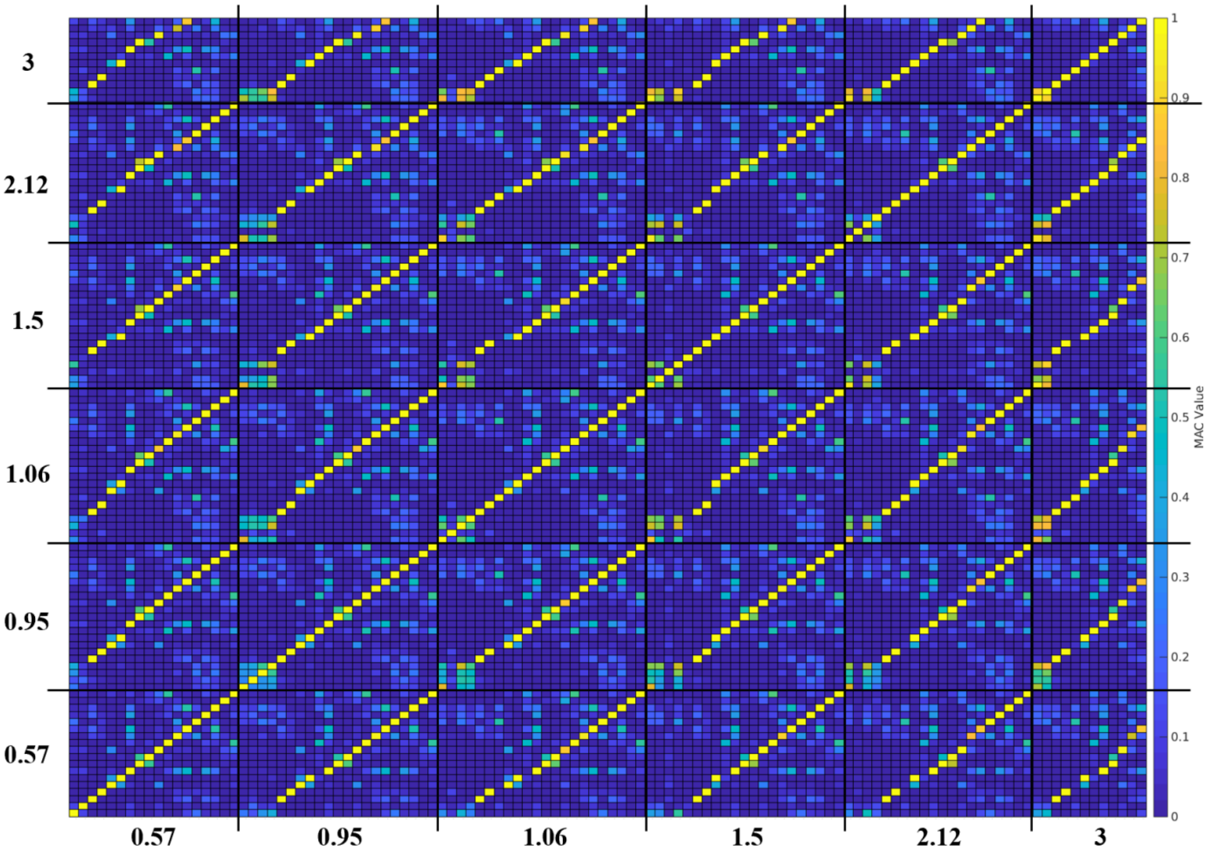

Fig. 11 MAC plot comparing consistency of ODS mode shapes across excitation level for structure without impact assembly

Next, the ODS obtained modes are compared to the finite element model obtained modes in Fig. 12 – Fig. 17. Additionally, the ODS obtained natural frequencies are compared to the finite element model modal frequencies in

Table 2 and Table 3. Bolded values in these tables indicate high modal consistency between the finite element model and ODS at that frequency.

Table 4 and Table 5 show only the frequencies for which the finite element model predicts mode shapes, with blanks in the ODS columns when ODS fails to predict that mode shape.

