Challenges and Considerations When Using Hydraulic Modal Shaking in Large-Scale Modal Testing

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ABSTRACT
As test articles become dimensionally larger, more complex, and massive in weight, combined with the need to excite them to higher than traditional levels in order to identify their nonlinear characteristics, modal shakers that can generate significantly higher force levels, have longer stroke lengths, and possess higher velocity limits are required. While large scale modal tests may be performed with electrodynamic modal shakers, hydraulic modal shakers become attractive since they can generate higher force levels at lower unit cost with a smaller spatial footprint. While test engineers familiar with electrodynamic modal shakers are familiar with the challenges of displacement and velocity limits and the relatively mild shaker nonlinear distortion due to amplifier gains and shaker flexure structural geometric nonlinearities, they probably are not as familiar with the unique set of challenges hydraulic modal shakers present. These unique challenges include significant nonlinear distortion in the shaker force, issues with the setup of the hydraulic power supply and the associated hydraulic hosing, velocity limits as they relate to potentially damaging the hydraulic actuator piston, and safety issues with operating high-pressure hydraulic systems. This paper addresses these unique challenges to help the test engineer to better utilize hydraulic modal shakers on large-scale modal tests.

Keywords: Electrodynamic Modal Shakers, Hydraulic Modal Shakers, Hydraulic Power Supplies, Large-Scale Modal Test, Nonlinear Distortion.

BACKGROUND
Traditionally electrodynamic modal shakers have been used for modal testing due to their versatility, linearity, and relative ease of setup. For modal tests of small to medium size test articles, they have sufficient stroke length and can provide a sufficient range of force levels to indicate, if not fully identify nonlinear characteristics. With modal test articles becoming dimensionally larger, more complex (e.g., possessing a significant number of nonlinear mechanisms), and massive in weight, combined with the need to excite them to higher than traditional levels in order to identify their nonlinear characteristics, modal shakers that can generate significantly higher force levels, have longer stroke lengths, and possess higher velocity limits are required. While large scale modal tests may be performed with electrodynamic modal shakers, hydraulic modal shakers become attractive since they can generate higher force levels at lower unit cost with a smaller spatial footprint. Four hydraulic modal shakers were used in modal testing the National Aeronautics and Space Administration's (NASA's) Ares I-

X Flight Test Vehicle (FTV) inside the Vehicle Assembly Building (VAB) in 2009 [1 - 5]. The Ares I-X FTV was 327 feet tall and weighed 1.8 million pounds. Figure 1 shows the Ares I-X FTV inside the VAB. More recently five inertial hydraulic modal shakers were used in modal testing the Space Launch System (SLS) [6, 7] Mobile Launcher (ML) [8, 9] inside the VAB in 2019 [10, 11]. The ML weighs over 10 million pounds and is over 360 feet tall. The ML Deck supports the SLS at eight attachment points located at the bottom of its two boosters, which connect to the ML Vehicle Support Posts (VSP). The ML Tower provides lateral support to the integrated SLS launch vehicle via the Vehicle Stabilization System (VSS) and supports the fuel, power, and data umbilicals running to SLS and MPCV. The ML Tower also provides crew access to the MPCV Crew Module (CM). The ML will serve as the modal test fixture supporting the Artemis 1 integrated vehicle during its ground vibration tests referred to as the integrated vehicle modal test (IMT). Figure 2 shows the ML rolling out from the VAB to Launch Pad 39B in September 2018.

Figure 1. Ares I-X FTV Inside VAB.

Figure 2. SLS Mobile Launcher Rolling Out to Launch Pad 39B, September 2018.

Figure 3 shows the two types of hydraulic modal shakers, an inertial horizontal and an inertial vertical, used during the ML modal test in 2019. Both hydraulic modal shakers have drip pans to capture any leaking hydraulic fluid and their hydraulic power supplies were not located on the ML to prevent them from adding to the ambient background vibration environment.

