# Modal Test and Model Correlation of NASA Plum Brook Station Mechanical Vibration Facility Head Expander – Lessons Learned from the Perspective of an Early-Career Engineer

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**ABSTRACT**

In preparation for the Sierra Nevada Corporation's (SNC) Dream Chaser spacecraft vibration test campaign at the Mechanical Vibration Facility (MVF) at NASA Plum Brook Station (PBS) in Sandusky, Ohio, a test-verified model of MVF is needed in order to be able to perform accurate pretest analysis used for determining response limits and abort levels. MVF was designed to vibration test MPCV Orion and was used to perform the system level vibration test of the European Service Module Structural Test Article (E-STA) in 2016. MVF is comprised of an 18 ft diameter annulus table that is driven with sixteen hydraulic vertical actuator assemblies and four hydraulic horizontal actuator assemblies, which allow it to perform single axis vibration testing in the vertical axis and in each of the two orthogonal horizontal axes without the need for reconfiguring the test article.

A head expander for the MVF Table has been designed and built that fills in the center opening providing a continuous flat mounting surface with a maximum diameter of 16.25 feet that expands the vibration testing capabilities of MVF. The MVF Table with this head expander will be used during the SNC Dream Chaser spacecraft vibration test campaign. Therefore, a critical element in a test-verified model of the MVF will be a test correlated finite element model (FEM) of the head expander. To obtain this, engineers from the Structural Dynamics Lab (SDL) at NASA Glenn Research Center (GRC) in Cleveland, Ohio performed a modal pretest analysis, conducted a modal test in July 2019, and most recently correlated the head expander finite element model to the modal test data up to 300 Hz. From the initial test preparations to the final delivery of a correlated finite element model, all efforts mentioned were led by the same early-career engineers at NASA GRC.

From the viewpoint of an early-career engineer, lessons learned about modal pretest analysis, modal testing, and finite element model correlation of the MVF Table expander head will be presented and discussed. This will include the importance of understanding the limitations of using uncorrelated finite element models in the modal pretest analysis and planning, the importance of orthogonality metrics in judging adequacy and accuracy of test mode shapes, and the importance of having the FEM match the as built hardware in the model correlation effort.

**Keywords:** Dream Chaser, Early-Career, European Service Module Structural Test Article (E-STA), Finite Element Model (FEM), Glenn Research Center (GRC), Modal, Modal Pretest Analysis, Modal Test, Mechanical Vibration Facility (MVF), Model Correlation, NASA, Orthogonality, Sierra Nevada Corporation (SNC), Plum Brook Station (PBS), Vibration Test

## INTRODUCTION

The Mechanical Vibration Facility is a part of the Space Environments Complex (SEC) at NASA Plum Brook Station located in Sandusky, Ohio. In addition to the Mechanical Vibration Facility, the SEC is also comprised of the Space Simulation Vacuum Chamber, the Reverberant Acoustic Test Facility (RATF), and the Electromagnetic Interference/Compatibility Test Facility. The test facilities housed at the Space Environments Complex are the world's largest and most powerful space environment simulation facilities [1].

The Mechanical Vibration Facility was designed to perform single axis sine vibration testing in the vertical axis and in each of the two orthogonal horizontal axes without the need for reconfiguring the test article for MPCV Orion class payloads with a test article of 75,000 pounds and a center of gravity elevation of 23 feet above the top surface of the MVF table. MVF was designed for vertical sine sweeps of up to 1.25 g peak from 5 to 150 Hz and horizontal sine sweeps of up to 1.0 g peak from 5 to 150 Hz [1]. The MVF is comprised of a reaction mass, four horizontal servohydraulic actuators mounted on steel pedestals, sixteen vertical servohydraulic actuators utilizing double-spherical couplings, the MVF aluminum vibration table, hydraulic supply system, table control system (TCON), vibration control system (VCON), and facility control system (FCS). The 4,650,000 lbs reaction mass includes an embedded steel plate for modal testing and is used to resist the transfer of vibratory energy from the hydraulic actuators, table, and test article into the shale bedrock foundation. The MVF reaction mass was designed such that it would not require a redesign for testing Altair (Lunar Surface Access Module) class payloads of 100,00.