Across all excitation levels and both structure configurations, the ODS procedure resolves a handful of mode shapes and their corresponding frequencies with high consistency with the finite element model. There is little difference in consistency from one excitation level to next, however the higher excitation levels tend to have mode shapes exhibiting slightly more consistency and natural frequencies slightly closer to those shapes and frequencies of the finite element model. A possible explanation for this observation is that having higher levels of excitation could be increasing the response of the structure, making response peaks more prominent relative to noise. It is also worth noting that the ODS method picks up the first L-bracket bending mode for all permutations of excitation level and structure configuration. Some selected mode shapes highlighting the overall consistency between the ODS and finite element model obtained mode shapes are shown in Fig. 18 – Fig. 20.

Table 2 Resonant frequencies of structure with impact assembly from finite element model and ODS

-	Finite Element Model	ODS: 0.57Grms	ODS: 0.95Grms	ODS: 1.06Grms	ODS: 1.5Grms	ODS: 2.12Grms	ODS: 3Grms
Modal Frequencies (Hz)	**10**	8	7	6	**10**	**10**	9
	30	11	8	9	**30**	**30**	10
	43	12	12	**11**	**44**	**44**	**11**
	62	**30**	13	**30**	**80**	**80**	**30**
	80	**43**	**30**	**43**	174	175	**44**
	192	**80**	**44**	**80**	**194**	**194**	**80**
	235	177	**80**	175	225	225	174
	302	**195**	175	**195**	**236**	**236**	**193**
	355	**236**	**195**	227	275	301	224
	460	**303**	228	**236**	**302**	329	**236**
	486	333	**236**	**302**	331	**353**	**301**
	531	**357**	**302**	331	**355**	408	326
	684-700	406	332	**356**	409	443	353
	711	**480**	**356**	408	**478**	**478**	454
	821	516	408	**480**	516	517	**476**
		630	**479**	518	630	628	517
		683-701	518	628	683-701	683-701	629
		752	628	683-701	**751**	**750**	683-701
		938	683-701	**752**	942	934	**748**
			752	938			936
			937				

Table 3 Resonant frequencies of structure without impact assembly from finite element model and ODS

-	Finite Element Model	ODS: 0.57Grms	ODS: 0.95Grms	ODS: 1.06Grms	ODS: 1.5Grms	ODS: 2.12Grms	ODS: 3Grms

Natural Frequencies (Hz)	**11** **31** **45** 66 **81** **198** **237** **315** 361 510 684-700 803 **831**	9 13 **31** **46** 177 **200** **238** **313** 337 452 634 688-701 **846** 947	8 11 12 **32** **46** 175 **199** 230 **238** **312** 335 450 634 688-701 **846** 946	8 11 12 **31** **46** **82** 175 **199** 230 **238** **311** 335 409 633 688-701 **843** 946	9 10 12 19 **32** **46** **199** 229 **238** **311** 332 451 631 688-700 **845** 945	8 11 12 **31** **46** **82** **198** 228 **239** **310** 330 453 688-700 **844** 940	**11** **31** **46** **82** **198** 227 **239** **311** 452 688-700 **842** 940

Table 4 High consistency frequencies for structure with impact assembly

-	Finite Element Model	ODS: 0.57Grms	ODS: 0.95Grms	ODS: 1.06Grms	ODS: 1.5Grms	ODS: 2.12Grms	ODS: 3Grms
Modal Frequencies (Hz)	10			11	10	10	11
	30	30	30	30	30	30	30
	43	43	44	43	44	44	44
	62						
	80	80	80	80	80	80	80
	192	195	195	195	194	194	193
	235	236	236	236	236	236	236
	302	303	302	302	302		301
	355	357	356	356	355	353	
	460						
	486	480	479	480	478	478	476
	531						
	684-700	683-701	683-701	683-701	683-701	683-701	683-701
	711	752	752	752	751	750	748
	821						

Table 5 High consistency frequencies for structure without impact assembly

-	Finite Element Model	ODS: 0.57Grms	ODS: 0.95Grms	ODS: 1.06Grms	ODS: 1.5Grms	ODS: 2.12Grms	ODS: 3Grms
Natural Frequencies (Hz)	11						11
	31	31	32	31	32	31	31
	45	46	46	46	46	46	46
	66						
	81			82		82	82
	198	200	199	199	199	198	198
	237	238	238	238	238	239	239
	315	313	312	311	311	310	311
	361						
	510						
	684-700	688-701	688-701	688-701	688-700	688-700	688-700
	803						
	831	846	846	843	845	844	842

