Chapter 33
Shaker-Amplifier System Characterization

Greta Colford, Kevin Craft, Andy Morello, Dustin Harvey, Colin Haynes, and Stuart Taylor

Abstract Equipment requirements for ground vibration testing of assemblies, sub-assemblies, and components are dependent on several factors. These include test configuration (e.g., size and mass), equipment performance parameters (e.g., stroke limit), and test architecture (e.g., single-axis or multi-axis). Equipment is historically selected based on similitude to past testing, engineering judgement, and/or rudimentary calculations based on Newton's second law of physics. From these methods alone it can be challenging to know if all desired test configurations and environments are achievable. As an alternative method to determine if a system is capable of running specific tests, a relationship can be developed between the electrical inputs and physical outputs to the system and used to predict if specific testing can be achieved. This paper explores and begins to quantify the relationship between the physical response of an electrodynamic shaker and DUT with the electrical signals of the power amplifiers and the data acquisition systems. Critical parameters of the system that impact shaker performance are identified using experimental data from three different ground vibration shaker tests. Understanding the relationship between the electrical and mechanical responses of an electrodynamic shaker system can provide test engineers with valuable information on the performance limitations of system, as well as knowledge to better utilize testing equipment resources and predict the testability of a proposed experiment. Objectives for this effort were fulfilled largely through experimental means, including post-processing of experimental data and reporting of findings to develop a knowledge database and technical approach, with an end-goal of developing metrics or tools upon which to base equipment requirements for future testing.

Keywords Electrodynamic shaker · Amplifier · Vibration · Environmental testing · Equipment requirements

33.1 Introduction and Motivation

Electrodynamic shaker systems are commonly used for a variety of vibration and shock testing applications. Choosing a vibration testing system that is mechanically and electrically capable of running the various tests on all test objects that need to be qualified can be challenging, especially if testing needs change over time. There are a multitude of parameters that can impact a shaker system's performance capability including test configuration and mounting fixtures, equipment performance parameters, and test design and architecture. Testing configuration parameters include the size, mass, and dynamic response of the device under test (DUT) and corresponding mounting fixtures. Shaker performance parameters are limited by the shaker stroke limit, peak velocity, peak force, material properties and thermal power limit of the armature coil, and armature size. The performance of a shaker system is also influenced by the maximum current and voltage outputs available from the power amplifier, as well as the outputs produced by the data acquisition drive or controller. To reduce the number of parameters that must be considered, manufacturers' rate their systems operating under a uniform set of conditions that are outlined in ISO 5344 and include a flat-band power spectral density from 20 to 2000 Hz using a non-resonant load made from a solid block of steel [1]. This approach allows shaker manufacturer ratings to be compared directly, and it defines performance capabilities under ideal conditions, but it does not clearly represent the capability of the shaker system under different conditions. Equipment for vibration testing also has been traditionally selected using basic calculations based on Newton's second law of physics and past testing experience, which can make it difficult to know if all required test configurations are mechanically or electrically possible.

G. Colford (✉) · K. Craft · A. Morello · D. Harvey · C. Haynes · S. Taylor
Los Alamos National Laboratory, Los Alamos, NM, USA
e-mail: gcolford@lanl.gov; kmcraft@lanl.gov; amorello@lanl.gov; harveydy@lanl.gov; cmhaynes@lanl.gov; sgtaylor@lanl.gov

The ultimate goal of this effort is to develop metrics or tools that predict shaker-amplifier testing capabilities of existing shakers currently in use for ground-vibration testing to better utilize shaker system equipment and provide test engineers insight to the testability of a specific DUT or test environment. This paper begins to investigate the relationship between the electrical outputs of the power amplifiers and the data acquisition system and the physical response of an electrodynamic shaker and DUT by looking at experimental data from three test series using three different shaker-amplifier systems.

33.2 Background

33.2.1 Physics of an Electrodynamic Shaker

An electrodynamic shaker works in a manner similar to a common loudspeaker, creating mechanical vibrations through the use of electrical current and the principals of electromagnetism [2]. At the center of the shaker there is a coil of wire suspended in a radial magnetic field, and when current is passed through the coil a force is generated and transferred to a structure that is often a table. The magnetic field is created using a permeable inner pole which transmits flux from one side of a magnet (or electromagnet), while the permeable outer pole transmits flux from the opposite side of the magnet. A radial flux field is created in the air gap between the inner and outer poles, and the interaction between the radial flux field and current from the coils is able to create vibrating motion [2]. The table, coil form, and coil together make up the armature assembly, to which the test object is mounted [3]. The armature is suspended and centered in the magnet structure through the use of springs or flexures. There is a proportional relationship between the current through the armature coil and the force applied by the shaker [4]. However, shaker manufacturers warn that this proportional relationship is not always accurate at all frequencies. Shaker systems consist of an electrodynamic shaker, a power amplifier, a measurement and control system, and an optional heat exchanger. The power amplifier's job is to take the drive signal from the controller and multiply the output by a fixed gain to be fed into the shaker. The heat exchanger can be used to cool the armature coil and amplifier so as to avoid overheating the hardware while in use, and it allows the shaker to operate at higher field levels. A basic schematic of an electrodynamic shaker is shown in Fig. 33.1.

