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Structural Dynamic Analysis of a Spacecraft Multi-DOF Shaker Table

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Abstract

Finite element enforced response analysis was performed on a three axis expander head shaker table to aid in the design of the table structure and vibration control system. The payload for this shaker system is a generic spacecraft with a multitude of flexible modes across a broad frequency band. A Craig-Bampton representation of the spacecraft was used to expedite analysis of multiple shaker table designs. The analysis examines the required forces in the actuators for a constant amplitude base acceleration sine sweep test, the resulting forces in the spacecraft and table attachment restraints, and the resulting accelerations on the spacecraft structure. The results show the spacecraft response is very high at the spacecraft center of gravity (CG) due to the high CG offset and cantilever effect of the low frequency spacecraft bending modes. The high response can be addressed by "notching" the input vibration levels to avoid over-testing the spacecraft. At frequencies above 25 Hz, the spacecraft modal effective masses are very small, and the response of the shaker table dominates the response. Anti-resonances of the shaker table in the frequency range of interest reduce the acceleration output and require much higher actuator forces to achieve the acceleration specification. These effects may require stiffening the shaker structure to move the modes out of the test frequency range or increasing the shaker table damping.

Introduction

NASA Glenn Research Center is building a vibration and acoustic test facility at the Plum Brook Station facility to provide testing for space vehicles. The mechanical vibration facility (MVF) will be used for single axis base shake vibration testing. ATA Engineering, Inc. is providing ongoing structural dynamics finite element analysis support for the MVF shaker table and foundation design. This paper describes the dynamic analysis of an early shaker table design. The shaker table, designed by TEAM Corporation, is required to produce base shake sine sweep accelerations of 1.25 g vertically and 1 g laterally up to 150 Hz for a spacecraft weighing as much as 75,000 lb (31,800 kg) and a CG offset as much as 23.67 ft (7.21 m) from the base. During the lateral shake, the vertical actuators are locked out and the spherical couplings act as vertical restraints to react the overturning moment. Results are shown to 180 Hz to allow for better visualization of modal behavior near 150 Hz.

Units

Values are given in both SI and U.S. Customary Units. The measurements and calculations were made in U.S. Customary Units.

Modeling and Analysis

Shaker Table Model

The shaker table NASTRAN finite element model is shown in figure 1. The table consists of sixteen cyclic symmetric sections arranged in a truss pattern modeled with shell (CQUAD4) elements. Mass less rod elements (CROD) is used to model the actuators. The actuators are modeled as an axial restraint and do not provide rotational stiffness. For lateral excitation in the Z-axis, there are sixteen vertical (Y-direction) actuators, two lateral links in the X-direction, and one lateral actuator in the Z-direction. The two links provide lateral restraint in the direction orthogonal to lateral excitation. For the vertical excitation configuration, three links provide lateral restraint in the two orthogonal axes. A 1 in. thick top plate lies on top of the truss structure to attach the vehicle and to provide increased lateral stiffness. The top plate is also meshed with shell elements. The total weight of the all-steel shaker table is 40,185 lb (18,230 kg).

Spacecraft Modeling

Two spacecraft models were used to evaluate the shaker table performance, as shown in figure 2. The first is a Craig-Bampton mass and stiffness matrix representation of a spacecraft to be tested on the shaker table. The second is a lumped mass representation of the spacecraft including rotational inertia properties located at the spacecraft center of gravity (CG). These models provide two extreme dynamic response cases for the shaker table. The Craig-Bampton model demonstrates the affect of spacecraft modes on the shaker table response. The spacecraft effective mass modes are dominant in the low frequencies, so the spacecraft has little mass loading effect on the shaker table at high frequencies. The lumped mass representation has no modal amplification but provides mass loading at all frequencies.

The spacecraft model Craig-Bampton reduction provides computational advantages over the explicit representation. The spacecraft has numerous modes in the frequency range of interest with very little modal effective mass, such as local panel modes. The spacecraft modes and the modal effective mass, listed in percent of total mass, are shown in table 1. Greater than 95% of the spacecraft high effective mass modes occur below 30 Hz for all directions except the torsional direction, which will not be excited by the shaker. The modes with low modal effective mass will have negligible effect on the shaker response and can therefore be eliminated from the analysis easily using single point constraints (SPCs). Multi-point constraint (MPC) equations are written to extract the responses at the spacecraft center of gravity and other nodes of interest. The MPC equation coefficients were developed using standard coupled loads analysis data recovery techniques and include the net load (CG acceleration) of the payload. Plotting elements (PLOTELs) are shown in figure 2 for visualization purposes.