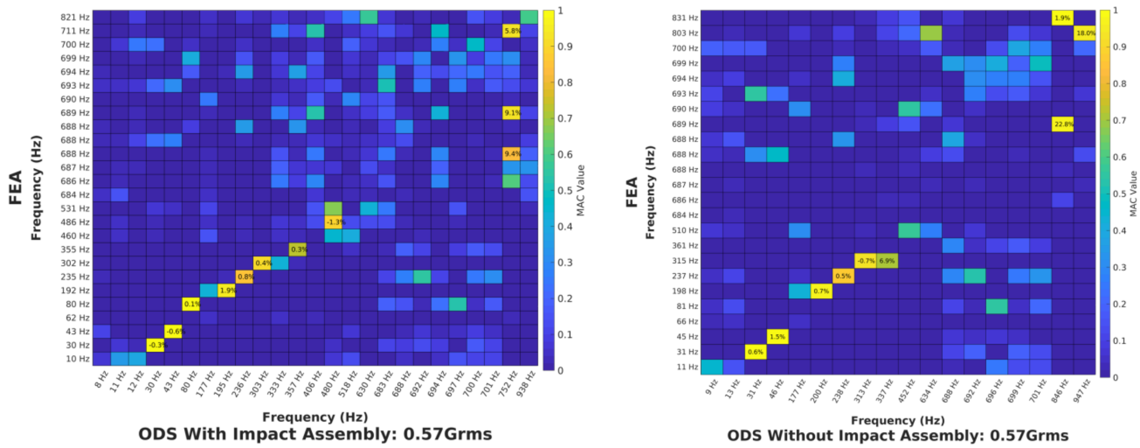

Fig. 12 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 0.57 Grms

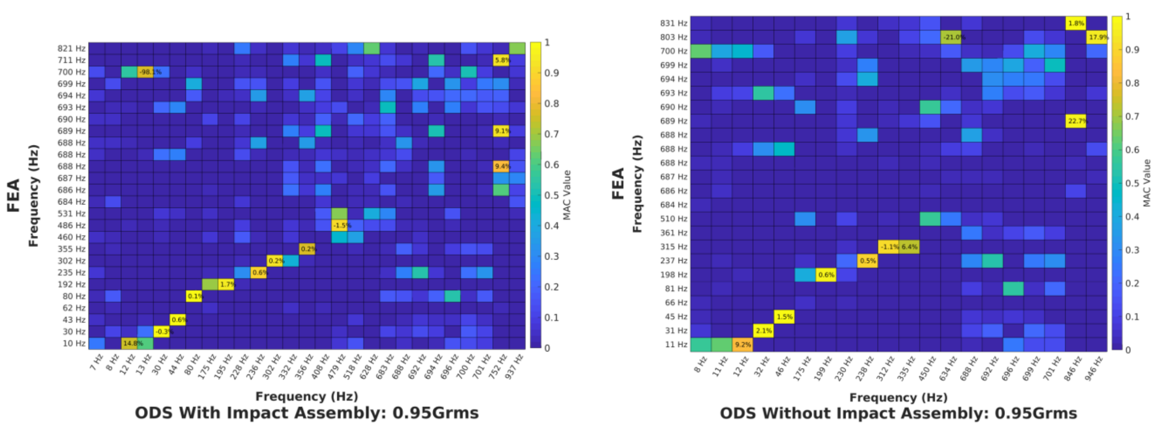

Fig. 13 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 0.95 Grms

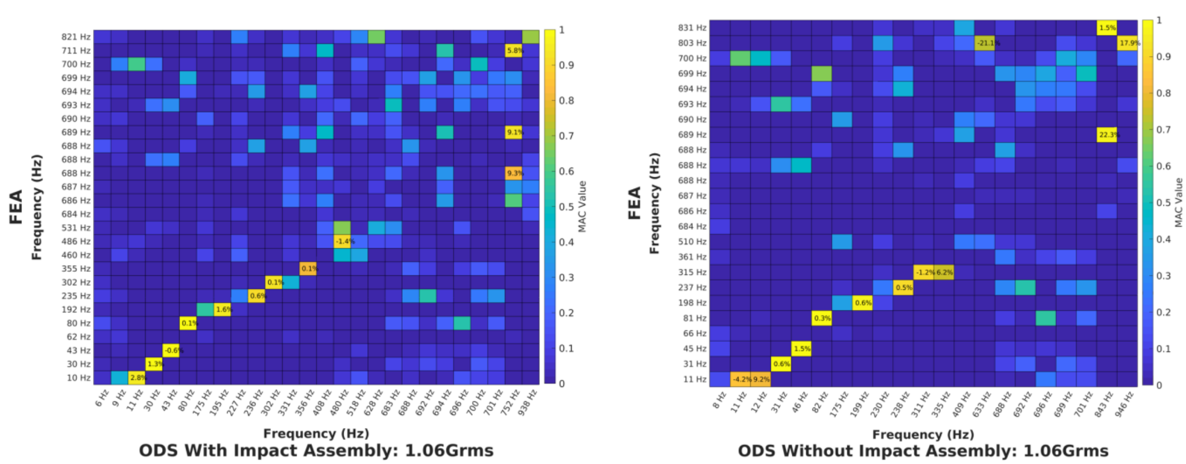

Fig. 14 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 1.06 Grms