The aluminum MVF table is essentially an 18-foot diameter annulus optimized for testing MPCV Orion class payloads having an 18-foot diameter attachment circumference. To facilitate the testing smaller payloads and spacecraft, an aluminum head expander was designed and built that fills in the center opening of the MVF table providing a continuous flat mounting surface with a maximum diameter of 16.25 feet. The MVF table with the head expander installed will be used for the SNC Dream Chaser spacecraft vibration test campaign. The head expander model will be used in the effort to create a test-verified finite element model of the whole MVF system prior to the Dream Chaser test campaign in order to perform more accurate pretest analysis used for determining response limits and abort levels. This paper will discuss the MVF head expander modal pretest analysis, modal test, and finite element model correlation effort.

## MODAL PRE-TEST ANALYSIS AND TEST PLAN

A modal pre-test analysis was performed to identify the instrumentation set that would be used during the modal test of the MVF head expander. This included using the finite element model of the head expander to identify the fundamental modes to be identified from the modal test (i.e., target modes), determine instrumentation locations and orientations, develop a Test Display Model (TDM), as well as excitation locations and orientations. The plan was to use the overhead crane and lifting straps to suspend the head expander at the Space Environments Complex. The model used for the pre-test analysis captured both the head expander and the crane suspension system. The crane and lifting straps were modeled with spring elements with arbitrary stiffness values along their longitudinal axes. It should be noted that for the purpose of the modal pre-test analysis, key areas such as the head expander bottom plates and circumferential and diametrical ribs were modeled as continuous plates with shared nodes at the 90° joints. The model developed for the pre-test analysis was able to adequately capture the head expander dynamics, however for the later model correlation effort updates were needed to better match the as-built hardware. The finite element model used for the pre-test analysis is shown in Figure 1.

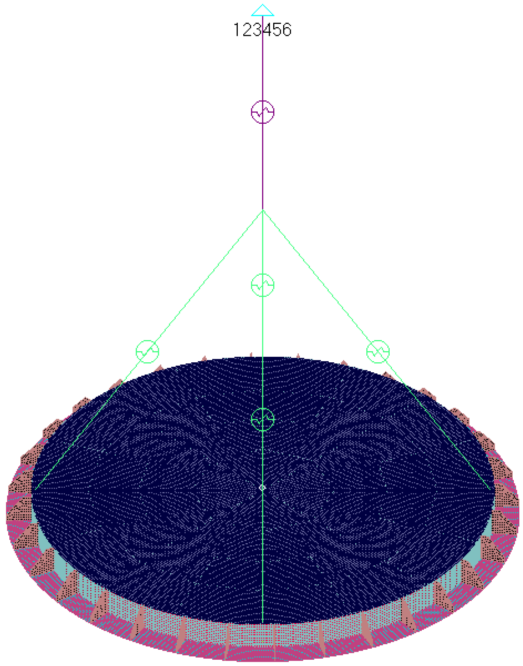



**Fig. 1** Head expander and crane suspension system finite element model

The first seventeen modes of the head expander and crane suspension model are shown in Table 1.

**Table 1** Pre-test finite element model mode frequencies and descriptions

| Mode | Frequency (Hz) | Description | Target |
|---|---|---|---|
| 1 | 0 | Head expander rigid body mode | - |
| 2 | 0 | Head expander rigid body mode | - |
| 3 | 0 | Head expander rigid body mode | - |
| 4 | 0.2 | Suspension bounce mode, Head expander vertical translation | Yes |
| 5 | 0.7 | Suspension pendulum mode, Head expander tilt, Rotation about X | Yes |
| 6 | 2 | Suspension pendulum mode, Head expander tilt, Rotation about Y | Yes |
| 7 | 115.8 | 1st Potato chip | Yes |
| 8 | 118.9 | 2nd Potato chip | Yes |
| 9 | 191.9 | Oil can | Yes |
| 10 | 221.3 | 1st Tri-lobe | Yes |
| 11 | 221.3 | 2nd Tri-lobe | Yes |
| 12 | 317.6 | 1st Quad-lobe | - |
| 13 | 321.2 | 1st Squeeze | - |
| 14 | 321.4 | 2nd Quad-lobe | - |
| 15 | 323.8 | 1st 2nd Bending across top plate | - |
| 16 | 323.8 | 2nd 2nd Bending across top plate | - |
| 17 | 346.1 | 2nd Squeeze | - |