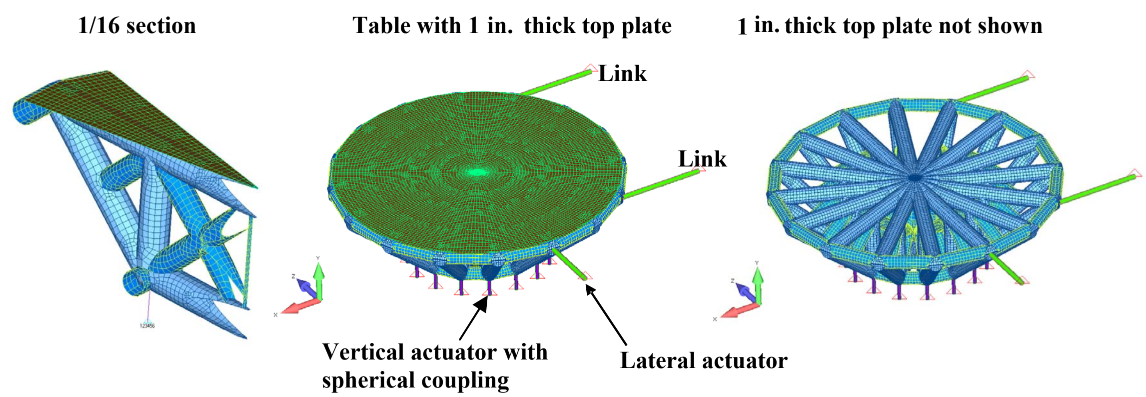

Figure 1.—Shaker table finite element model in the Z-axis lateral excitation configuration.

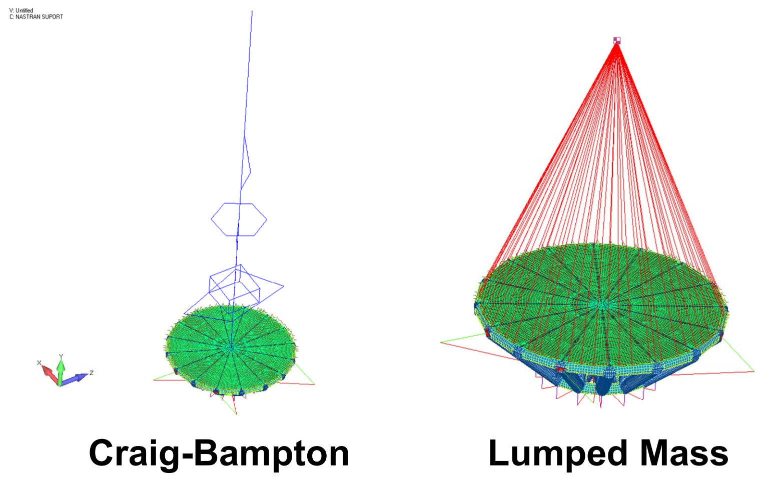

Figure 2.—Shaker table models with Craig-Bampton and lumped mass spacecraft representations.

TABLE 1.—SPACECRAFT MODAL EFFECTIVE MASSESAS PERCENT OF TOTAL MASS

Mode	Frequency, Hz	Tx, %	Ty, %	Tz, %	Rx, %	Ry, %	Rz, %
1	2.36	19.24	0.00	0.33	0.11	0.00	6.30
2	2.37	0.33	0.00	19.32	6.36	0.00	0.11
3	6.76	47.04	0.00	0.00	0.00	0.00	47.14
4	6.98	0.00	0.00	52.43	52.69	0.01	0.00
5	8.73	2.89	0.02	0.00	0.00	0.00	2.55
6	10.86	0.02	0.00	6.12	8.84	0.00	0.02
7	11.55	19.23	0.28	0.00	0.00	0.01	26.19
8	12.14	0.00	0.00	12.34	16.59	0.41	0.00
9	12.96	1.36	65.83	0.00	0.00	0.00	1.76
10	14.36	4.86	15.49	0.00	0.00	0.00	7.36
11	15.46	0.12	0.90	0.68	0.96	0.15	0.19
12	15.48	0.19	0.81	1.31	1.82	0.36	0.30
13	16.83	0.00	0.26	4.64	7.98	1.88	0.00
14	17.15	0.91	3.89	0.12	0.21	0.31	1.81
15	17.48	0.00	0.00	0.00	0.00	0.00	0.00
16	17.72	0.00	0.00	0.00	0.00	0.08	0.00
17	18.28	0.25	3.02	0.00	0.00	0.02	0.49
18	18.59	0.01	0.00	0.03	0.06	57.51	0.02
19	19.60	0.24	0.04	0.01	0.02	0.09	0.44
20	20.02	0.04	0.98	0.00	0.00	0.02	0.06
21	20.84	0.00	0.05	0.02	0.07	0.52	0.00
22	21.79	0.00	0.00	0.00	0.00	0.12	0.00
23	22.13	0.03	1.33	0.00	0.00	0.00	0.07
24	23.45	0.00	0.02	0.00	0.00	0.00	0.00
25	24.28	0.08	0.01	0.00	0.00	0.01	0.10
26	24.71	0.00	0.00	0.00	0.00	0.00	0.00
27	26.24	0.00	0.00	0.00	0.00	0.03	0.00
28	26.60	0.00	0.00	0.00	0.00	0.00	0.00
29	26.90	0.00	0.00	0.00	0.00	0.01	0.00
30	27.17	0.08	0.12	0.02	0.02	0.26	0.12
31	27.44	0.00	0.04	0.00	0.00	0.00	0.00
32	28.10	0.00	0.00	0.00	0.00	6.89	0.00
33	28.31	0.00	0.00	0.00	0.00	0.00	0.00
34	28.66	0.00	0.03	0.00	0.00	0.45	0.00
35	29.32	0.00	0.21	0.00	0.00	0.08	0.00
	Total	**96.93**	**93.36**	**97.39**	**95.75**	**69.23**	**95.06**

