Full Circle Mechanical Dynamic Characterization Including Experimental Modal Analysis and Finite Element Analysis

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Abstract: During operation, it was observed that a specific mechanical system experienced undesirable vibration and it became necessary to understand and mitigate this phenomenon. This document investigates the tools, methodology, and results of the dynamic characterization of the system. The characterization makes use of the experimental modal analysis (EMA) methods of single input multiple output (SIMO) and single input single output (SISO). The validity of the theory of reciprocity is confirmed to minimize measurement error, cost, and time of repeat testing. Finite element analysis (FEA) is used in choosing transducer and modal impact locations to adequately characterize the system. Single degree of freedom (SDOF) and multiple degree of freedom (MDOF) curve fitting is used to fully characterize the system's mode shapes and natural frequencies. The EMA characterization results are used to modify and validate the FEA model so that FEA can be used to model potential structural modifications to the system to mitigate the undesirable vibration. Structural modifications are chosen, implemented, and their effectiveness is quantified using EMA. A qualitative evaluation of the methodology of FEA validation by EMA and tuning of the model to match the experimental results is discussed.

Keywords: Case Study, Non-destructive Testing, Signal Analysis

INTRODUCTION

Undesirable vibration occurred during operation of a mechanical system and investigation was necessary to understand the event. A combination of experimental modal analysis (EMA) and finite element analysis (FEA) was successfully used in a feedback loop to improve the efficacy of the investigation. FEA provided insight into preferred transducer and modal impact locations needed to utilize single input multiple output (SIMO) and single input single output (SISO) EMA methods, and the results of those tests were then used to refine the FEA model as needed. Modifications were added to the mechanical system to alter its natural frequencies. EMA and FEA was performed on these modifications and compared to ensure similar trends in frequency shifts.

INITIAL EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis was performed using a modal impact hammer, accelerometers, and data acquisition equipment. For further information on performing

EMA, refer to the Shock and Vibration Handbook [1]. Single input multiple output testing was performed first, making use of transducers already in place in a single axial line along the cylinder measuring radially. The primary function of these transducers was system condition monitoring during normal operation, but, in this case, they were used in conjunction with a modal hammer while the system was not operational. The frequency response functions (FRF) obtained from the SIMO testing were preliminarily curve fit using a single degree of freedom (SDOF) curve fitting software to produce representations of the system's mode shapes. The investigation of these mode shapes indicated that there were not enough measurements made to characterize the system because there were shapes more complicated than the standard rigid body modes and bending modes. A finite element analysis can be used to determine the resonant frequency and mode shapes of a system and provide insight to measurement locations.

FEA DESCRIPTION

ANSYS mechanical was used to conduct the modal analysis of the mechanical system [2]. A modal analysis calculates the resonant frequencies of a structure based on its mass and stiffness assuming no excitation. The advantage of a simulation is the ability to make changes to the structure and quickly determine how these changes affect its resonant frequencies. Also, the simulation produces mode shapes for each of the resonant frequencies which aids in determining whether they are rigid bodies, bending, or shell modes.

A modal analysis of the mechanical system was created to both determine optimal locations for modal measurements and provide insight into how slight modifications to the test stand may affect its resonant frequencies. The solid models used to create the drawings for fabrication were also used to develop the finite element analysis model. These models were simplified for use in the FEA model to increase computational efficiency while not reducing accuracy. These simplifications include filling minor holes in components and removing noncritical radii and chamfers. Some components were excluded such as minor tubing and small fastener hardware that were not considered structurally significant. The structure consists of primarily isotropic metallic materials.

There are several bolted sections of the mechanical system. They are connected through bolts of several sizes and can be torqued to various values. These torque values were used to adjust the model to match measured data. Connections between components consisted of either frictional or bonded contact. Bonded contact was used between small internal components and for bolts that were threaded into attaching components. Frictional contact was used between major bolted connections and could also be adjusted to match measured results. The model used for the modal analysis is shown in Figure 1.