33.2.2 How to Choose a Shaker

The traditional way to choose a shaker system that is mechanically capable of running a specific test uses a simple application of Newton's second law. Currently it is recommended by Senteck Dynamics, a shaker system manufacturer, to use the following method to choose a shaker system for a specific test:

First, determine the frequency range of the test, and then calculate the required peak values for displacement, velocity, and acceleration based on the test specification. The calculations to find the peak values are different depending on what type of test you are running, which can include sine, random, and shock [3]. Next, determine the total moving mass that the shaker will have to move, which includes the mass of the shaker armature, device under test (DUT), and any mounting fixtures or hardware needed to rigidly attach the DUT to the armature. Then, the total moving mass is compared to the shaker payload limits published by the manufacturer [3]. If it is within the recommended mass limits, then find the force required from the shaker by multiplying the total moving mass by the acceleration required by desired test profile, as seen in Eq. 33.1:

$$F_{required} = M_{total} * A_{peak} \tag{33.1}$$

Where M_{total} is the total moving mass that the shaker will have to move, A_{peak} is the peak acceleration determine by the desired test profile, and $F_{required}$ is the force required by the shaker. If the force required is more than the shaker is rated for, a different, more powerful shaker may be required [3].

Often shaker manufactures will suggest to apply at least a 30–40% margin of error on the calculated force (up to 100% margin in some specific test cases) to the specifications published by the manufacture to account for possibility of experiencing non-ideal conditions. However, shaker systems have been to known to exceed the published mechanical capabilities successfully in some test cases.
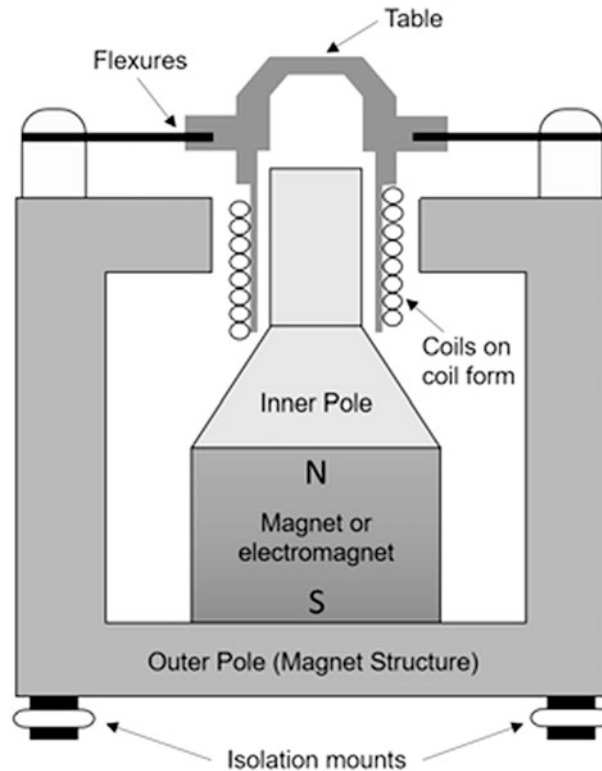
Fig. 33.1 Basic schematic of an electrodynamic shaker (cross-section)

33.2.3 Modeling of Electrodynamic Shaker Systems

Previous research efforts have modeled electrodynamics shaker systems for the purpose of characterizing the system, estimating parameters, and developing a virtual testing environment to exercise the testability of a test item. A popular approach to describe shaker-amplifier systems is the use of lumped parameter models coupled with electrical models. A basic electro-mechanical model was developed by Lang and Snyder to predict the shaker vibrational modes, the effects of isolating the shaker, and predicting maximum sine drive performance [2]. A mobility-based lumped parameters models was developed in Tiwari et al. for a medium-sized shaker, and Ricci et al. developed a coupled electro-mechanical lumped parameter model and vibration controller [5] [6]. Smallwood characterized an electrodynamic shaker system using a passive, two-port network where all variables considered are complex functions of frequency. Assumptions for this method include that the system is linear, and the output can be described by a single pair of output variables, force and acceleration of the shaker table [7]. Hoffait et al. developed a virtual shaker testing tool to predict the dynamic behavior of a coupled shaker-test specimen assembly for a specific shaker, though the methodology can be applied to other shaker systems [8]. Additionally, virtual shaker testing has been studied by Manzato et al. and Martina and Harri [9] [10].