(green)	is>1%
(yellow)	is<1% and >0.1%

Analysis Cases and Results

The NASTRAN "large mass method" was used to provide a fixed base constraint to the shaker table for the modal and forced response analyses. The large mass was connected to the foundation-side ends of all the actuators with a rigid element. The actuator rod elements do not provide rotational stiffness, so only the actuators being forced in their axial direction will drive the shaker table. In other words, when a force is applied to the large mass in the vertical directions, the lateral actuators do not resist any motion in the vertical direction. This allows for the enforced response analyses in all three directions to be completed using the same model without reconnecting the mass to the driven actuator ends.

The objective of the enforced motion analysis is to examine the actuator forces when the spacecraft base is excited at the required input levels of 1.25 g vertically and 1 g laterally. This is done by first enforcing an acceleration to the ends of the actuators, and then taking the transfer function between the actuator force and the acceleration at the spacecraft base and multiplying by the required input level. The results for each step are included to provide insight into the final transfer function.

Modal Analysis

Table 2 summarizes the system (shaker table and spacecraft) modal frequency results. The shaker table rocking mode couples with the spacecraft bending modes at approximately 6 Hz. In this frequency range, the system modal frequencies using the Craig-Bampton flexible spacecraft representation are lower compared to lumped mass spacecraft model.

TABLE 2.—SYSTEM MODAL FREQUENCIES FOR THE TWO SPACECRAFT REPRESENTATIONS

	Mode Shape	Lumped Mass S/C, (Hz)	Craig-Bampton S/C, (Hz)
1	Rocking, X	6.8	5.6
2	Rocking, Z	7.1	6.0
3	Vertical, Y	45.3	128.7
4	Lateral, Z	47.6	58.9
5	Lateral, X	59.5	66.9

At frequencies above 30 Hz, the Craig-Bampton flexible spacecraft model has little contributing effective mass, thereby creating less impact on the shaker table modes. Above 30 Hz, the modal frequencies are higher for the system model with Craig-Bampton spacecraft representation compared to the lumped mass spacecraft model. This is especially true in the vertical direction, when the first vertical table mode frequency increases from 45.3 Hz for the lumped mass model to 128.7 Hz for the Craig-Bampton model. Figures 3 and 4 show the lumped mass model and Craig-Bampton model rocking, vertical, and lateral mode shapes.

Enforced Motion Analysis

The enforced motion analysis is shown for one lateral (X-axis) and the vertical direction (Y-axis). The other lateral direction (Z-axis) results are very similar to the X-axis. For all enforced motion cases, a 1% damping ratio was assumed. The results are compared for the lumped mass and Craig-Bampton flexible spacecraft (S/C) models in figures 5 to 13. For the enforced motion analysis, the sequence of results is outlined. In the first graph, the average spacecraft base response due to a constant acceleration at the actuator ends is shown. The average spacecraft base response is computed using a constraint element attached to each of the spacecraft attachment points. In the next graph, the average actuator force required to create this acceleration is shown. In the last graph, the transfer function of the actuator force divided by the average base response is shown. This is the required force to create the 1.25 g vertical or 1 g lateral base drive.