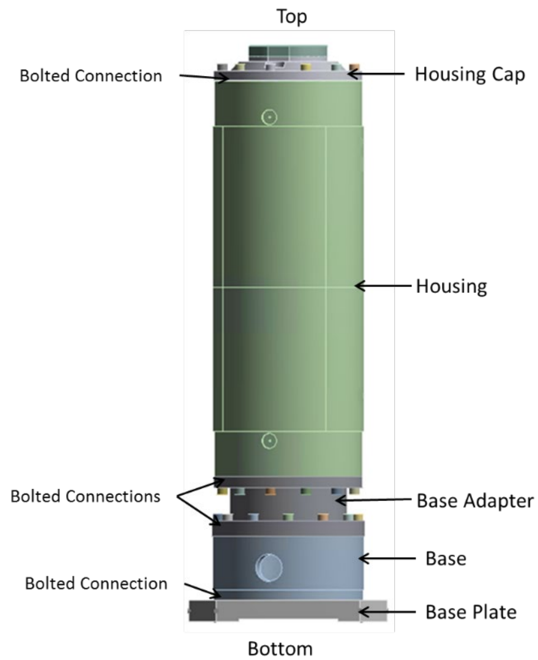


Figure 1: Mechanical System Model Description

Three orthogonal springs were added to each location of the base plate where it connected to the ground. Each of the three springs were aligned with the global X, Y, Z coordinate axes. The spring stiffness was adjusted to aid in aligning the model's resonant frequencies with the measured data. A coarse mesh was applied to the structure consisting of 2nd order tetrahedral elements. Figure 2 shows the spring locations anchoring the mechanical system to ground and analysis mesh.

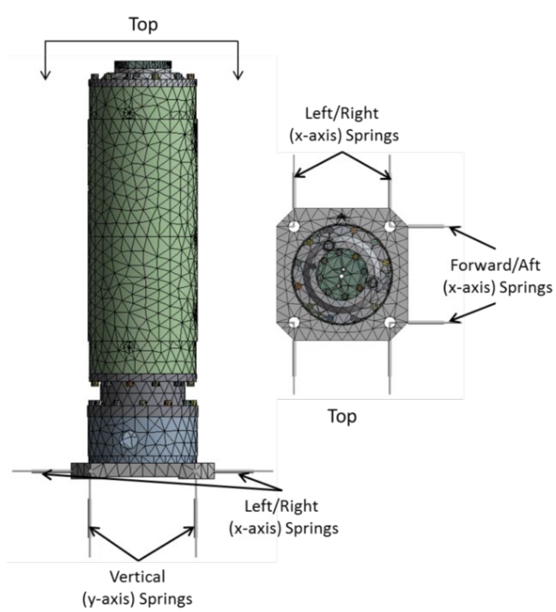


Figure 2: Model mesh and spring to ground boundary conditions

MODIFIED TRANSDUCER PLACEMENT AND IMPACT LOCATIONS

Initial stiffness values were assigned to the ground springs and bolts for the first modal analysis. The intent of this analysis was to determine accelerometer locations on the housing that would be suitable for determining mode shapes. The key at locating these accelerometers was to determine the antinodes of the structure that captured the important mode shapes. The initial FEA model exposed several shell modes that occurred in the frequency range of interest and identified more measurement locations than used for the initial SIMO EMA are needed to differentiate the shell modes from bending modes. Important mode shapes include; 1st bending, shell, and 2nd bending. Figure 3 shows the suggested minimum accelerometer locations determined by the initial modal analysis and the actual number of locations used in the roving hammer tests. Further points were added during the roving hammer empirical modal tests to increase mode shape fidelity.

