Washington University in St. Louis

[Washington University Open Scholarship](https://openscholarship.wustl.edu/)

[Mechanical Engineering and Materials Science](https://openscholarship.wustl.edu/mems500)

Mechanical Engineering & Materials Science

11-11-2018

Modal Analysis and Vibration Test of NASA MSFC Shaker Table

Brian Mincks Washington University in St. Louis

David Peters Washington University in St. Louis

Follow this and additional works at: [https://openscholarship.wustl.edu/mems500](https://openscholarship.wustl.edu/mems500?utm_source=openscholarship.wustl.edu%2Fmems500%2F81&utm_medium=PDF&utm_campaign=PDFCoverPages)

Recommended Citation

Mincks, Brian and Peters, David, "Modal Analysis and Vibration Test of NASA MSFC Shaker Table" (2018). Mechanical Engineering and Materials Science Independent Study. 81. [https://openscholarship.wustl.edu/mems500/81](https://openscholarship.wustl.edu/mems500/81?utm_source=openscholarship.wustl.edu%2Fmems500%2F81&utm_medium=PDF&utm_campaign=PDFCoverPages)

This Final Report is brought to you for free and open access by the Mechanical Engineering & Materials Science at Washington University Open Scholarship. It has been accepted for inclusion in Mechanical Engineering and Materials Science Independent Study by an authorized administrator of Washington University Open Scholarship. For more information, please contact digital@wumail.wustl.edu.

Modal Analysis and Vibration Test of Shaker Table

Brian Mincks

Marshall Space Flight Center

July 13, 2018

Reviewed by NASA Mentor Ron Burwell ET40

Mentor Signature Here

National Aeronautics and Space Administration George C. Marshall Space Flight Center Marshall Space Flight Center, AL 35812

July 13, 2018

To: ET40 Vibrations Team From: Brian Mincks Subject: Modal Analysis and Vibration of UD T4000-H Shaker

The Unholtz-Dickie T4000A horizontal shaker (UD T4000A-H) was subject to both a modal tap test and a subsequent vibration test. The modal test was completed on July 2, 2018 and the vibration test was completed on July 9, 2018. The FRF's from the modal test and the acceleration profiles and duration schedules from the vibration test are shown in the attached tests and procedures.

Please direct any questions or comments to Brian Mincks at (740) 348-6826, brian.mincks@nasa.gov, or b.d.mincks@wustl.edu.

Buan Whiche

Brian Mincks Structural Dynamics Intern NASA MSFC ET40

Unholtz-Dickie T4000A Modal Analysis & Vibration Test

Abstract

A shaker can be used to simulate launch vibrations and check responses of structures forced at different frequencies. When vibrating at certain frequencies during tests, structural modes of the shaker table itself can cause the test to abort by accelerating too much or by pushing too much electrical gain through the system. Furthermore, structural modes can produce misleading data at these modal frequencies and cause the test article to be under-tested or over-tested. A modal roving hammer test of the horizontal shaker table is conducted to characterize these modes of the shaker table. Two cases were tested in an attempt to simulate the boundary condition of the table on the shaker: free-free and free-fixed. The free-free case revealed a stretching mode at 1334.2Hz while free-fixed showed two stretching modes at 576.7Hz and 1372.3Hz. A subsequent vibration test revealed controlling 20in from the shaker attachment point best controls these modes without drastically over-testing or under-testing. **Introduction**

The goal of this experiment is to characterize the structural modes of the UD T4000A horizontal shaker in an effort to better understand how to control it at these resonances. A control accelerometer is attached to the shake table and relays how many g's the test article is feeling to the control system. The control system adjusts power to the shaker in an attempt to shake the control accelerometer at a specified level. Due to the continuity in the shaker table structure and the finite location at which a control accelerometer can sense, the control accelerometer can be subject to more or less g's relative to the rest of the structure depending on where it is in the mode shape. If the control accelerometer is in a resonance of the mode shape, it will not take much power to shake the control accelerometer at the specified level and the rest of the mode shape will feel less g's (under testing). Conversely, if the control accelerometer is in a node, the control system will push the shaker harder than necessary. This causes the rest of the mode shape to feel more g's (over testing). Furthermore, the control system may be forced to abort the test in this case because it puts too much gain through the amps in an attempt to push the control accelerometer to the specified levels.

Damage and test abort problems usually occur in the axis of vibration so the in-axis component of the modes is all that is considered in this report. The plate stretching mode responds completely in axis and usually causes the most problems. The stretching modal frequency in Hz of any structure is calculated as in Eq. 1 [1].

$$
f = \frac{V}{AL} \tag{1}
$$

Here, *V* is the speed of sound of the material defined as $V = \int_{0}^{E}$ $\frac{\mu}{\rho}$ where *E* is the elastic modulus of the material and ρ is the density. *L* is the length of the structure in the axis of stretching. *A* is a constant dependent on the boundary condition and is defined in Eq. 2.

$$
A = \begin{cases} 2, & \text{Free} - \text{Free} \\ 4, & \text{Free} - \text{Fixed} \end{cases}
$$
 (2)

A modal roving hammer test is performed to reveal the stretching modes that Eq. 1-2 predict. The table has a free-forced boundary condition on the table itself. A forced boundary condition cannot be simulated in the modal tap test so the plate will be simulated as both free-free and free-fixed to reveal all frequencies around which the stretching mode might appear. A subsequent vibration test is performed to sweep through the modal frequencies discovered in the modal tests. The swept sine test will be repeated, but controlled at various locations along the length of plate. This should reveal how much vibration gain or attenuation is being felt around the plate

Apparatus & Procedures

Part 1: Modal Analysis

The experiment starts with the free-free modal test. The equipment list for the modal tests is seen below in Table 1.