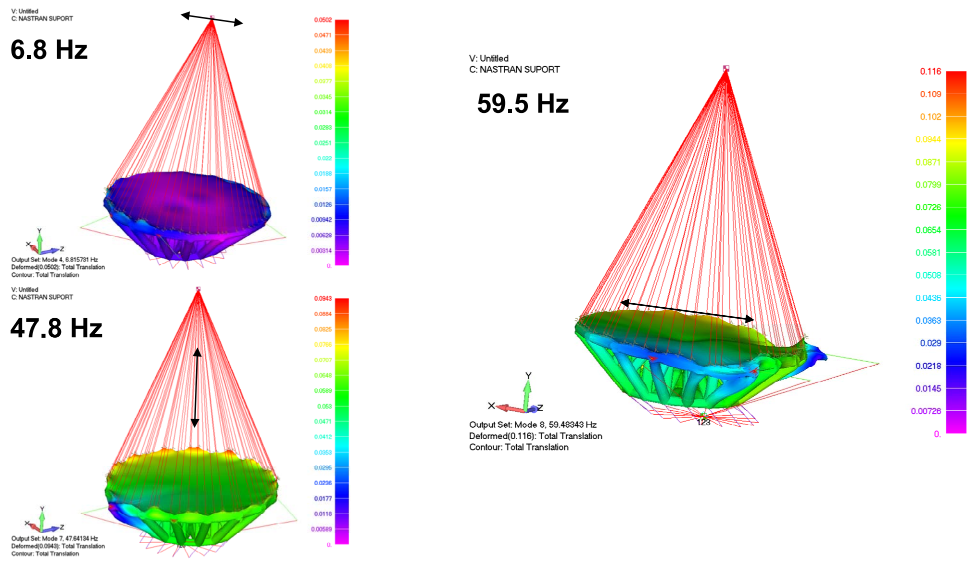

Figure 3.—Mode shapes for shaker table with lumped mass spacecraft representation.

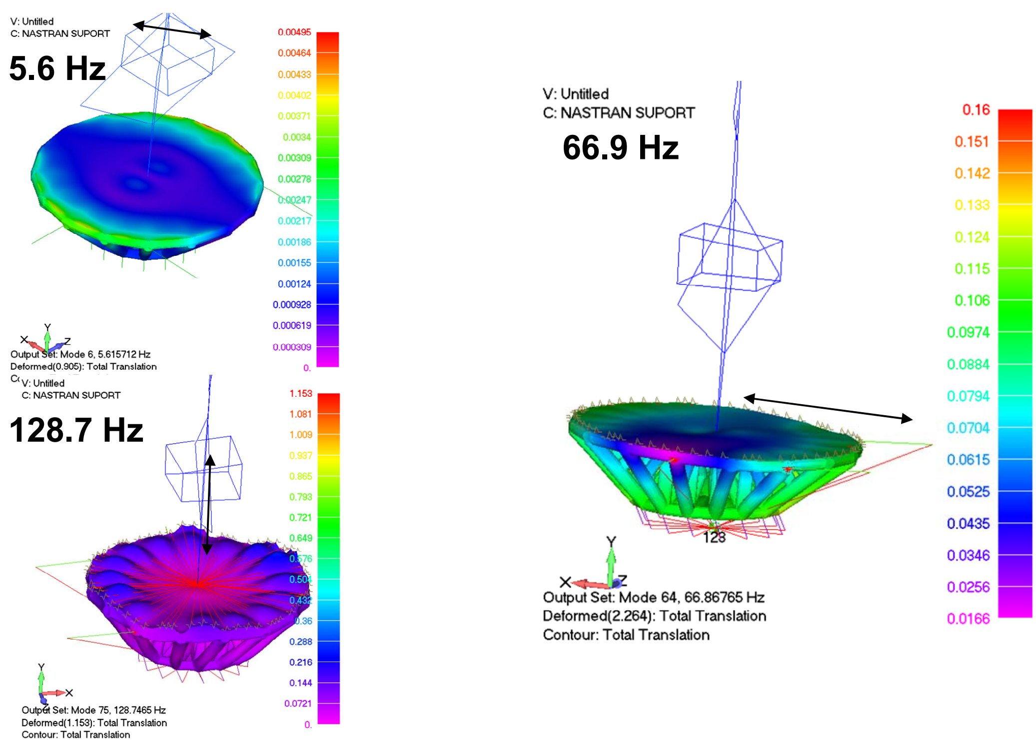

Figure 4.—Mode shapes for shaker table with Craig-Bampton spacecraft representation.

Figure 5 shows the average response at the spacecraft base due to a 1.25 g vertical acceleration at the actuator ends. Figure 6 shows the average vertical actuator load to produce a 1.25 g response at the actuator ends. Figure 7 shows the required actuator force to produce a 1.25 g average response at the spacecraft base. The forces are compared to a limit of 16,000 lb (71,170 N), which is set by the actuator rating. The actuator force requirements are below the actuator limit up to nearly 150 Hz for both the Craig-Bampton model and the lumped mass model except at the flexible spacecraft axial mode.
Figure 1 shows the response of the flexible spacecraft CG due to a 1.25 g base shake. The CG acceleration level for the fixed mass is not shown, as it is the same as the base acceleration level. The CG acceleration level is 20 g at the axial spacecraft mode, so the input levels would need to be reduced to prevent damaging the spacecraft. Thus, the high actuator force levels for the spacecraft axial mode would be reduced as well.