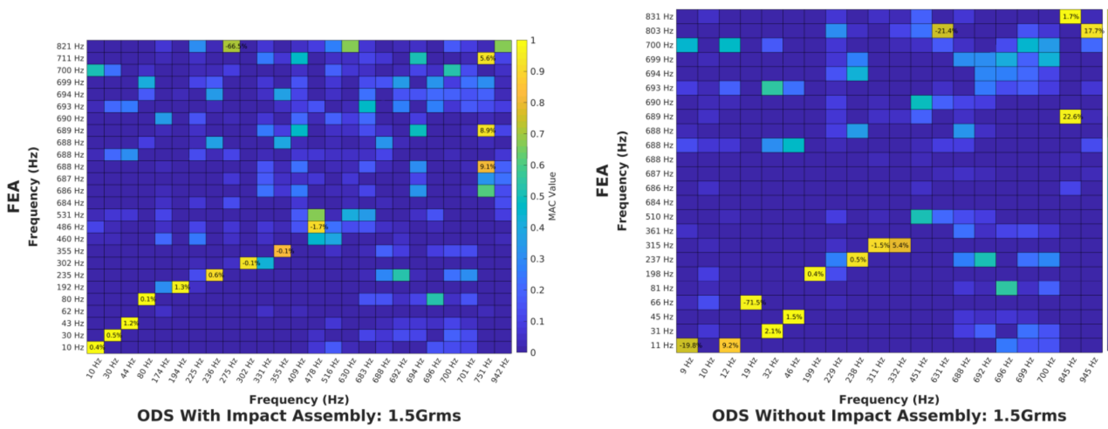

Fig. 15 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 1.5 Grms

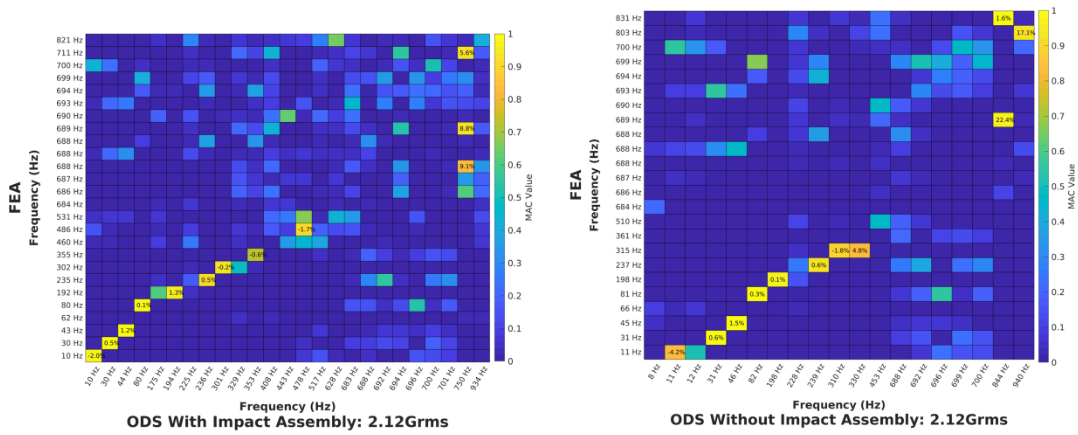

Fig. 16 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 2.12 Grms

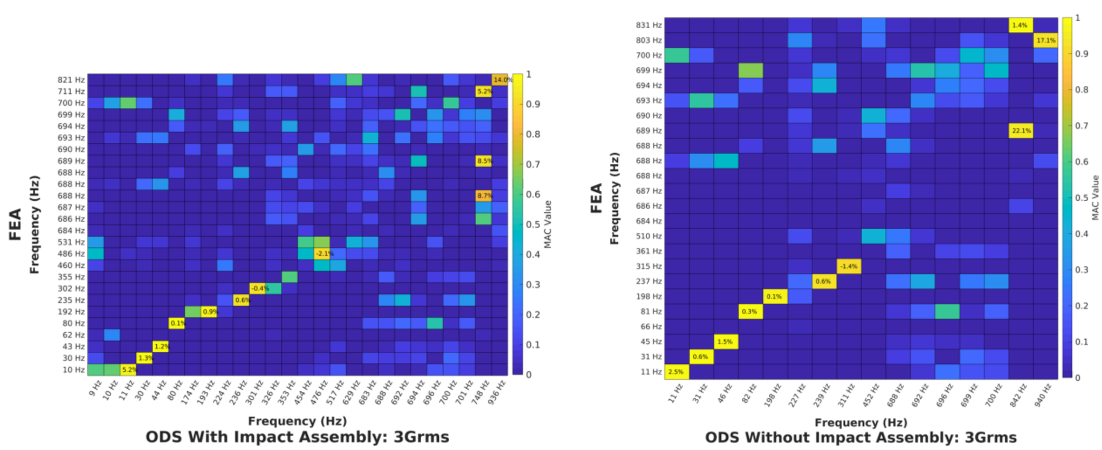

Fig. 17 MAC plots comparing ODS to finite element model mode shapes for