Another method to define shaker system performance is through the use of "performance curves", which are plots of acceleration versus frequency on a log scale and are also referred to as Q-curves. The performance curves are defined by displacement, velocity, and acceleration limits of the shaker under maximum ideal conditions. In general, at low frequencies the shaker is limited by the shaker stroke, mid-range frequencies are limited by the velocity of the shaker, and the higher frequencies are limited by the maximum force (acceleration) of the shaker. Performance curves are generated using Newton's second law approach and are usually supplied by the shaker manufacturer.

These existing models (based on Newton's second law) are helpful when applied to linear systems in a specific frequency range but might not be suitable for tests requiring higher frequency ranges or complex test fixtures. One intention behind developing an empirical model and using experimental data to derive shaker-amplifier system performance capabilities is to increase the fidelity of a predictive metric or tool to account for non-ideal responses during testing like armature resonance and test object dynamics.

33.3 Experimental Data

To develop a relationship between the mechanical and electrical components and responses of a shaker system, three different test series were analyzed. Each test series ran similar ground-testing, random-vibration environments in the X, Y, and Z axes, but utilized different test objects and shakers systems. For each test, data from the amplifier current, amplifier voltage, data acquisition drive, and accelerometer responses from the shaker table and test object were collected.

33.3.1 Test Set-up

Information about the testing set-up for each of the three ground vibration tests are in Table 33.1. Information on the rated shaker performance values as well as physical details of the shaker armatures is given in Appendix A.

Test A was a ground vibration test using an Unholtz-Dickie Corp. S452–16 shaker with a SAB30F power amplifier. The test ran two different environments along with a baseline random environment to characterize the test body during testing. A head expander and slip table was used to mount the test object to the armature for the X and Y/Z axes, respectively. Two different mounting fixtures were also used to attach the DUT to the head expander and slip table, and both were tested under all three environments (Fig. 33.2).

Test B was a ground vibration test utilizing an Unholtz-Dickie Corp T-1000 shaker with a SALL-300 power amplifier. Test B ran two environments in addition to the baseline environment. The DUT in this test was mounted directly on the armature plate, without a slip table or head expander. The DUT was rotated on the table to achieve all three test axes (Fig. 33.3).

Test C was a ground vibration test on the Unholtz-Dickie T-4000-3″ shaker paired with a SAI-300 power amplifier. Test C had the largest total moving mass compared to Test A and B, and used a head expander and slip table with extension to mount the test object to the shaker armature. Three environments and the baseline random environments were run in Test C, and the DUT was also tested under multiple temperatures during the test series. The Test C DUT was also mounted to the slip table or head expander using two different mounting methods while testing the baseline environment (Fig. 33.4).

Table 33.1 Test configurations used for each test series

| Test	Amplifier/Shaker System	Test Orientation	Total Moving Mass (kg)	Fixture/Mounting Method
A	Unholtz-Dickie Corp. SAB30F / S452–16	X	100	Head expander
		Y	163	Slip table
		Z	163	Slip table
B	Unholtz-Dickie Corp. SALL-100 / T-1000	X	178	Directly on armature
		Y	208	Directly on armature
		Z	208	Directly on armature
C	Unholtz-Dickie Corp. SAI-300 / T-4000-3" CSTA	X	1400	Slip table and slip table extension
		Y	1400	Slip table and slip table extension
		Z	923	Head expander
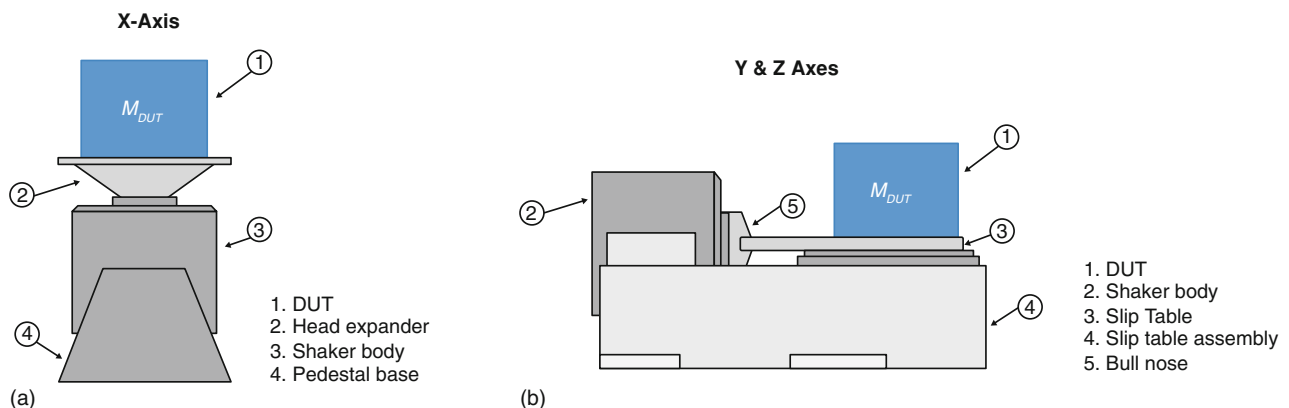
Fig. 33.2 (**a**) Test A: X-axis configuration. (**b**) Test A: Y and Z axes configuration