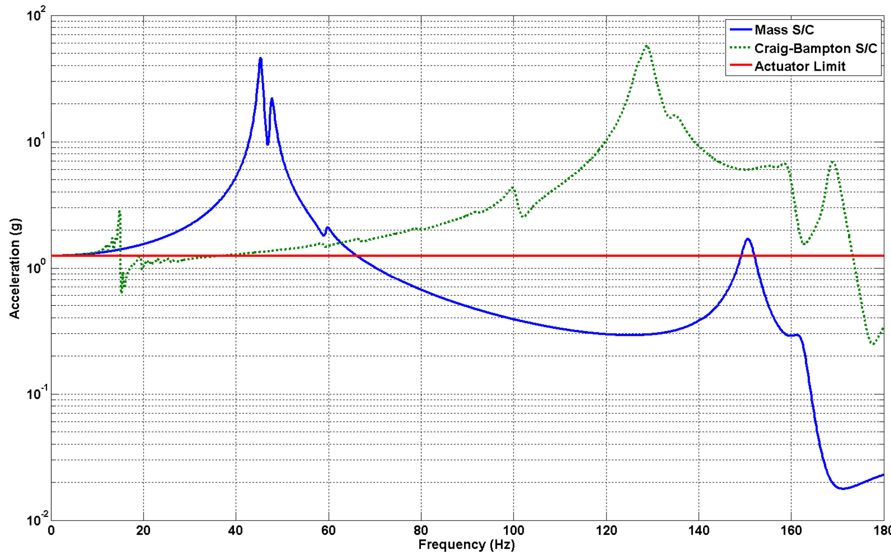

Figure 5.—Average response at spacecraft base due to 1.25 g vertical (Y-axis) drive for lumped mass and Craig-Bampton models.

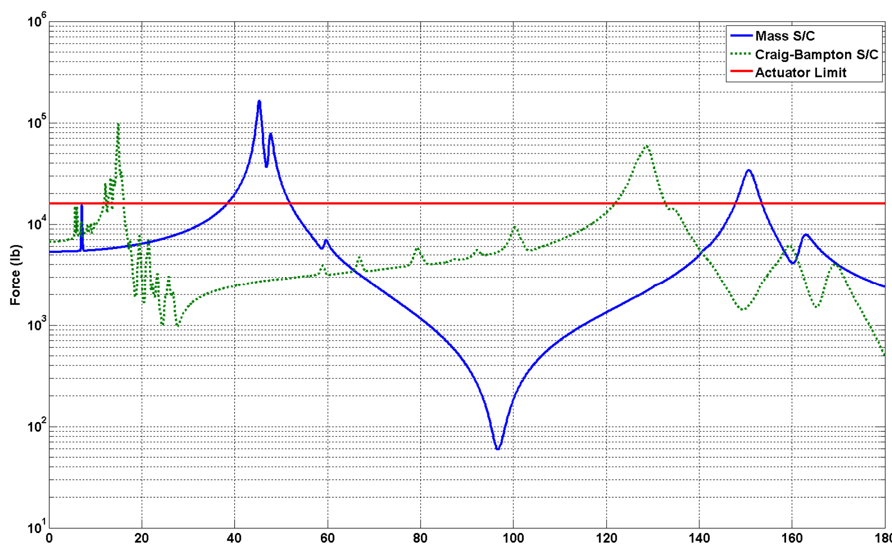

Figure 6.—Required average actuator force to produce 1.25 g vertical (Y-axis) drive at actuator ends for lumped mass and Craig-Bampton models.

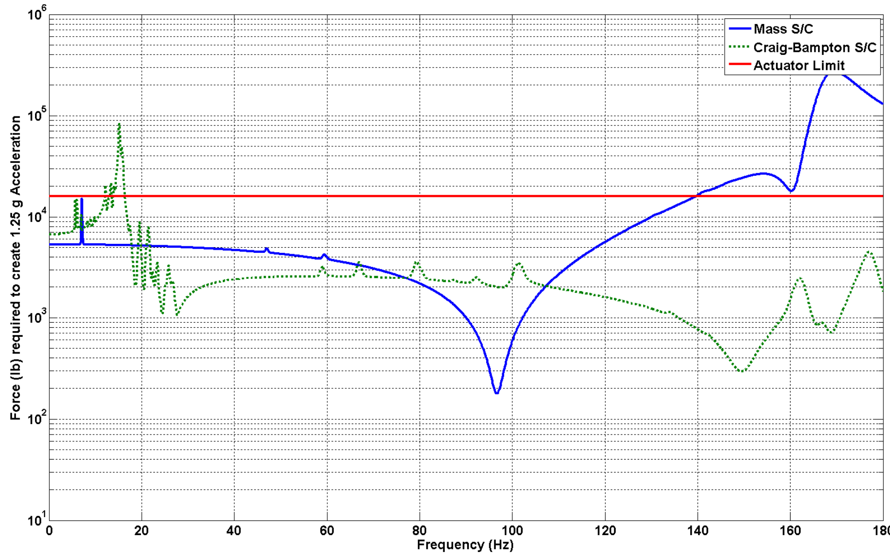

Figure 7.—Required average actuator force to produce 1.25 g vertical (Y-axis) drive at S/C base for lumped mass and Craig-Bampton models.