structure with (left) and without (right) impact assembly at 3 Grms

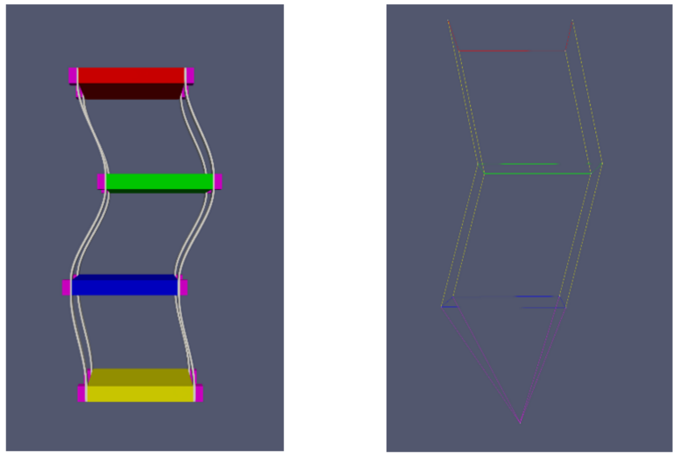

Fig. 18 Finite element model mode shape at 45 Hz (left) and ODS at 46 Hz (right) for 0.95 Grms

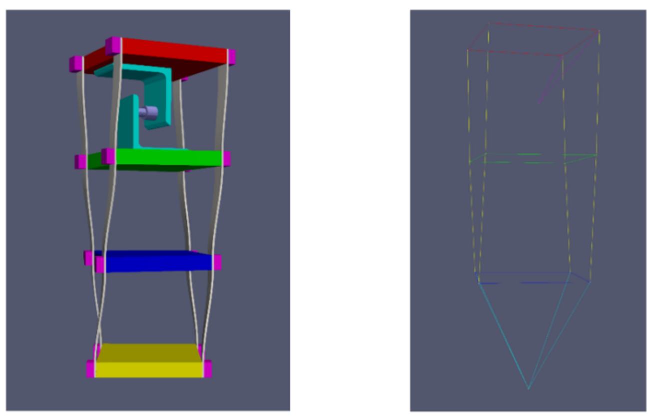

Fig. 19 Finite element model mode shape at 80 Hz (left) and ODS at 80 Hz (right) for 1.06 Grms