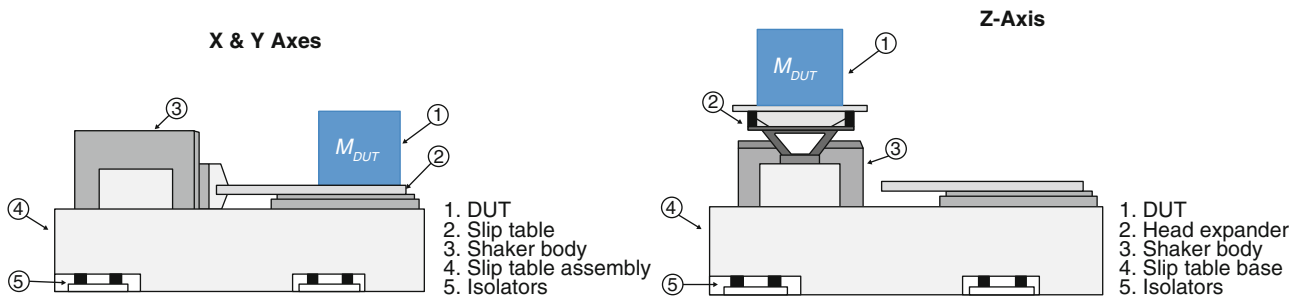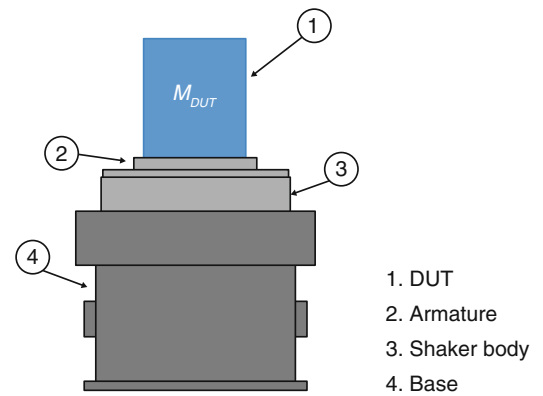**Fig. 33.3** Test B configuration. The test body was rotated to achieve testing on three axes

Fig. 33.4 (**a**) Test C: X and Y axis configuration. (**b**) Test C: Z-axis configuration

33.3.2 Test Environments

A total of five environments, plus the baseline environment, between the three tests were analyzed. The environments run for each test are outlined in Table 33.2. The table also outlines the input acceleration levels and frequency range of the test profile. The input, or control accelerometer location, was located on the shaker table for Test A and C, while the control accelerometer location was on the DUT for test B. It should also be noted that there were limits applied at other points on the DUT for some of the tests.

The baseline environment and Environment 1 were common between Tests A, B, and C and Environment 2 was common between Tests B and C. While the input acceleration may vary for each test within in a single environment, the environments are grouped together for comparison and represent similar environments across tests. For this analysis, the primary focus will be on Environment 1, Environment 2, and the baseline environment.

33.4 Analysis

For each of the three test series, the physical and electrical responses were compared and analyzed. Two different metrics were used to look at the relationship between the power amplifier electrical outputs and the physical response of the shaker and DUT and to start identifying testing parameters important to understanding shaker performance.

The first metric was computing transfer functions between the amplifier current and shaker table response, amplifier voltage and the shaker table response, and the data acquisition drive voltage and the shaker table response. This analysis provides insight on the frequency dependence of the relationship between electrical inputs to the shaker and the mechanical response. The second metric was computing the overall RMS values for each of the amplifier outputs and comparing to the overall RMS level of the acceleration response. The second approach explores the shaker-amplifier interactions under different testing conditions and environments. Applying these two metrics to experimental data under real testing conditions begins to reveal the complexities that a model must be able to capture to provide the desired predictive capabilities.