The crane suspension modes (modes 4 – 6) and the head expander elastic body modes up to 300 Hz (modes 7 – 11) were defined as the target modes. The suspension modes were needed for head expander test boundary condition confirmation and the head expander elastic body modes were used for the finite element model correlation. The head expander elastic body target modes were chosen based on the frequency band capabilities of the MVF. An instrumentation list with accelerometer location and direction was then compiled based on capturing the dynamics of the target modes. The list consisted of 50 node locations on the head expander and crane suspension system with a total of 66 degrees of freedom (DOF). The 66 DOF included: 4 biaxial nodes, 6 tri-axial nodes, and 40 uni-axial nodes. Figure 2 shows the head expander and crane suspension system with the modal test degrees of freedom with respect to a global cylindrical coordinate system.

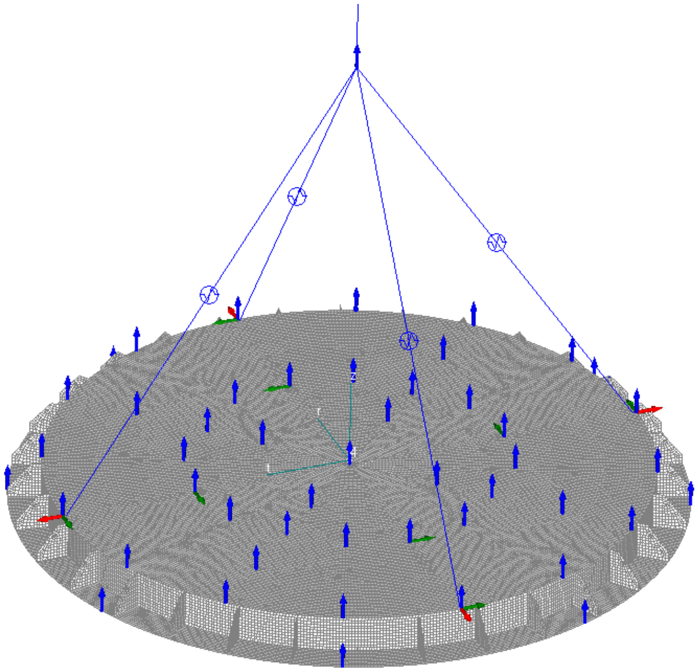


**Fig. 2** Modal test instrumentation degrees of freedom

Computational matrix reduction methods where then used to reduce the mass and stiffness matrices of the full FEM to the test analysis model (TAM) which only included the 66 DOF that would be measured during the modal test. Cross- and self-orthogonality were then computed for the FEM and TAM which were used to determine that the proposed 66 DOF for instrumentation were sufficient for the modal test and target mode parameter estimation [2]. The results from the pre-test orthogonality check are shown in Figure 3 and Figure 4.

| FEM Mode | Freq (Hz) | TAM Mode | Freq (Hz) | %Difference | PORTHO | XORTHO |
|---|---|---|---|---|---|---|
| 1 | 0.00 | 1 | 0.00 | 3.71 | 100.00 | 84.20 |
| 2 | 0.00 | 2 | 0.00 | 53.79 | 100.00 | 85.16 |
| 3 | 0.00 | 3 | 0.00 | 19.46 | 100.00 | 98.88 |
| 4 | 0.19 | 4 | 0.19 | 0.00 | 100.00 | 100.00 |
| 5 | 0.74 | 5 | 0.74 | 0.00 | 100.00 | 100.00 |
| 6 | 2.08 | 6 | 2.08 | 0.00 | 100.00 | 100.00 |
| 7 | 115.82 | 7 | 120.23 | 3.81 | 86.08 | 100.00 |
| 8 | 118.91 | 8 | 122.46 | 2.99 | 88.86 | 100.00 |
| 9 | 191.86 | 9 | 205.31 | 7.01 | 76.22 | 99.96 |
| 10 | 221.32 | 10 | 246.74 | 11.49 | 64.56 | 93.74 |
| 11 | 221.33 | 11 | 246.78 | 11.50 | 64.55 | 93.74 |