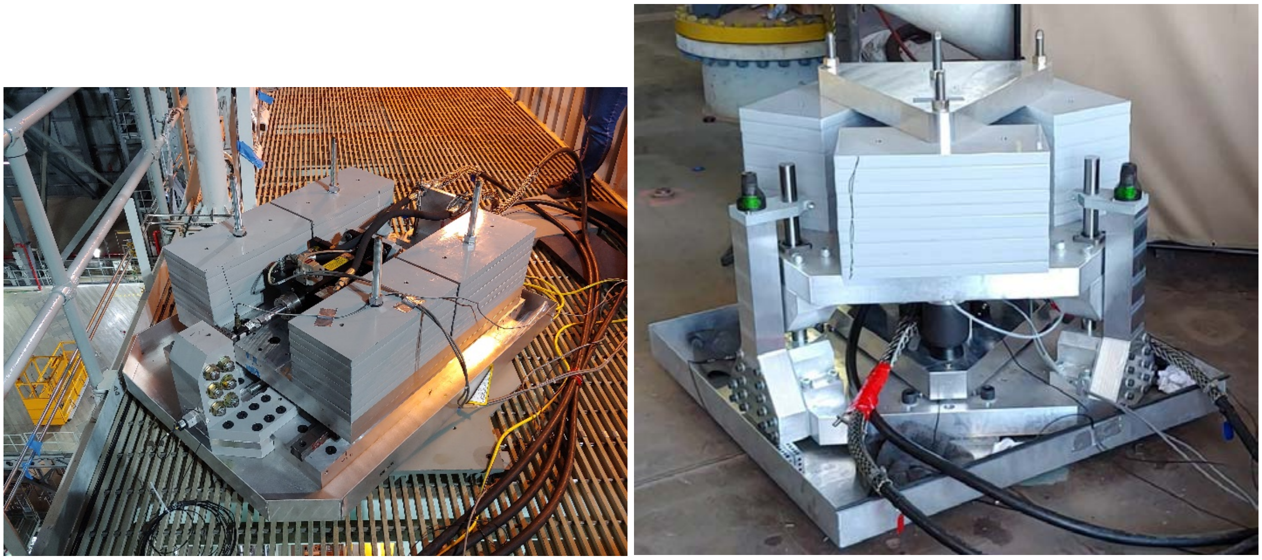

Figure 3. SLS ML Modal Test Modal Hydraulic Shakers: Inertial Horizontal Shaker (left) and Inertial Vertical Shaker (right).

While test engineers familiar with electrodynamic modal shakers are familiar with the challenges of displacement and velocity limits and the relatively mild shaker nonlinear distortion due to amplifier gains and shaker flexure structural geometric nonlinearities, they probably are not as familiar with the unique set of challenges hydraulic modal shakers present. The unique challenges associated with hydraulic modal shakers include significant nonlinear distortion in the shaker force, issues with the setup of the hydraulic power supply and the associated hydraulic hosing, velocity limits as they relate to potentially damaging the hydraulic actuator piston, and safety issues with operating high-pressure hydraulic systems. This paper addresses these unique challenges to help the test engineer to better utilize hydraulic modal shakers on large scale modal tests.

HYDRAULIC ACTUATOR BASICS

Hydraulic actuators are comprised of a piston in a housing with pressurized hydraulic fluid used to exert a pressure to the face of the piston to generate a force on the piston, which is transmitted through the piston shaft to what it is attached to with an equal and opposite force exerted on what the hydraulic actuator housing is attached to. Hydraulic actuators can be a single acting or plunger configuration, double acting configuration, or a double acting double ended (i.e. double rod) configuration as shown in Figure 4. The single acting or plunger configuration relies on what the hydraulic actuator is pushing against or an internal spring to retract the piston when the hydraulic fluid pressure is decreased (e.g., hydraulic floor jack). The double acting and double acting double ended configurations exert push/pull forces and are extended/retracted by changing the pressure differential across the piston. The double acting double ended (double rod) configuration by virtue of the piston being supported in the off-axis directions on both ends provides potentially greater lateral support that can withstand higher off-axis loads than a similar double acting hydraulic actuator.

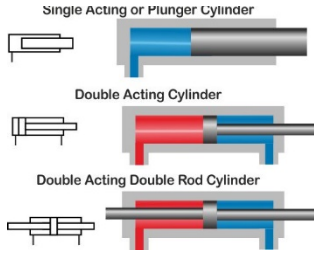

Figure 4. Hydraulic Actuator Configurations.