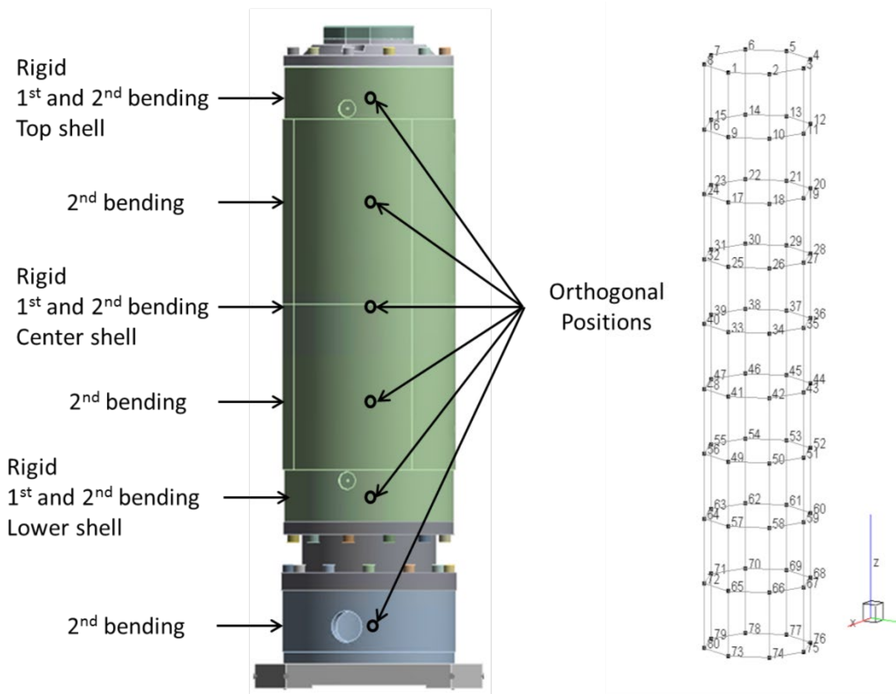


Figure 3: : Suggested accelerometer locations (left) and actual location for the roving hammer empirical modal tests (right)

ROVING HAMMER EMPIRICAL MODAL ANALYSIS

In order to accurately differentiate between and clearly display the mode shapes within the frequency range of interest, ten concentric circles of eight equally spaced radial measurement locations were excited using the SIMO roving hammer technique. Before measuring all eighty points, the measured response locations from the initial SIMO testing were impacted and the response was measured at one of the initial impact locations. The frequency response functions generated from the initial SIMO measurement closely matches that of the SISO roving hammer measurement, thus, the theory of reciprocity is validated for use with this measurement. Figure 4 shows the normalized FRF of the standard mechanical system. This significantly sped up the measurement process as one modal impact hammer and one fixed location accelerometer were used to collect the measurements rather than moving a column of accelerometers for each measurement. Each of the eighty points were impacted radially three times to collect an average measurement at each location. A multiple degree of freedom (MDOF) curve fitting software generated the mode shapes within the frequency range of interest of the mechanical system. The resonant frequencies and their respective mode shapes were then compared with the FEA results.

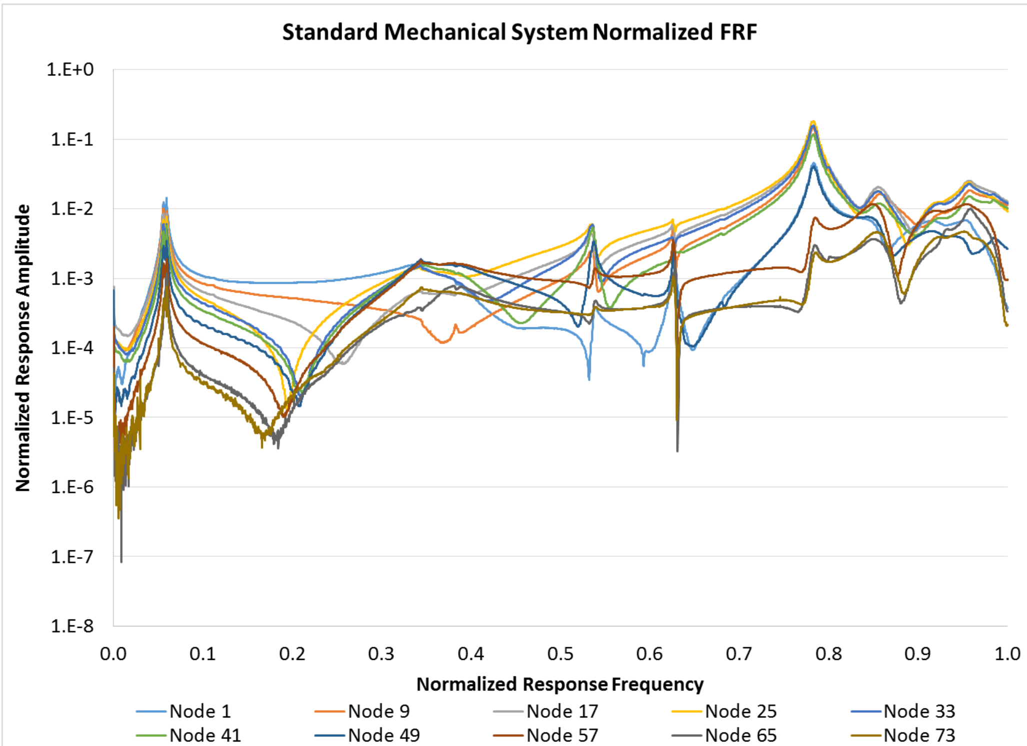


Figure 4: Normalized FRF of the standard mechanical system.