**Fig. 3** TAM/FEM frequency and orthogonality comparison

FEM Self Orthogonality Table

FEM Shapes

| | | | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | POgg | 0.0 | 0.0 | 0.0 | 0.2 | 0.7 | 2.1 | 115.8 | 118.9 | 191.9 | 221.3 | 221.3 | 317.6 | 321.2 | 321.4 | 323.8 | 323.8 | 346.1 |
| FEM Shapes | 1 | 0.0 | 1.00 | | | | | | | | | -0.02 | 0.01 | | | | | -0.03 | |
| | 2 | 0.0 | | 1.00 | | | | | | | | 0.01 | 0.02 | | | | -0.03 | | |
| | 3 | 0.0 | | | 1.00 | | | | | | | | | -0.13 | | | | | |
| | 4 | 0.2 | | | | 1.00 | | | | | 0.03 | | | | | -0.07 | | | |
| | 5 | 0.7 | | | | | 1.00 | | | | | -0.04 | | | | | -0.02 | 0.02 | |
| | 6 | 2.1 | | | | | | 1.00 | | | | | 0.04 | | | | 0.02 | 0.02 | |
| | 7 | 115.8 | | | | | | | 1.00 | | | | | | 0.09 | | | | |
| | 8 | 118.9 | | | | | | | | 1.00 | | | | | | | | | 0.22 |
| | 9 | 191.9 | | | | 0.03 | | | | | 1.00 | | | | | 0.02 | | | |
| | 10 | 221.3 | -0.02 | 0.01 | | | -0.04 | | | | | 1.00 | | | | | 0.02 | -0.02 | |
| | 11 | 221.3 | 0.01 | 0.02 | | | | 0.04 | | | | | 1.00 | | | | 0.02 | 0.02 | |
| | 12 | 317.6 | | | -0.13 | | | | | | | | | 1.00 | | | | | |
| | 13 | 321.2 | | | | | | | 0.09 | | | | | | 1.00 | | | | |
| | 14 | 321.4 | | | | -0.07 | | | | | 0.02 | | | | | 1.00 | | | |
| | 15 | 323.8 | | -0.03 | | | -0.02 | 0.02 | | | | 0.02 | 0.02 | | | | 1.00 | | |
| | 16 | 323.8 | -0.03 | | | | 0.02 | 0.02 | | | | -0.02 | 0.02 | | | | | 1.00 | |
| | 17 | 346.1 | | | | | | | | 0.22 | | | | | | | | | 1.00 |

**Fig. 4** FEM self-orthogonality matrix

Modal test impact locations were chosen based on mode anti-node locations for the target elastic body modes. Two vertical impact locations and two radial impact locations were selected. Two large modal hammers and two modal mallets were selected to excite the dynamics of the head expander. Single impact tests were conducted at the four impact locations as well as a multiple impact test utilizing multiple hammers simultaneously at the four locations. All hammers and mallets that were used have calibrated load cells at the tip to measure the force input into the structure. These force inputs were used when calculating the frequency response functions from the time history data prior to modal parameter estimation.

## MODAL TEST EXECUTION

The modal test of the MVF head expander was performed over a two-day period. This included test setup and instrumentation installation, instrumentation check-outs, data collection, preliminary data quality checks, preliminary modal extractions verifying suspension modes and frequencies, tear down, and clean-up of the facility. The modal test data collection portion was comprised of ten impact tests as well as ambient environment data collected on each of the two days. The impact tests conducted included eight single impact tests, one single impact test on the disconnected crane, and one multi-impact test. Four of the eight single impact tests on the head expander were done with a soft hammer tip with the intention of exciting the low frequency modes and gather information on the head expander rigid body modes and crane suspension modes. The other four head expander single impact tests as well as the multi-impact test used a much harder hammer tip to excite the target modes.

Move response shapes for the head expander and crane suspension system were computed using the single impact soft hammer tip tests. The suspension modes were found to be below 5 Hz. This finding confirmed that there was satisfactory frequency separation between these modes and the first elastic body mode of the head expander, which verified the suspension system provided a “free-free” boundary condition.