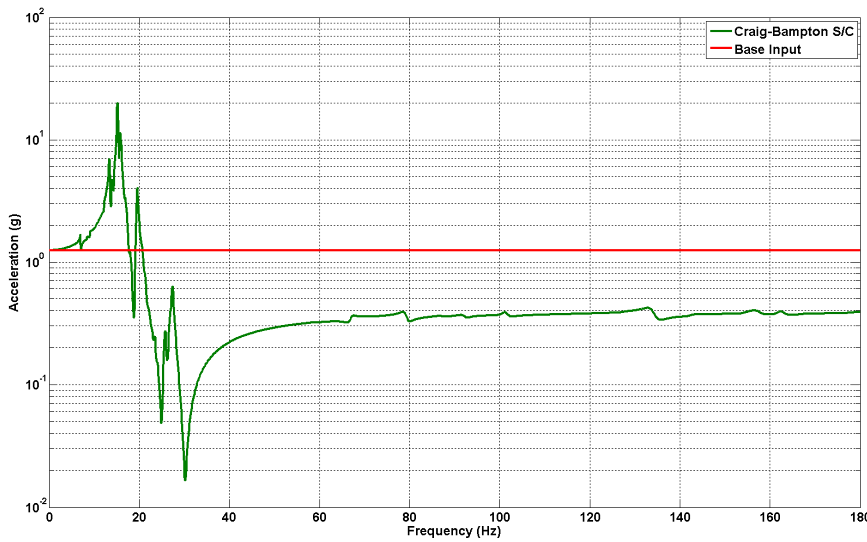

Figure 8.—Response at spacecraft CG due to 1.25 g vertical (Y-axis) base drive for Craig-Bampton model.

Figure 9 shows the average spacecraft base response due to a 1 g lateral acceleration at the actuators for both models. Figure 10 shows the average lateral actuator load to produce 1 g at the actuator ends, compared with an actuator limit of 90,000 lb (400,340 N). Figure 11 shows the lateral actuator force required to produce 1 g average response at the spacecraft base, which is found by dividing the required actuator force by the spacecraft average base acceleration. For the lumped mass model, the actuator forces exceed the force limit at the rocking mode at 6 Hz, and at the anti-resonances of the shaker table at 67 and 121 Hz. For the Craig-Bampton flexible spacecraft, the actuator forces exceed the force limit at the coupled spacecraft/shaker table bending modes and at the shaker table anti-resonances. Figure 12 shows the CG response for the Craig-Bampton model due to a 1 g lateral (X-axis) base shake. As was the case for the vertical shake, the CG response exceeds 1 g at the low order modes, so notching of the input signal would reduce the high actuator force loads at these frequencies. However, the actuators would not be able to provide the needed force input to attain the 1 g lateral base accelerations at the shaker table anti-resonance at 75 Hz.

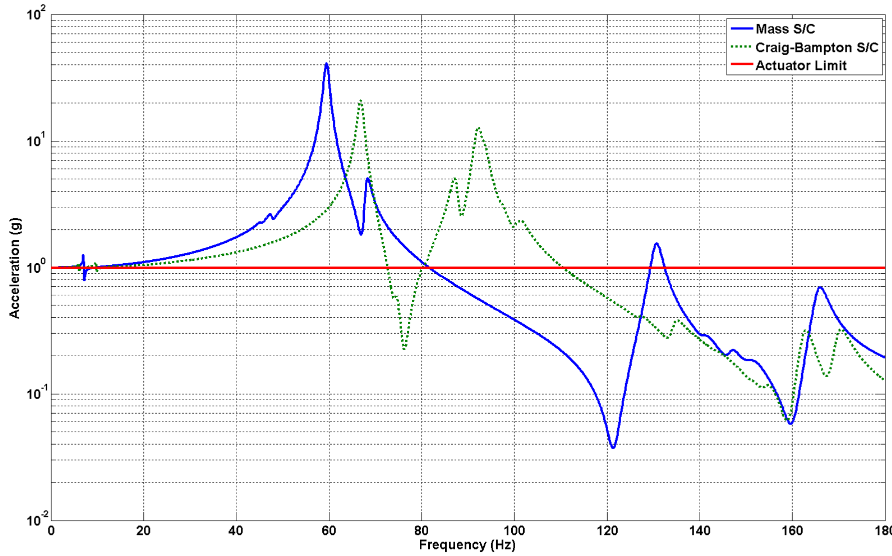

Figure 9.—Average response at spacecraft base due to 1 g lateral (X-axis) drive for lumped mass and Craig-Bampton models.