Hydraulic actuators used for vibration and shock testing are comprised of a stiff double acting double ended hydraulic actuator and a high frequency response electrohydraulic servo valve [12]. A servo controller sends an electrical command signal to the electrohydraulic servo valve that positions the servo valve in a single-stage servo valve or the pilot valve in a multi-stage servo valve [13]. The servo controller uses position feedback of the hydraulic actuator piston and the position of the multi-stage valves, usually measured with a Linear Variable Displacement transducer (LVDT) [14], to allow the piston to be placed in the desired starting position (usually mid-stroke). The phase lag associated with electrohydraulic servo valve influences the various feedback gains. Hydraulic actuators are inherently unstable and the servo controller is required to set the desired piston position. In most cases one would set the piston to the center position just prior to initiating the desired motion. In the case of vertically positioned actuators, one would retract all actuators prior to pressuring the system down. The servo controller, its electrical command signal to the electrohydraulic servo valve, and the actuator position and valve position signal feedback is often referred to as the "inner loop". The outer loop control is generally based on force or acceleration feedback and may be open or closed loop in nature.

The electrohydraulic servo valve is comprised of a spool or spools, if multi-stage, with lands sliding in a sleeve, which has porting grooves matched to the lands. The openings between the lands and grooves form four flow control orifices that simultaneously directs pressurized hydraulic fluid produced from the hydraulic power supply to one side of the hydraulic actuator and directs the exhausted oil from the other side back to the hydraulic power supply reservoir [15]. Higher flow rate requirements of the hydraulic actuator requires multi-stage electrohydraulic servo valves, where each stage is hydraulically, not mechanically, linked to its successor. For the double acting and double acting double ended hydraulic actuator configurations, two hydraulic fluid hoses run between the hydraulic power supply and the hydraulic actuator with the feed line supplying the higher pressure hydraulic fluid and the lower pressure return line bringing back hydraulic fluid from the hydraulic actuator to the hydraulic power supply reservoir. Figure 5 shows how the motion of the servo valve sends high pressure source hydraulic fluid (Ps) to and pulls the lower pressure return hydraulic fluid (Pr) from a double acting hydraulic actuator in order to make it extend and retract. Figure 6 shows a 4 stage electrohydraulic servo valve [15].

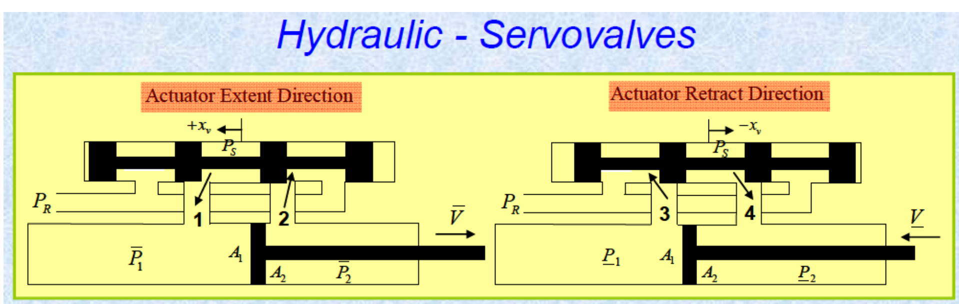

Figure 5. Servo Valve Extending and Retracting Hydraulic Actuator [2].