Table 33.2 Vibration environments for Test A, B, and C

Baseline				
The baseline environment was conducted in all test series and axes. It consisted of a flat spectrum random excitation from 10–2500 Hz. Each test/environment had varying levels of random excitation.				
Environment 1				
Test/ test orientation	**X**	**Y**	**Z**	**Frequency range**
Test A	0.805 G_{rms}	0.089 G_{rms}	0.418 G_{rms}	20–400 Hz
Test B	0.415 G_{rms}	0.904 G_{rms}	0.394 G_{rms}	8–1000 Hz
Test C	0.703 G_{rms}	1.46 G_{rms}	0.658 G_{rms}	10–1000 Hz
Environment 2				
Test/test orientation	**X**	**Y**	**Z**	**Frequency range**
Test B	0.123 G_{rms}	0.356 G_{rms}	0.305 G_{rms}	4–4000 Hz
Test C	0.784 G_{rms}	0.848 G_{rms}	0.570 G_{rms}	10–4000 Hz
Environment 3				
Test/test orientation	**X**	**Y**	**Z**	**Frequency range**
Test C	0.183 G_{rms}	0.082 G_{rms}	0.609 G_{rms}	4–1000 Hz
Environment 4				
Test/test orientation	**X**	**Y**	**Z**	**Frequency range**
Test B	2.78 G_{rms}	8.20 G_{rms}	9.00 G_{rms}	14–2300 Hz
Environment 5				
Test/test orientation	**X**	**Y**	**Z**	**Frequency range**
Test A	2.58 G_{rms}	0.636 G_{rms}	0.419 G_{rms}	20–1700 Hz

33.4.1 Transfer Functions

There are some trends in the literature and shaker system documentation in regard to the amplifier voltage and current outputs required by the shaker based on frequency that were seen while analyzing the data. At low frequencies, the voltage required is small, but the current required is large. At higher frequencies, the current required is less, and the voltage required is at a maximum, with the exception being at the resonant frequency. At the resonant frequency of the moving mass, the current and voltage requirements are at a minimum [11]. Another trend that was observed was the proportional relationship between the amplifier current and the force imparted on the shaker table and DUT. In Fig. 33.5, transfer functions between the amplifier current and DUT for the baseline environment event for Test A were computed for three different stages: −6 dB, −3 dB, and at full level. It can be observed that the transfer functions are overlapping, regardless of the level of the test, indicating a proportional relationship between the amplifier output current and the force produced by the armature which transfers to the acceleration response of the shaker table. This proportional trend was observed in all three test series, for all test axes and environments. Therefore, the analysis is focused on full-level events only when computing the transfer functions. The same trend can be seen for the amplified voltage and shaker acceleration response, as observed in Fig. 33.6.

Comparing Figs. 33.5 and 33.6, the overall shape of the transfer function is the same though at slightly different amplitudes. The same characteristics are present in both transfer functions, and when a transfer function between the accelerometer response and the data acquisition drive is plotted the same characteristics are observed. Electrically this makes sense, as current and voltage are related, and the voltage output from the amplifier is a scaled value of the output voltage of the data acquisition drive. Between each test environment, test axis, and test series, the relationships seen in the transfer functions for amplifier current, amplifier voltage, and data acquisition drive with the shaker acceleration response remain generally the same.