FEA MODIFICATION AND VALIDATION

Several parameters in the FEA model were adjusted to obtain a reasonable comparison with the measured data. The springs at the base of the model were adjusted to match the stiffness of the floor beneath the physical mechanical system. These primarily adjusted the rigid body modes. The bolt torque and friction coefficients at the major joints were adjusted to match the upper and lower shell and bending frequencies more accurately. After several iterations of adjusting bolt torque and fiction coefficient, it was determined that removal of the bolt torque and bonding the major joints of the mechanical system matched the roving hammer results best.

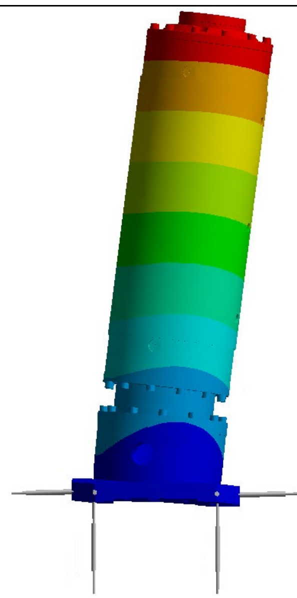
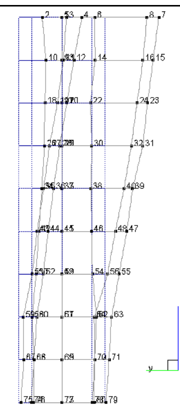
Once the model predicted the major modes of the mechanical structure, a total of 25 modes were calculated and the resulting frequencies were normalized across the normalized range of interest from 0 and 1. Table 1 lists the resulting modes and a description of the mode shape.

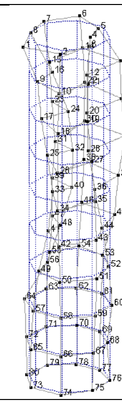
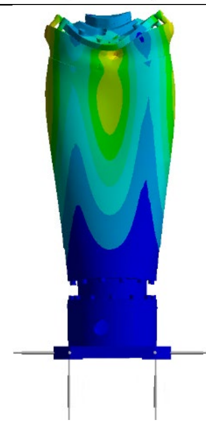
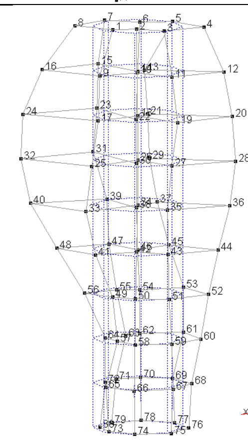
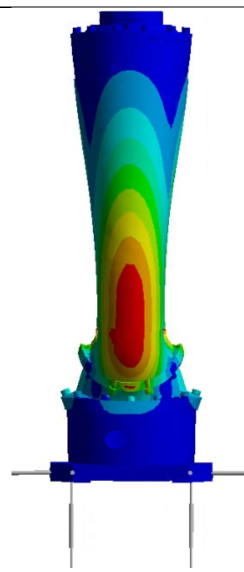
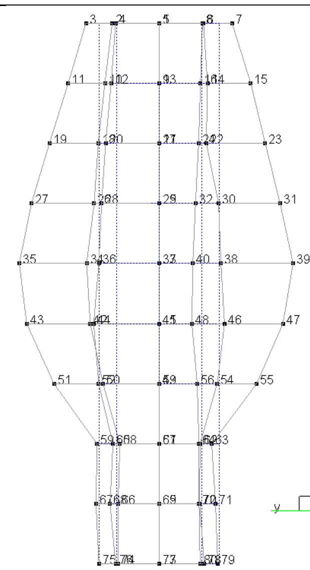
Table 1 Modes of interest of the mechanical system.