## MODAL TEST RESULTS

Modal parameters were estimated for the elastic body target modes of the head expander following the on-site test period. Initially, the data from the two vertical single impact tests was used for modal parameter estimation of the elastic body target modes; however, the test self-orthogonality matrix from that data revealed that the modal parameters estimated for the 1$^{st}$ tri-lobe mode and the modal parameters estimated for the 2$^{nd}$ tri-lobe mode were the same. This meant that the two vertical impact locations intended to excite the tri-lobe modes only excited one of the tri-lobes. The test self-orthogonality matrix is shown in Figure 5.

**Test Self Orthogonality Table**

| | | | Test Shapes | | | | |
|---|---|---|---|---|---|---|---|
| | | | 1 | 2 | 3 | 4 | 5 |
| | | Ott | 119.7 | 123.9 | 202.7 | 232.7 | 232.8 |
| Test Shapes | 1 | 119.7 | 1.00 | 0.09 | | | |
| | 2 | 123.9 | 0.09 | 1.00 | | | |
| | 3 | 202.7 | | | 1.00 | | |
| | 4 | 232.7 | | | | 1.00 | 0.78 |
| | 5 | 232.8 | | | | 0.78 | 1.00 |

**Fig. 5** Test self-orthogonality results – single impact test data

Next, the multi-impact test data was used for modal parameter estimation to determine whether that test run was able to excite all the target elastic body modes, most notably both tri-lobe modes. The test self-orthogonality matrix from the multi-impact test data is shown in Figure 6.

**Test Self Orthogonality Table**

| | | | Test Shapes | | | | |
|---|---|---|---|---|---|---|---|
| | | | 1 | 2 | 3 | 4 | 5 |
| | | Ott | 119.9 | 123.9 | 202.8 | 232.8 | 233.5 |
| Test Shapes | 1 | 119.9 | 1.00 | | | | |
| | 2 | 123.9 | | 1.00 | | | |
| | 3 | 202.8 | | | 1.00 | | |
| | 4 | 232.8 | | | | 1.00 | |
| | 5 | 233.5 | | | | | 1.00 |

**Fig. 6** Test self-orthogonality results – multi-impact test data

The cross-orthogonality matrix for this new set of test modes is shown in Figure 7 and the full tabular comparison of the test and FEM is shown in Figure 8.

**FEM/Test Cross Orthogonality Table**

| | | | FEM shapes | | | | |
|---|---|---|---|---|---|---|---|
| | | | 1 | 2 | 3 | 4 | 5 |
| | | Otg | 115.8 | 118.9 | 191.9 | 221.3 | 221.3 |
| Test Shapes | 1 | 119.9 | 0.99 | | | | |
| | 2 | 123.9 | | 1.00 | | | |
| | 3 | 202.8 | | | 0.87 | | |
| | 4 | 232.8 | | | | 0.60 | 0.79 |
| | 5 | 233.5 | | | | 0.82 | 0.57 |

**Fig. 7** Test/FEM cross-orthogonality results – multi-impact test data, uncorrelated FEM

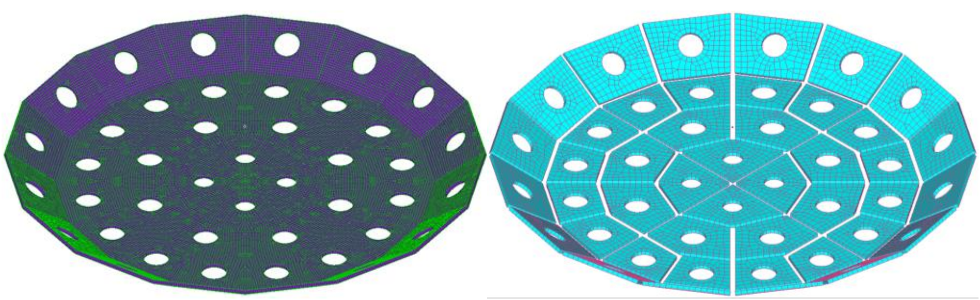
| Test Mode No. | FEM Mode No. | Test Freq Hz | FEM Freq Hz | %Freq Diff | Cross Ortho | CRSS XOrtho 3% | CRSS XOrtho all |
|---|---|---|---|---|---|---|---|
| 1 | 1 | 119.93 | 115.82 | -3.42 | 99.50 | 99.71 | 99.72 |
| 2 | 2 | 123.94 | 118.91 | -4.06 | 99.86 | 99.95 | 99.96 |
| 3 | 3 | 202.75 | 191.86 | -5.37 | 87.34 | 87.34 | 87.35 |
| 4 | 5 | 232.82 | 221.33 | -4.93 | 79.40 | 99.57 | 99.63 |
| 5 | 4 | 233.49 | 221.32 | -5.22 | 81.85 | 99.77 | 99.81 |