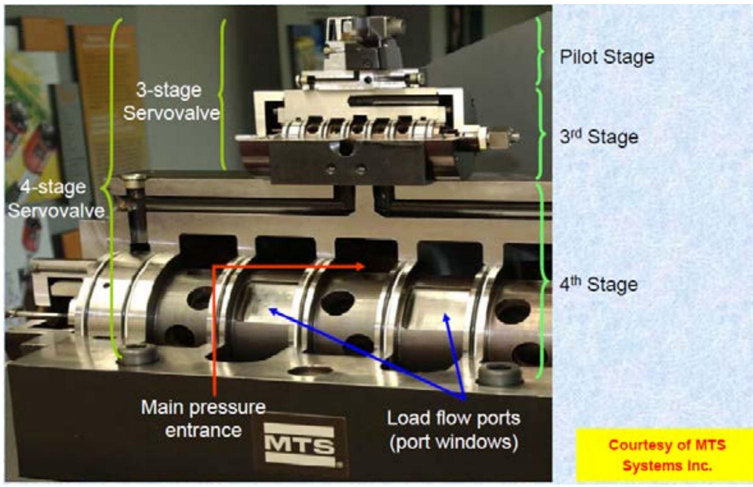

Figure 6. 4 Stage Electrohydraulic Servo Valve [15].

The servo controller in addition to generating the electrical command signal for the servo valve to move the hydraulic piston, based upon the shaker controllers drive signal, also generates a high frequency dither signal, whose frequency should be well above the modal test frequency range of interest. The purpose of the dither signal is to cause the servo valve to undergo very small amplitude oscillations (i.e., dither) to prevent sticking from occurring. The dither signal is generally set to a frequency an octave or more above the resonant frequency of the pilot valve where the system transfer function has minimal gain. While this dither signal should be high enough to be outside the modal test frequency range of interest, it may produce local responses in the test article, particularly close to drive point locations, and therefore can affect the data acquisition system (DAQ) full scale settings, particularly for these channels. Even if the dither frequency is well outside the modal test frequency range of interest, it is recommended that at least one set of data be acquired at an appropriately high sampling frequency to capture the dither signal and the test article's response to it while the shaker drive signal to the servo controller is off. In the unlikely event that the test payload of interest has a localized resonance associated with the dither frequency, the dither may be shifted (typically higher) in frequency to minimize the unwanted effects.

Hydraulic actuators have a dynamic stroke limit and a mechanical stroke limit. When the hydraulic actuator reaches it dynamic stroke limit, the piston starts to engage its dashpots that begin to bleed off the hydraulic fluid that is applying the force on the piston in order to try and reduce the piston's velocity prior to reaching its mechanical stroke limit where it impacts the housing's mechanical stops. The test engineer is cautioned not to operate the hydraulic actuators to exceed their dynamic stroke limits for two reasons. First, if the dashpots are engaged the hydraulic actuator force has significantly more nonlinear distortion due to the dashpots bleeding off the hydraulic fluid. Therefore, to minimize the hydraulic actuator's nonlinear distortion the dynamic stroke limits should not be exceeded. Second, typically the distance between the dynamic and mechanical stroke limits is relatively small, so there is a good chance that if the hydraulic actuator exceeds its dynamic stroke limit it will reach its mechanical stroke limit. Repeated contract with the mechanical stops, particularly at high velocity, can permanently damage the hydraulic actuator piston and its housing.

The hydraulic power supply consists of a hydraulic fluid reservoir tank, hydraulic pump, heat exchanger to maintain the hydraulic fluid temperature, and filtration to maintain the cleanliness of the hydraulic fluid. It is estimated that at least 75% of hydraulic system failures and maintenance activities are the result of contaminated hydraulic fluid. Fluid conditioning plays a critical role in maintaining the performance of all hydraulics equipment. This is especially true in the case of servo-hydraulic equipment where the precision clearances and high relative velocities between the moving parts place extreme demands on the hydraulic fluid [16]. Oil cleanliness should be tested on a regular basis based on system usage and any time there is a major component failure such as a damaged bearing or valve. When the system is idle, it is highly recommended that oil be continually circulated through the filtration system and that a near operational temperature be maintained. If the modal test is to be performed in either a very cold or hot environments, care needs to be taken that the hydraulic power supply heat exchanger is capable of properly maintaining the temperature of the hydraulic fluid. In addition, if the hydraulic equipment will be exposed to high humidity, oil contamination due to the introduction of water must be prevented [16]. Proper filtration and temperature control are essential for maintaining the quality of the hydraulic oil and the health of the hydraulic actuator. From a modal testing perspective, the hydraulic power supply with its hydraulic pump represents a vibration exciter that needs to be located so that it is vibrationally, and possibly acoustically, isolated from the test article. Of course, the vibration and acoustic isolation of the hydraulic power supply needs to be balanced against the available length of the hydraulic fluid hoses and their isolations from the test article as well. Therefore, vibration and acoustic isolation of the hydraulic power supplies and their hoses needs to be taken into consideration when selecting the layout of the modal hydraulic shakers.