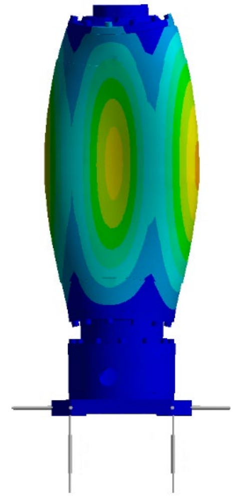
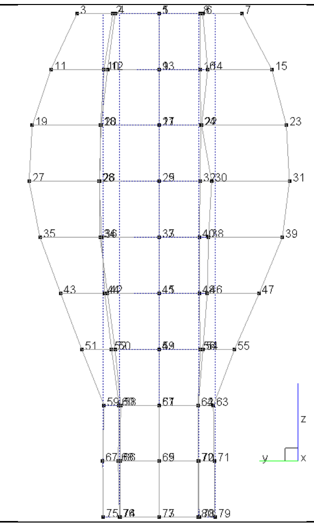
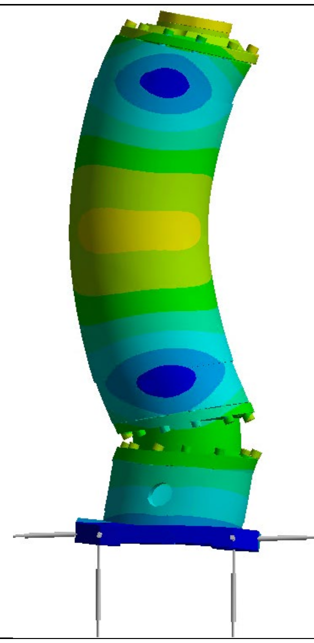
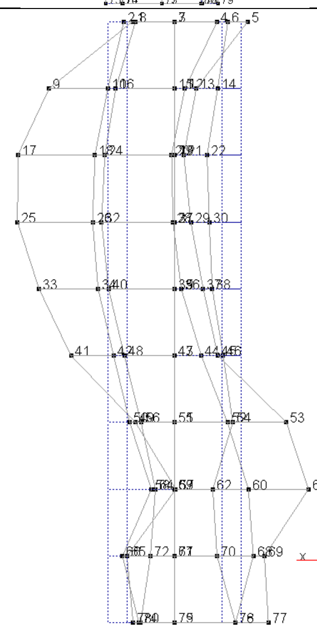
Mode	Model Frequency	Measured Frequency	Shape Description
1	0.040	0.055	Rigid
2	0.040	0.059	Rigid
8	0.305	0.346	1st Bending
10	0.470	0.417	Upper End Shell
12	0.518	0.536	Lower End Shell
19	0.800	0.835	Central Shell
21	0.835	0.855	2nd Bending

Table 2 shows a representation of the major mode shapes from the model analysis and the roving hammer test. Note that the deformed shapes are exaggerations of the actual mode shapes.

Table 2: Major mode shapes of the test stand.

Mode Shape Description	Normalized Frequencies	Model Mode Shapes	Measured Mode Shapes
Rigid	0.040 - Modeled 0.055 - Measured		

1st Bending	0.309 – Modeled 0.346 – Measured		
Upper Shell	0.475 – Modeled 0.417 – Measured		
Lower Shell	0.518 – Modeled 0.536 – Measured		

Central Shell	0.798 – Modeled 0.783 – Measured		
2nd Bending	0.838 – Modeled 0.855 – Measured		

SYSTEM MODIFICATION

The modal analysis adds value to the modal testing results by aiding in determining mode shapes and performing quick parametric studies that show how the resonant frequencies of the structure shift due to geometry modifications. These geometry modifications include lengthening sections of the mechanical system, adding mass, or adding stiffeners.

A series of changes were implemented to the mechanical system both in the FEA model and the physical system. The first was an addition of a steel strap located near the center of the system. The intention of the strap was to shift the frequencies of the central shell modes. The second was an extension of the base component which was intended to shift the first bending frequency. Refer to Figure 1 for the location of the base component. Modal analysis was performed for each change and compared back to the standard system analysis and the EMA results.

The EMA roving hammer results of the mechanical system, shown in Figure 5, with a lengthened base primarily shifted the first bending mode down by 0.1 normalized frequency units, but the second bending mode shifted slightly up and its damping decreased. The FEA results showed a similar shift in the first bending mode and shows similar trends in behavior for the higher frequency modes. Table 3 lists the normalized 1st bending, center shell, and 2nd bending frequencies for the standard mechanical and lengthened base FEA models.

The EMA roving hammer results of the mechanical system, shown in Figure 5, with a steel strap near centrally located on the housing primarily showed that the modification was not as effective. The central shell shifted down slightly by about 0.02 normalized frequency units rather than up and the mode was split into two modes due to non-symmetric stiffness. The shift down in frequency is a result of the straps preloading the housing reducing its circumferential strength. The FEA results show a very little change in frequencies as indicated by the EMA tests. Table 3 lists the normalized 1st bending, center shell, and 2nd bending frequencies for the standard mechanical and steel strap FEA models.