**Fig. 8** Test/FEM comparison – multi-impact test data, uncorrelated FEM

Both the Test/FEM mode frequency comparison (Figure 8) and the Test/FEM cross-orthogonality matrix (Figure 7) indicated that changes needed to be made to the head expander FEM during the model correlation in order to match the test data [3].

## FINITE ELEMENT MODEL CORRELATION

In order to finalize the MVF head expander work with a correlated FEM, a model needed to be created that matched the as-built hardware. The FEM that was used for the modal pre-test analysis and planning modeled key areas such as the bottom plates and circumferential and diametrical ribs as continuous plates with shared nodes at the 90° joints. For the model correlation effort to be successful, an updated FEM needed to correctly model the individual aluminum plates of the head expander and how they are joined together. The individual plates in the updated model were joined together using rigid elements which better represented the stiffness of the complete joint penetration welds capped with fillets used for the actual hardware. A key area of comparison between the two finite element models is shown in Figure 9 to highlight an example of the modeling differences mentioned.



(a) (b)

**Fig. 9** (a) Head expander bottom plate from the pre-test analysis FEM (b) Head expander bottom plates from the updated FEM

The bottom plates are just one example of the updates made to the finite element model. Additional updates to the circumferential and diametrical ribs were made to match the as-built hardware. Another key factor of a model correlation is that the weight of the correlated FEM matches the measured weight of the hardware. During the modal test a load cell provided by NASA Plum Brook Station was used to measure the weight of the head expander in the suspended test configuration. Note that this measurement includes the weight of the lifting hardware that was positioned beneath the load cell. The load cell measured a weight of 23,560 lbs. All lifting hardware was weighed using a pallet jack also provided by PBS.

Mass elements were added to the finite element model at the appropriate locations to account for the lifting hardware included in the head expander weight measurement. The lifting hardware that contributed to the measurement of the head expander totaled 462.8 additional lbs. Thus, it can be concluded that the head expander itself weighs approximately 23,097 lbs. Table 2 summarizes the measured weight of the head expander, the weight of the head expander 3D assembly model provided by

NASA PBS, and the weight of updated FEM. Note that the updated FEM does include mass elements for the lifting hardware, however the recorded weight in Table 2 is just of the plate elements used to model the head expander.

**Table 2** Head expander measured weight and model weights

| Head expander | Weight (lbs) |
|---|---|
| Actual hardware | 23,097 |
| CAD | 24,928 |
| FEM | 24,747 |

It should be noted here that even the head expander 3D assembly model (CAD) is significantly heavier than the actual hardware. It has been noted by PBS engineers that the actual thickness of the head expander top plate does not match the drawings or 3D assembly model. Manufacturing of the head expander caused a variation in top plate thickness that ranges from 2.5” to 3.0”. To account for this variation, finite element models with a top plate thickness of 2.75” and 2.5” were developed. The corresponding model weights are shown in Table 3.

**Table 3** FEM weights based on top plate thickness

| Top plate thickness (in) | Weight (lbs) |
|---|---|
| 3.00 | 24,747 |
| 2.75 | 24,008 |
| 2.50 | 23,269 |

Based on the results shown in Table 5, it was decided that the top plate of the FEM that would be used for Test/FEM mode frequency and orthogonality comparisons would have a thickness of 2.5”. It should be noted that the variations of the actual hardware due to manufacturing will add complexity to any further modeling tasks with this model. Correctly representing the mass and stiffness in a model is difficult to do without accurate drawings of the hardware.