Figure 33.7 is a comparison of the baseline environment for the X test axis direction for each of the three tests, looking at the transfer function between the amplifier current and the table accelerometer response. There are different dynamic responses seen between each test, which could be attributed to the different shaker systems, testing configurations, or mounting methods used. For example, Test B had the test object attached directly to the armature table and had a flatter response over a greater frequency range compared to the other tests that had additional mounting hardware. This difference in response could possibly be connected to the DUT dynamics coupling less with the shaker or the design of the shaker system among other potential reasons. There are many complex responses seen in the experimental data, and the exact cause can be difficult to identify without additional testing. Characterizing these responses across tests and shaker systems can be helpful in understanding shaker performance and DUT response.
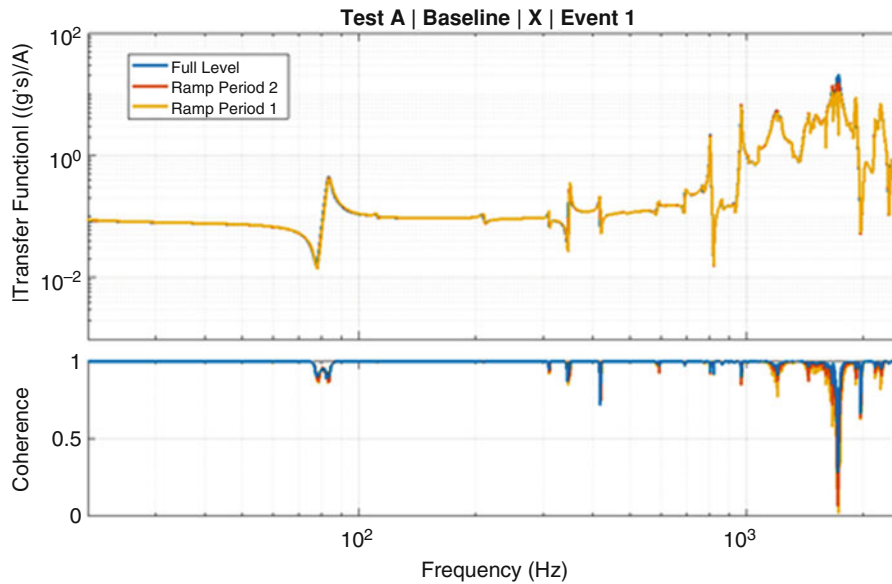
Fig. 33.5 Test A: transfer function between the accelerometer response and amplifier current for the baseline environment during ramp period 1 (-6 dB), ramp period 2 (-3 dB), and the full level test comparison
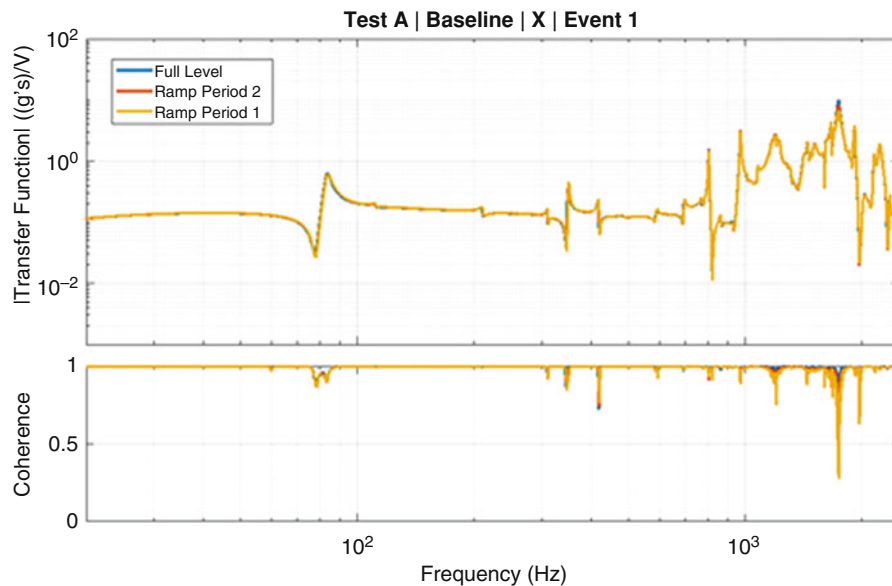

Fig. 33.6 Test A: transfer function between the accelerometer response and amplifier voltage for the baseline environment during ramp period 1 (-6 dB), ramp period 2 (-3 dB), and the full level test comparison

As expected, the largest peaks in the transfer functions are near the armature resonance values for the armature, slip table, or head expander, depending on the configuration used in the test. There is an interesting response for Tests A and C around 80 Hz, where there is a dip and rise in magnitude. The shifting seen around this same frequency in Test A and C over the course of each test may coincide with the different mounting methods used to attach the DUT to the slip table or head expander.