HYDRAULIC ACTUATOR NONLINEAR DISTORTION

The source of the hydraulic actuator force nonlinear distortion is due to the servo valve not being a linear flow controller that has a nonlinear flow response, which is a function of the pressure across the valve. The oil flow through the control orifices is a function of both the area of the orifice and the square root of the pressure drop across the orifice. It is this relation of flow to the square root of the pressure drop that is a major source of the hydraulic actuator force nonlinear distortion [12,15]. A detailed discussion of the nonlinearities of the servo controller can be found in [12,15]. If the shaker drive signal is a fixed frequency sine wave, the nonlinear flow characteristics of the servo valve produces a hydraulic actuator force that contains predominantly odd superharmonics of the fundamental drive frequency. Compounding this nonlinear effect is the hydraulic actuator acts as a lightly damped single-degree-of-freedom (SDOF) oscillator that amplifies pressure oscillation close to its oil column resonance frequency. The oil column resonance frequency is a function of the hydraulic fluid column height and cross-sectional area, the effective bulk modulus of the hydraulic fluid, and the total mass the hydraulic actuator is supporting, including the mass of the piston itself. Figure 7 shows the equation for the oil column resonance frequency [15]. The effective bulk modulus of the hydraulic fluid is a function of pressure and the amount of entrained air in the hydraulic fluid. The more entrained air, the lower the effective bulk modulus (i.e., this is why we bleed our car's brake lines after working on the brake system). In addition to entrained air in the hydraulic fluid lowering its effective bulk modulus and decreasing the hydraulic actuator peak force capability, formation of air bubbles in the hydraulic fluid can lead to cavitation in the hydraulic pumps, which can cause them to fail.

Any superharmonic that is close to the oil column resonance frequency will be amplified by the oil column resonance and lead to further increased nonlinear distortion in the hydraulic actuator force. Figure 8 shows examples of the nonlinear distortion that were observed in a hydraulic modal shaker force time history during a linear sine sweep. During higher frequencies in the sine sweep, the fundamental harmonic of the modal shaker force has a higher amplitude than the unfiltered modal shaker force while at lower frequencies the opposite is true with the unfiltered modal shaker force having a significant impulsive spike occurring at the positive and negative peaks of the unfiltered modal shaker force. Figure 9 shows the corresponding PSD waterfall plot clearly showing the dominance of the odd superharmonics.

Figure 10 shows the normal probability plot of a hydraulic modal shaker force time history where the shaker drive signal was a bandwidth limited Gaussian signal [11]. The hydraulic modal shaker force exhibits significant non-Gaussian behavior because the empirical cumulative distribution function (ECD) does not lie on a straight line when plotted in a normal probability plot and has a kurtosis value of 1.5. Both the normal probability plot and the kurtosis value being less than 3 show the positive and negative excursions are not as large as that of a Gaussian distribution. The test engineer is encouraged to look at the ECD in addition to the traditional histogram (i.e., empirical probability density function (EPDF)) because the deviations from a Gaussian distribution in the tail areas (i.e., where the excursions are occurring) can be sometimes hard to see when looking at the EPDF.