	Item	NEMS/SN	Cal Date	Cal Due
R1	Accelerometer	LW147719	9/20/14	9/20/15
R ₂	Accelerometer	LW147963	2/28/15	2/28/16
	Hammer/Load Cell	LW40109	N/A	N/A
	20g hammer mass	N/A	N/A	N/A
	Plastic hammer tip	N/A	N/A	N/A
	Data Acquisition Front End	45034708	N/A	N/A
	Dell M6400 Computer	TL13A	N/A	N/A
	Free Shaker Table	N/A	N/A	N/A
	Fixed Shaker Table	N/A	N/A	N/A

Table 1 Equipment List

Accompanying, relevant calibration documents are available in Appendix A.1. A schematic showing the equipment setup is seen below in Figure 1.

To simulate the free-free boundary condition, the shaker is setup on foam blocks. Simple, checkout tests revealed the first structural mode was more than ten times that of the first rigid body mode signifying the free-free simulation is valid [1]. Figure 2 shows this setup of the shaker table on foam.

Figure 2 Free-Free test setup

To simulate a free-fixed boundary condition, the table is left on the shaker. Lubricating oil that flows during shaker operation is pumped in between the table and its support structure to create the free boundary condition. The shaker is left locked to fix the other end. Figure 3 shows the free-fixed configuration.

Figure 3 Free-fixed test setup

The tap test utilized two response accelerometers and six tap locations. The plate geometry, accelerometer locations, and hammer tap locations are seen below in Figure 4.

Figure 4 Geometry and tap locations on shaker table (dimensions in inches)

Here P1 – P6 signify the tap locations of the hammer and R1 and R2 are the accelerometer locations. The accelerometers are glued onto the table using Cyanoacrylate. The test excites frequencies over 1000Hz so glue must be used to ensure good energy transfer. Taps on point P1-P3 happen along the thickness of the plate in the +Z direction while taps on points P4-P6 happen in the -Z direction.

All of the data is taken using LMS Impact Testing 13A software. Table 2 shows all of the test setup parameters. The impact scope parameters are all codependent. Once two are defined, the other two are calculated. Bandwidth is set to 1600Hz because tests usually abort around what is suspected to be the stretching mode at 700Hz. In order to prevent leakage in the data, 1600Hz is chosen to ensure at least twice the frequency of interest is measured [1]. Acquisition time was set next at 5.12s. The plate rang for approximately three seconds in the free-free case when struck with the hammer. In order to observe the entire impact with a margin of safety, 5.12s is chosen. Once the Impact scope parameters are chosen, the tip and hammer masses has to be addressed. A 10 – 20 dB drop in impact energy is desired across the bandwidth of interests. This prevents the introduction of leakage through non negligible energy being input at a frequency that's not being measured [1]. This can be accomplished with many different hammer mass and tip configurations. For this experiment, a hard plastic tip and two 20g masses were used. All the trigger settings were chosen from what the software suggests. A few test taps will yield suggested values similar to those seen below in Table 2. The data is slightly windowed to ensure the entire impact is observed without having lengthy acquisition time. If excluded from the table below, use the default settings.

Part 2: Vibration Test

Once the modes have been found and characterized, a vibration test is conducted to sweep across the modal frequencies. The test uses 4 accelerometers at varying lengths from the shaker attachment point. The same test will be run 4 times with the exception of changing the control accelerometer. This should reveal what the rest of the structure is feeling compared to what the control accelerometer feels. The test equipment list is seen below in Table3.

Accompanying, relevant calibration documents are available in Appendix A.2. A schematic of the test setup is seen in Figure 5.

Figure 5 Vibration test setup schematic

Figures 6 shows a detailed drawing of the accelerometer locations on the table.

Figure 6 Accelerometer locations for vibration test (dimensions in inches)

Figure 7 shows a picture of the actual test setup.

Figure 7 Vibration test setup

The test is controlled and recorded with the VibrationVIEW software. The test sweeps from 40Hz – 2000Hz at 0.5g. The test sweeps through the frequencies at 5 octaves/min with an abort range of $\pm 50dB$. All of the other settings are default. The first test controls with an accelerometer at P1, the second test controls with an accelerometer at P2, etc. All the other accelerometers in these tests simply record the response.

Results

Part 1: Modal Analysis

The frequency response functions (FRF's) at each tap location of the free-free tap test are seen below in Figure 8. The free-fixed FRF's are seen in Figure 9. Both Figures 8 and 9 are obtained directly from LMS.