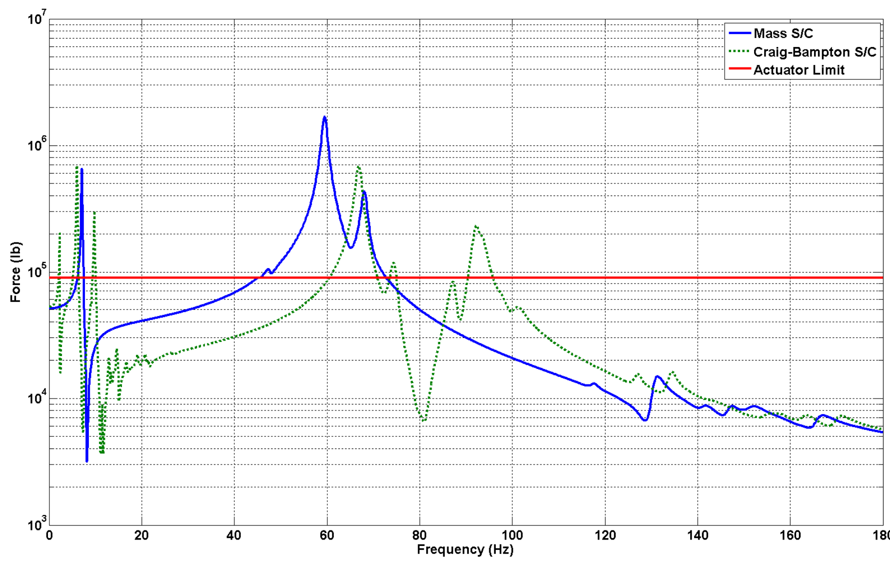

Figure 10.—Required average actuator force to produce 1 g lateral (X-axis) drive at actuator ends for lumped mass and Craig-Bampton models.

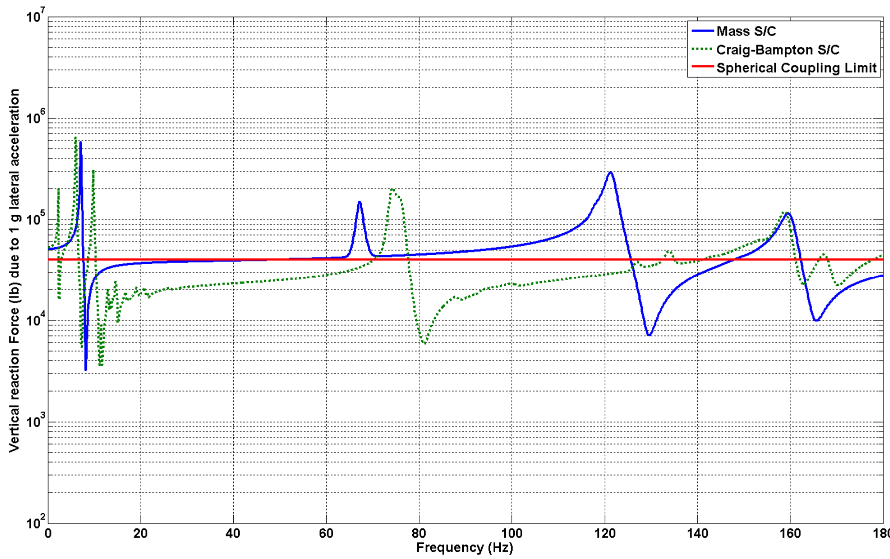

Figure 11.—Required average actuator force to produce 1 g lateral (X-axis) drive at S/C base for lumped mass and Craig-Bampton models.