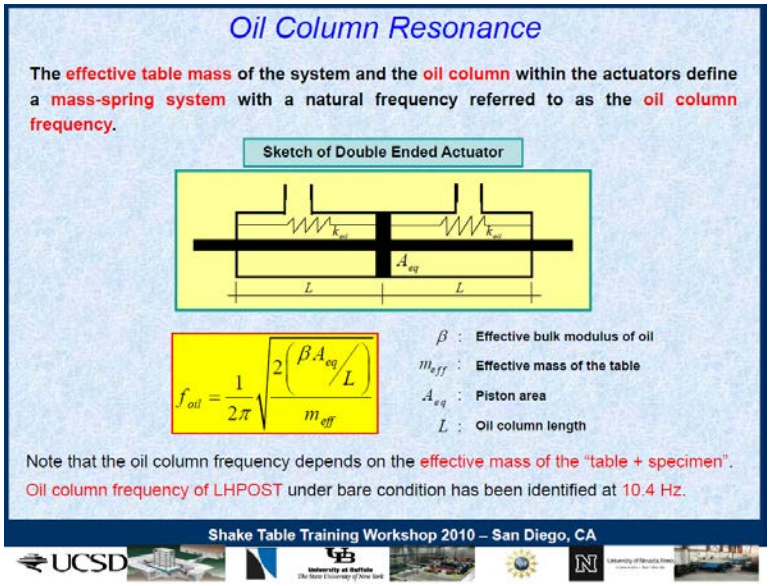

Figure 7. Oil Column Resonance Frequency.

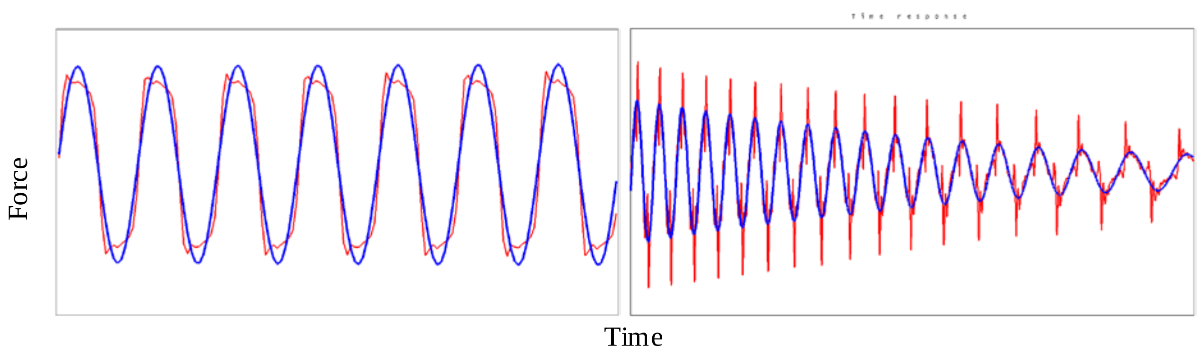

Figure 8. Hydraulic Modal Shaker Force During a Linear Sine Sweep. High Frequency (left) and Low Frequency (right). Unfiltered (red), Fundamental Harmonic (blue)

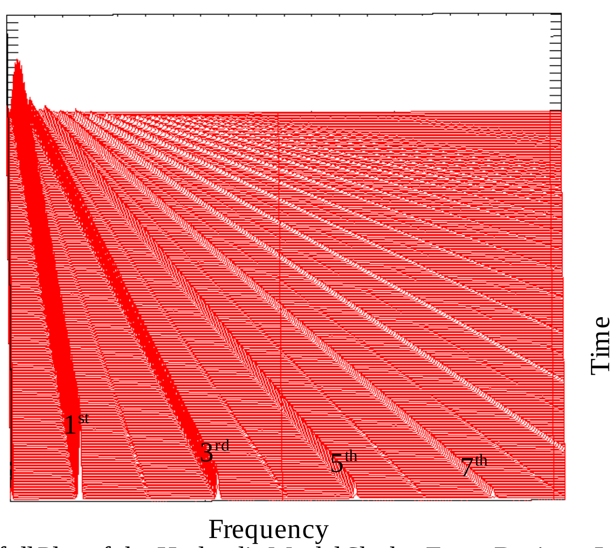

Figure 9. PSD Waterfall Plot of the Hydraulic Modal Shaker Force During a Linear Sine Sweep.

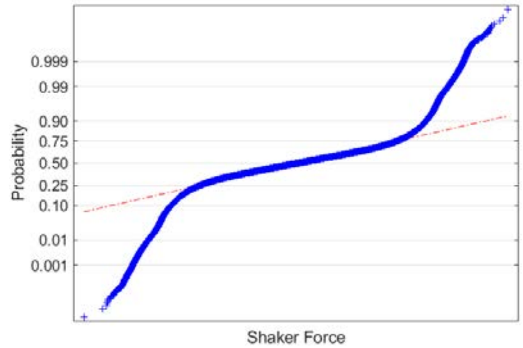

Figure 10. Normal Probability Plot of a Hydraulic Modal Shaker Force Time History With Gaussian Random Drive Signal.