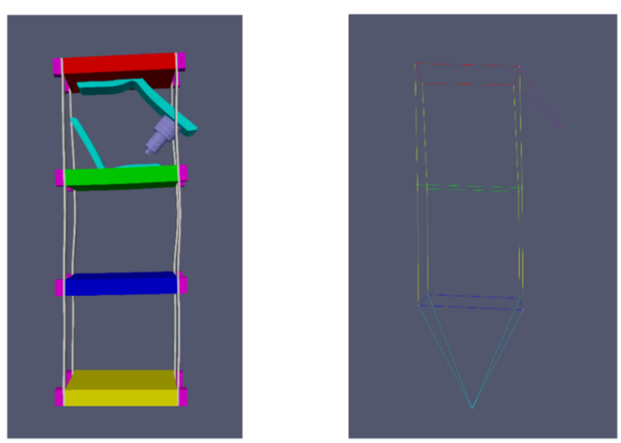

Fig. 20 Finite element model mode shape at 486 Hz (left) and ODS at 486 Hz (right) for 2.12 Grms

Across all excitation levels, the ODS procedure picks up modes that don't exist in the finite element model. These are a result of peaks in the TF's that are likely noise and not truly representative of actual modes. Another discrepancy worth noting is that the ODS procedure fails to resolve the mode at 66 Hz for the structure without the impact assembly and 62 Hz for the structure with the impact assembly across all excitation levels. The missed mode shape is an out-of-plane rocking motion, as can be seen in Fig. 21, while the excitation for the vibration testing was only in-plane. Since operational techniques only predict modes that are excited, this and other unexcited modes are unable to be estimated by ODS, as is emphasized in

Table 4 and Table 5.

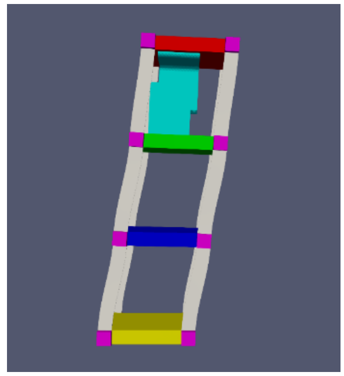

Fig. 21 Finite element model mode shape at 62 Hz showing out of plane response

Despite introducing observable nonlinear behavior, the addition of the impact assembly does not result in a significant decrease in consistency with the model obtained modes. In fact, at some excitation levels, it results in notably higher consistency. A possible explanation for this is that the induced impact provides more excitation to the structure that wouldn't have been induced by base excitation alone. The other primary difference between the configurations with and without the impact assembly is an increase in TF peaks for the structure with the impact assembly that are not representative of modes in the finite element model. This is somewhat expected since the structure with the impact assembly is essentially subjected to additional impulse forcing upon each impact. Ultimately, the impact assembly appears to result in somewhat better prediction of modes obtained from the finite element model, while also predicting more modes that likely are not actually modes.

CONCLUSION

Without traditional modal testing to use as a basis for assessing the accuracy of the ODS obtained modal parameters, no conclusive statements can be made regarding the accuracy of the ODS obtained shapes and frequencies. The finite element model can be used to estimate the consistency but is not a perfect representation of the structure and cannot be used as a "truth" estimate of the actual mode shapes and frequencies. Most importantly, the model's fixed base approximation fails to represent the dynamics of the vibration shaker used for the vibration testing. Despite these limitations, there are some important takeaways from this research regarding the effect of excitation level and intermittent impact on the ODS obtained modal parameters. Especially for the structure without the impact assembly, as excitation level increases, the modal consistency appears to increase slightly. Without more reference data, concluding that increasing excitation level results in higher modal consistency would be unreasonable, however it is a noteworthy observation worth further investigation. The addition of the impact assembly appears to result in somewhat higher modal consistency, however it also results in noisier data, leading to more peaks that aren't representative of modes. This research suggests that the usage of the ODS method for modal parameter estimation is promising for structures with varying levels of excitation and known nonlinear impact. However, further research is needed to fully understand the limitations of using ODS as a substitute for modal data.

ACKNOWLEDGEMENTS

The authors would like to especially thank Dan Roettgen, Bryan Witt, and Anthony Jaramillo for acquiring and sharing the vibration and modal data of the academic structure used in this paper.

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