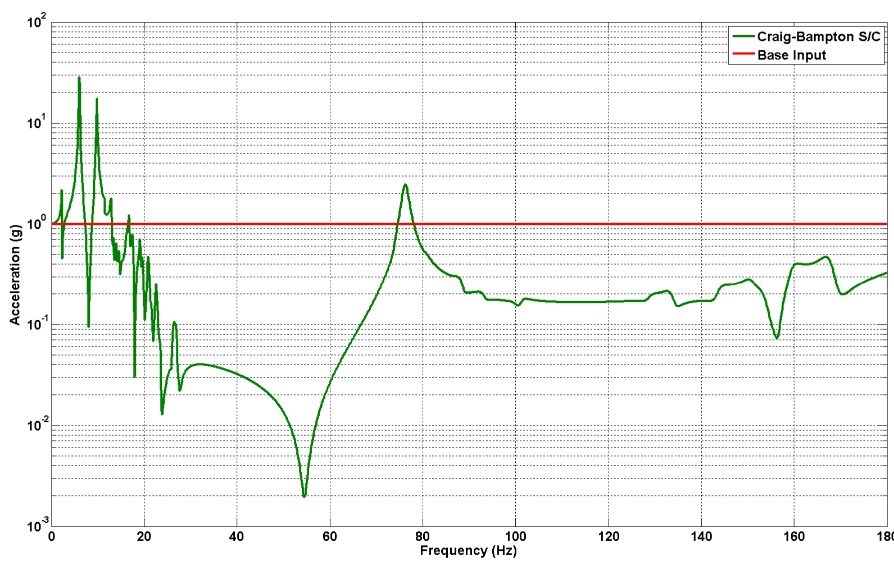

Figure 12.—Response at spacecraft CG due to 1 g lateral (X-axis) drive at S/C base for Craig-Bampton model.

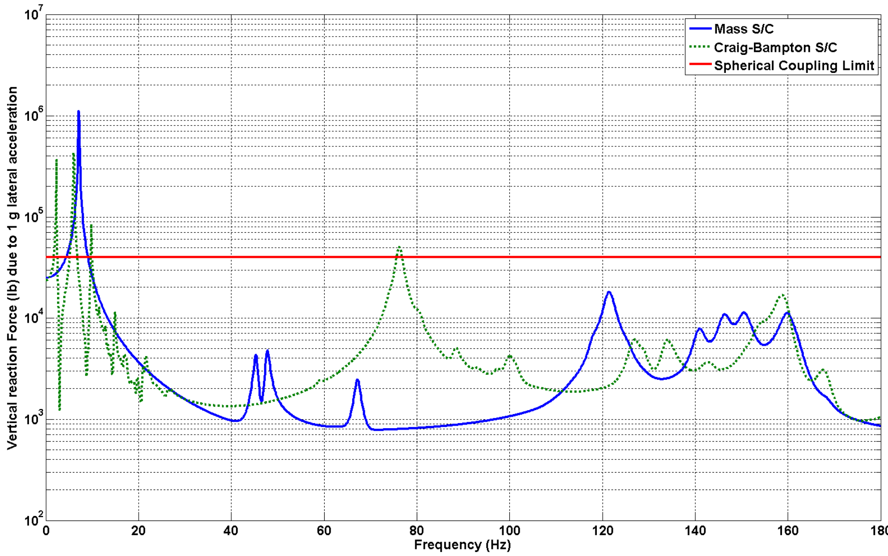

Figure 13.—Required vertical actuator spherical coupling reaction force due to 1 g lateral (X-axis) drive at S/C base for lumped mass and Craig-Bampton models.

For the lateral shake, the vertical actuator spherical couplings must react the overturning moment due to the CG offset. Figure 13 shows the required spherical coupling reaction force for a 1 g base shake, compared with a 40,000 lb (177,930 N) limit. The force limit is set by the maximum allowable spherical coupling load. For the lumped mass model, the forces in the spherical coupling exceed the limit only at the 6 Hz rocking mode. For the Craig-Bampton model, the forces in the spherical coupling exceed the limit for the lateral spacecraft modes below 10 Hz, and at the 75 Hz shaker table anti-resonance.

Conclusions

The structural dynamic analysis of the shaker table demonstrates the complexities in the design. Due to the size of the test article, the vibration level, and the testing frequency range, it is nearly impossible to create a shaker table that will not have modes in the test frequency range up to 150 Hz. At low frequencies, the low order spacecraft modes could cause very high accelerations, requiring reduced acceleration input levels. At frequencies above the low order spacecraft modes, the shaker table will be able to provide adequate force inputs to the spacecraft except at the shaker table anti-resonances. During lateral excitation, the radial location of the vertical actuators inboard of the spacecraft attachment points makes it very difficult for the vertical actuator spherical couplings to react the overturning moment. The vertical actuators radial location has been moved outboard in a later design.

The shaker table design applies a lateral excitation force to the center of the table, complicating the table design and the system dynamics. A later design configuration incorporates two lateral actuators tangent to the table boundaries, simplifying the table design and improving the overall system dynamics.

Continuing work has included structural optimization of the shaker table for reducing the weight of the table while providing similar or improved performance, as well as incorporating models of the foundation including the lateral actuator support pedestals, the concrete foundation and the underlying soil.

The rigid mass modeling approach is shown to be a much more conservative approach than the flexible representation. The spacecraft has low modal effective mass at high frequencies, so the unloaded shaker table dynamic performance is improved at the higher frequencies.

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13. SUPPLEMENTARY NOTES

14. ABSTRACT
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