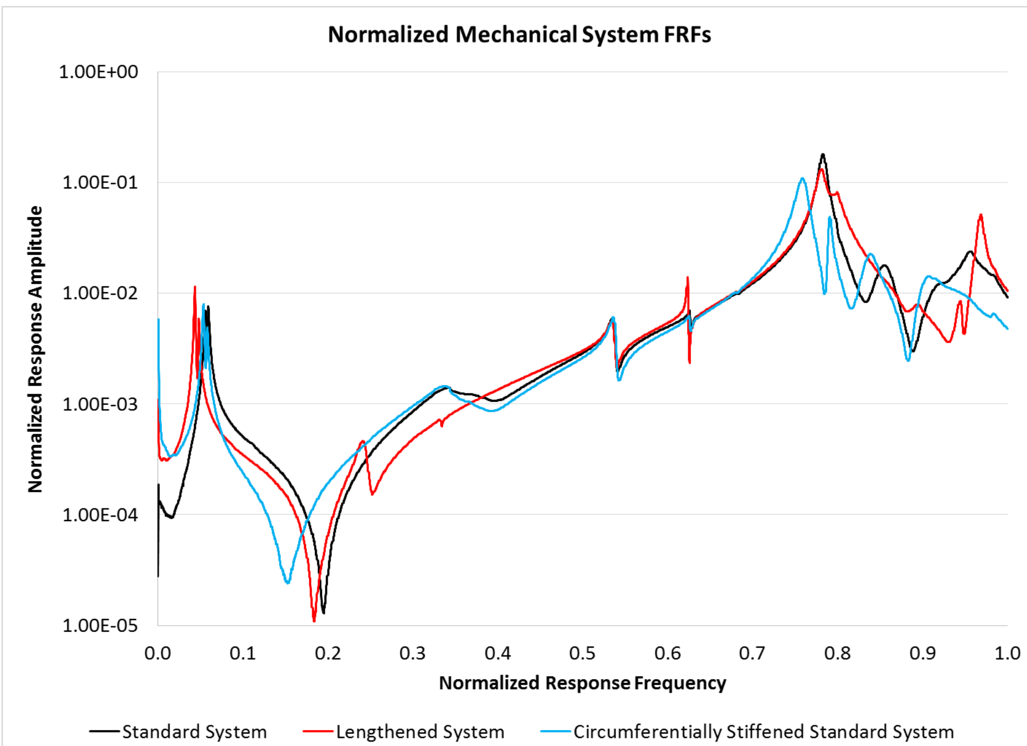


Figure 5: Normalized FRFs of the standard system and modified system

Table 3: FEA normalized frequencies for the standard system and the two variations.

Mode Shape	Standard System	Lengthened System	Circumferential Strap
1st Bending	0.309	0.247	0.312
Central Shell	0.798	0.798	0.809
2nd Bending	0.838	0.782	0.792

CONCLUSION

This paper has presented a successful implementation of a feedback loop using EMA and FEA in concert to increase the effectiveness of an investigation into aberrant performance in a mechanical system. FEA provided recommended transducer and modal impact locations for SIMO and SISO EMA methodologies. The combination of analytical and experimental approaches resulted in improved understanding of the mechanical system and allowed appropriate actions to be taken to address the undesirable vibrations.

FUTURE WORK

Recommended future work focuses on improving the fidelity of both the analytical and experimental analyses. An FEA harmonic analysis on the mechanical system would add information about the severity of each resonant frequency. An FRF over the entire frequency domain could be created from the harmonic analysis and compared with the measured data. The advantage of the harmonic analysis gives the ability to inflict a sinusoidal force to the mechanical system and predict the response at specific locations. The location of the force and locations of response could be the same as the location of the strike of the modal hammer and the locations of the accelerometers.

EMA testing of the axial and torsional modes would increase the fidelity of the analysis by characterizing their effect on the mechanical system and identifying their presence in the radially excited data already collected.

Other structural modifications could be considered and tested to increase the vibration mitigation of the desired modes.

REFERENCE

[1] *Shock and Vibration Handbook*, 4^{th} Edition. Editors Cyril M. Harris, Charles E. Crede, McGraw-Hill Professional Publishing, 1995, chapter 21.

[2] ANSYS Mechanical Version 19.