Figure 33.8 shows a comparison between Tests A, B, and C in the X axis for Environment 1. Similar trends can be seen in Environment 1 compared to the baseline environment, including the resonance values for the DUT, armature, and related mounting fixtures. There are some lower coherence values in this environment, but responses are slightly more similar between tests, especially between 0 and 400 Hz.
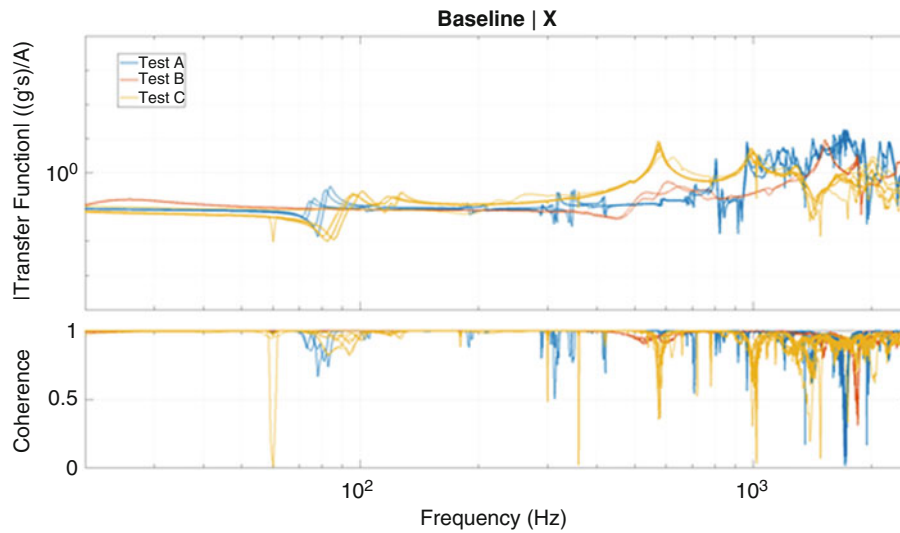
Fig. 33.7 Transfer functions between control response and amplifier current for the baseline environment
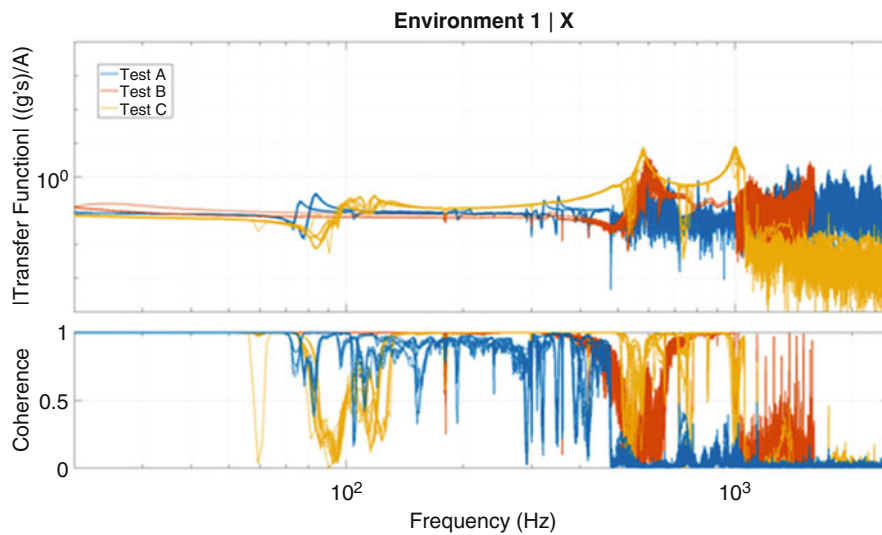

Fig. 33.8 Transfer functions between control response and amplifier current for Environment 1

33.4.2 RMS Plots

For each event in a test series, the RMS of the amplifier current, amplifier voltage, and data acquisition drive was computed and plotted with the RMS value of the corresponding acceleration response of the control locations for on-axis responses. The goal in this comparison was to find a method to map the amplifier limitations of the shaker. Fig. 33.9 is a plot of Test A with the RMS levels of all environments and test axes computed over the full frequency range of each environment. Each environment and test axis combination follows a linear trend, and the trend line could be extended and plotted with the limitations of the amplifier or shaker to find the maximum current or acceleration RMS values potentially available for a test environment. It should be recognized that these trend lines are for one DUT and will not necessary encompass all desired DUTs for this specific shaker system since total moving mass is an important parameter that impacts shaker performance.