With the completion of a finite element model that more closely matched the as-built hardware, analytical comparisons were done between the FEM mode shapes and the test mode shapes. The frequency comparison of the target elastic body modes is shown in Figure 10 and the Test/FEM cross-orthogonality matrix is shown in Figure 11.

| Test Mode No. | FEM Mode No. | Test Freq Hz | FEM Freq Hz | %Freq Diff |
|---|---|---|---|---|
| 1 | 1 | 119.93 | 117.89 | -1.70 |
| 2 | 2 | 123.94 | 123.20 | -0.59 |
| 3 | 3 | 202.75 | 200.56 | -1.08 |
| 4 | 4 | 232.82 | 226.29 | -2.81 |
| 5 | 5 | 233.49 | 227.08 | -2.75 |

**Fig. 10** Test/FEM mode frequency comparison

**FEM/Test Cross Orthogonality Table**

| | | FEM shapes 1 | 2 | 3 | 4 | 5 | Test |
|---|---|---|---|---|---|---|---|
| | Otg | 117.9 | 123.2 | 200.6 | 226.3 | 227.1 | CRSS |
| Test Shapes 1 | 119.9 | 1.00 | | | | | 1.00 |
| 2 | 123.9 | | 1.00 | | | | 1.00 |
| 3 | 202.8 | | | 1.00 | | | 1.00 |
| 4 | 232.8 | | | | 1.00 | | 1.00 |
| 5 | 233.5 | | | | | 1.00 | 1.00 |
| FEM | CRSS | 1.00 | 1.00 | 1.00 | 1.00 | 1.00 | |

**Fig. 11** Test/FEM cross-orthogonality results

The results shown in Figure 10 and Figure 11 indicate that the head expander FEM is correlated with respect to the target elastic body modes [3].

## CONCLUSION

From the perspective of an early-career engineer many lessons were learned during the modal pre-test analysis, test plan, test execution, and model correlation of the MVF head expander. The modal hammer single impact locations based on the anti-node locations of the elastic body target modes using an uncorrelated finite element model did not adequately excite all target modes. The modal hammer multi-impact test well excited the hardware. A lesson was learned that it is important to understand the limitations of using uncorrelated finite element models in the modal pretest analysis and planning. One should have a healthy skepticism of model predictions and always follow best practices (e.g., instrumenting interfaces, etc.) Another lesson learned is the importance of having the capability and flexibility to change excitation locations and orientation.

Test data was collected over a two-day period at NASA Plum Brook Station in Sandusky, Ohio. Some data analysis was performed during the on-site portion of this task, however the majority of the work with the data such as Test/FEM frequency comparisons and cross- and self-orthogonality checks were conducted post-test at NASA Glenn. It has already been stated that it was an incorrect assumption that the single impact tests conducted were adequate to excite all target elastic body modes. A lesson learned was that preliminary model correlation should also be done concurrently during the modal test to help ensure all target modes have been identified and there are no testing anomalies that jeopardize the model correlation effort. This allows for an informed decision to be made on if/when a test configuration should be broken.

The model correlation effort for the MVF head expander required a completely new finite element model that matched the as-built hardware and all lifting hardware. The individual plates in the updated model were joined together using rigid elements which better represented the stiffness of the complete joint penetration welds capped with fillets used for the actual hardware. In other words, the 90° joints of the head expander model were welded together using rigid elements. Had it been needed spring elements could have also been added to the finite element model at the joint regions in order to tune a model parameter for the correlation effort. A lesson learned was that correlation efforts are more efficient when high strain energy region joints are modeled such that additional compliance to the joint aids in the correlation. Adding compliance as opposed to trying to increase the stiffness has a higher overall effect on the correlation of the model.

Overall, the modal test and model correlation of the MVF head expander was a success. The work was primarily done by the same early-career engineer with guidance along the way from more experienced structural dynamics engineers. Lessons learned from the beginning modal pre-test analysis to the final stages of the correlated FEM had deeper impact since all work could be done continuously without interruption or having to pass the work onto another engineer.

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