Figure 8 FRF's of free-free modal tap test

Figure 9 FRF's of free-fixed modal tap test **1.** But C 2.1 M is onlined integrated though typics.

The quality of the data appears to be good. There is some noise in the data but it's all relatively small compared to the modal peaks. The
sm

All the peaks in all the test configurations correspond to a stretching mode. LMS directly animates the mode shapes that it is recording. The free-free stretching mode at 1334.2 Hz is seen below in Figure 10. The first and second modes of the free-fixed case at 576.7 Hz and 1360.6 Hz are seen in Figures 11 and 12, respectively.

Figure 10 Free-free stretching mode at 1334.2 Hz, 0.02% damping

Figure 11 Free-fixed stretching mode at 576.7 Hz, 0.67% damping

Figure 12 Free-fixed stretching mode at 1360.6 Hz, 1.19% damping

Part 2: Vibration Test

Figure 13 shows the vibration profile controlling at P1. Figure 14 shows the vibration profile controlling at P2. Figure 15 shows the acceleration profile controlling at P3. Figure 16 shows the vibration profile controlling at P4. All of the acceleration profiles are exported directly from VibrationView

Figure 13 Vibration profile controlling at P1

Figure 14 Vibration profile controlling at P2

Figure 15 Vibration profile controlling at P3

Figure 16 Vibration profile controlling at P4

The data looks good in quality. There doesn't appear to be any outliers that signify something went wrong in the tests.

Discussion

Part 1: Modal Analysis

The FRF peaks denote the modal stretching frequencies and align well with the predicted analytical first stretching mode frequencies as seen in Table 3.

Table 4 Experimental-analytical comparison of modal frequencies

The discrepancies in the data can be accredited to the irregular plate geometry. Equations 1 and 2 model a fixed-length uniform plate. The plate is not of fixed length and is riddled with mounting and bearing holes which both directly contradict the assumptions that are necessary to use Equations 1 and 2 accurately. The exaggerated error in the free-fixed case is due to an imperfect fixed boundary condition. While the plate is locked in the shaker, the whole shaker/plate system can still move because the shaker sits on airbags.

Part 2: Vibration Test

Point 1 appears to be a node. Figure 13 shows the acceleration profile controlling at point 1. While it stays on its 0.5g line through the frequency, all the other points have vibration gains over 20 at the first stretching mode. The control system has to push the shaker hard to get this node up to a 0.5g while the others are resonating. Point 2 appears to be approaching a resonance. Figure 14 shows the control system doesn't have to push the shaker too hard to get Point 2 up to the specified acceleration. It's also seen that the Point 1 node is barely moving and Points 3 and 4 are resonating a little harder at the first stretching mode. Figures 15 and 16 show similar trends with points 3 and 4. It appears that point 4 resonates the hardest followed by point 3 and then point 2. This was analyzed by exporting all the vibration view data to Excel.

At each control location, the acceleration gains were analyzed by comparing the responses to the control. Table 5 shows these trends.

Control Point	Vibration Gain					
	P1	P2	P3	P4		
P1		34.59554	45.88156	60.12159		
P ₂	0.009636		1.573561	1.753507		
P3	0.010218	0.628204		1.10232		
P4	0.010724	0.559686	0.878997			

Table 5 Vibration gains across table at varying control points

This makes sense from the results from the modal test. The animations in Figures 11 and 12 show both ends moving like an accordion. However, that is for a free fixed boundary condition. Since the 'fixed' end is actually forced in the actual vibration test, a Galilean transformation must be applied to the frame of reference so that the free-fixed mode shapes can be applied. In this transformed reference frame, the mode shape is viewed from the perspective of the shaker head and the end (P4) will be resonating the most while the attachment point (P1) will be fixed.

Conclusion

Stretching modes exist in the shaker table. The free-free configuration has a stretching mode at 1334.2 Hz and the free-fixed configuration has two modes at 576.7 Hz and 1360.6 Hz. The freefree test aligns quite well with the analytical solution while the free-fixed case test setup could be improved to make a more fixed end. These results can be used to help control the shaker through the stretching modal frequencies. A Galilean transformation of the reference frame is conducted to the accordion-like mode shape from the free-fixed test so it is viewed from the perspective of the shaker. In this reference frame the accordion mode will be stationary at the attachment point to the shaker and be resonating at the full length. Table 5 concisely depicts how this length dependent resonance affects the vibration gain in the structure. To best control the shaker table and to provide an accurate, safe, abort free test through the stretching modes, control around 20 inches from the attachment head. The vibration gains felt throughout the rest of the structure only get as big as 1.75. This should prevent the test from aborting and ensure the test article is not being damagingly over-tested.

References

1. Rost, Robert, Allemong, Randal. (2018, May 15-17). *Practical Data Acquisition and Experimental Modal Analysis Theory and Applications*.

Appendix A.1 Modal Test accelerometer calibration information

Ch2/Free Side

 $cos\int shaket$ 5^{18}

Appendix A.2 Vibration Test accelerometer calibration information