However, when the RMS values are computed to only include the common frequency range (0 to 400 Hz) between the three environments, the trend is linear between all the environments and test orientations, as seen in Fig. 33.10. This observation indicates a potential to use RMS values over specific frequency bands to predict the acceleration level response or the amplifier current and voltage requirements.
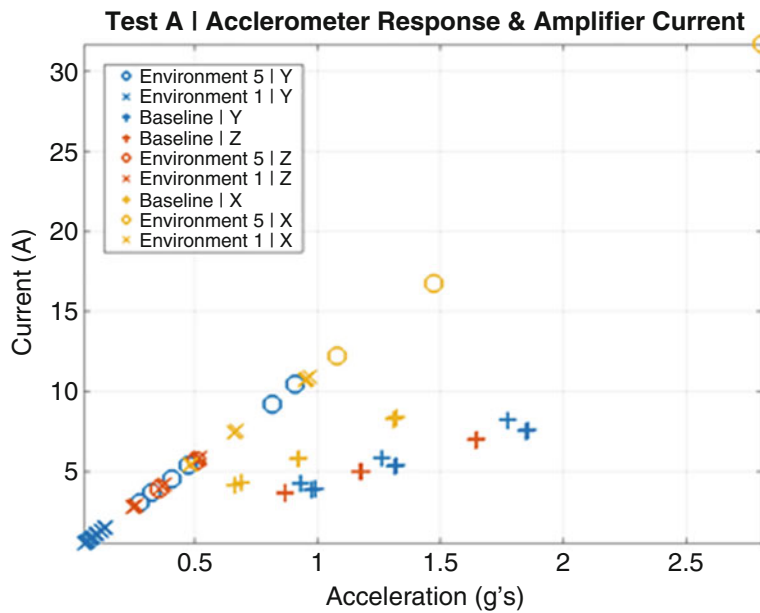
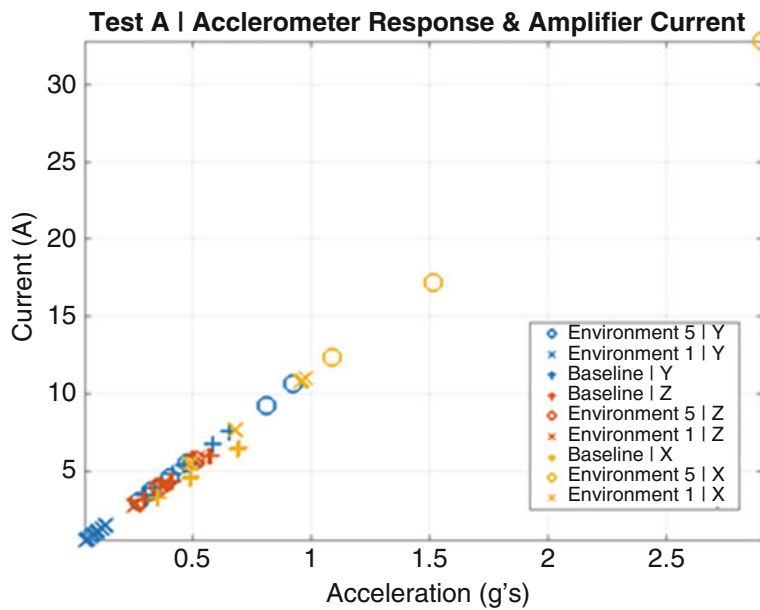**Fig. 33.9** RMS Acceleration vs. RMS Amplifier Current for Test A

Fig. 33.10 RMS Acceleration vs. RMS Current for Test A, 0–400 Hz

33.5 Conclusion

The overall goal of this work is to develop metrics or tools to predict the electrical and mechanical limitations of a shaker amplifier system and a specific DUT for ground vibration testing applications. This paper provides initial analysis and characterization of the mechanical and electrical relationship between electrodynamic shakers and power amplifiers during vibration testing. Three different ground vibration tests with varying shaker systems, test objects, and mounting fixtures were evaluated using both transfer functions and RMS values to begin to understand critical parameters that can impact shaker performance. Ground vibration testing is complex, and there are many factors that influence the dynamic response of both the shaker and the DUT. Mass, DUT mounting fixtures, and resonant frequencies all had a significant effect on the shaker-amplifier system response. Future work includes continuing to identify significant parameters for the shaker-systems using additional methods and metrics, collecting and analyzing additional experimental data with different test objects and environments, and developing and deploying a predictive capability for future test planning.

Appendix A: Shaker System Performance Specifications

Test	A	B	C
System model	SAI30F-S452–16	SALL100-T1000	SALL300-T4000–3"
Sine force (pk)	5500 lbf (24.5 kN)	11,000 lbf (48.9 kN)	40,000 lbf (178 kN)
Random force (rms)	5500 lbf (24.5 kN)[1]	8000 lbf (35.6 kN)[2]	40,000 lbf (178 kN)[3]
Shock force (4 ms, half sine)	50 g: 120 lbs. (54 kg) 100 g: 36 lbs. (16 kg)	–	50 g: 1800 lbs. (818 kg) 100 g: 700 lbs. (318 kg)
Usable frequency range	DC to 3000 Hz	–	2 to 25,000 Hz
Maximum acceleration (sine, pk)	90 g	100 g	140 g
Maximum acceleration (random, rms)	75 g	–	135 g
Maximum velocity (sine sweep)	70 in/s	70 in/s	85 in/s
Maximum velocity (shock)	130 in/s	–	165 in/s
Stroke length (pk-pk)	2 in	1 in.	3 in (76 mm)
Armature weight	60 lbs	95 lbs	285 lbs
Armature diameter	17.5 in	–	25.5 in.
Armature resonance	2300 Hz	2300 Hz	2000 Hz
Amplifier model	SAI-30F	SALL-100	SALL-300

[1]Random force rating based on flat spectrum from 20–2000 Hz with 160 lbs. (73 kg) non-resonant load
[2]Typical random rating with flat spectrum and load greater than 350 lbs. (158.8 kg)
[3]Typical random rating with flat spectrum and non-resonant load greater than 1000 lbs. (453.6 kg)

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Greta Colford graduated with a BS degree in mechanical engineering from Michigan Technological University in December 2018 and now works as a Post-Baccalaureate student engineer in the Engineering Technology and Design division at Los Alamos National Laboratory.