Depending upon the relative phasing between the fundamental and superharmonic force components, the peak amplitude of the unfiltered shaker force time history may be greater than or less than that of its fundamental harmonic. Total Harmonic Distortion (THD) is a measurement of the amount of harmonic distortion present and is the ratio of the sum of the power of all superharmonics to the power of the fundamental harmonic [17]. A THD of 0% indicates no superharmonics present and thus no nonlinear distortion. For reference a square wave has a THD of 48.3%, a sawtooth wave has a THD of 80.3%, and a symmetric triangle wave has a THD of 12.1%. THD however does not contain information about the relative phasing of the harmonics and therefore does not provide insight into the waveform of the shaker force time history. Figure 11 shows how the relative phasing between the fundamental and superharmonic force components affects the peak amplitude of the shaker force time history. These three shaker force time histories have the same superharmonic amplitudes, and therefore the same THD, where the only difference is the relative phasing of the 3rd superharmonic.

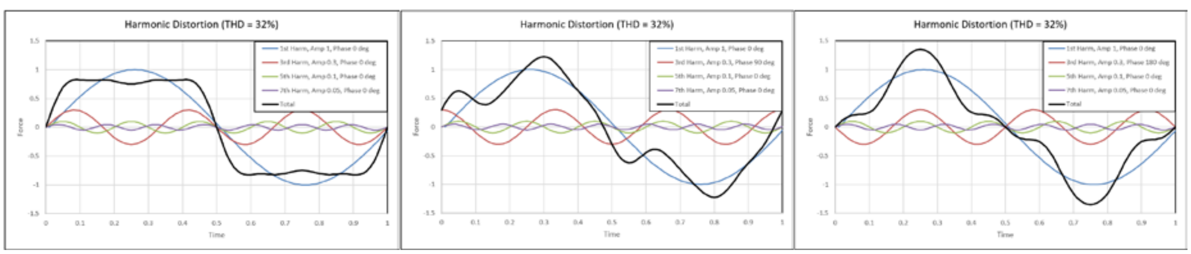

Figure 11. Harmonic Distortion, 3rd Harmonic Phase Angle 0 deg (left), 90 deg (center), 180 deg (right).

If the modal hydraulic shakers are being driven with continuous or burst random signals, the nonlinear distortion of the hydraulic actuator force may require nonlinear filtering to achieve a desired spectral content (e.g., need to adjust the shaker drive signal around 5 Hz to adjust the hydraulic modal shaker force around 15 Hz due to a dominant 3rd superharmonic).

SHAKER PERFORMANCE LIMITS

Electrodynamic modal shakers have performance limits defined by its peak displacement, peak velocity, and peak force. These three parameters in turn specify a low frequency displacement limit curve, possibly a mid frequency velocity limit curve, and a high frequency acceleration limit curve, which form straight line segments on a log frequency vs log acceleration plot. The low frequency displacement limit is a function of the peak displacement limit, which for an electrodynamic model shaker is a function of the armature flexure design and has a slope proportional to frequency squared (i.e., 40 dB/dec or 12 dB/oct). The mid frequency velocity limit is a function of the maximum current the shaker amplifier can produce and has a slope proportional to frequency (i.e. 20 dB/dec or 6 dB/oct). If the shaker amplifier is sized large enough, it does not impose a velocity limit and the electrodynamic modal shaker only has a low frequency displacement limit and a high frequency acceleration limit. The high frequency acceleration limit is simply the peak force divided by the total mass the shaker is moving (i.e., the sum of the armature mass and the mass the shaker is driving on). Figure 12 shows an example of a theoretical electrodynamic modal shaker performance limit curves, where the solid black line represents the performance curve when the shaker is pushing on 50 lbm. Note that when this shaker is pushing on 100 lbm, the acceleration limit curve is lowered sufficiently that the performance curve only consists of the displacement limit and acceleration limit curves.

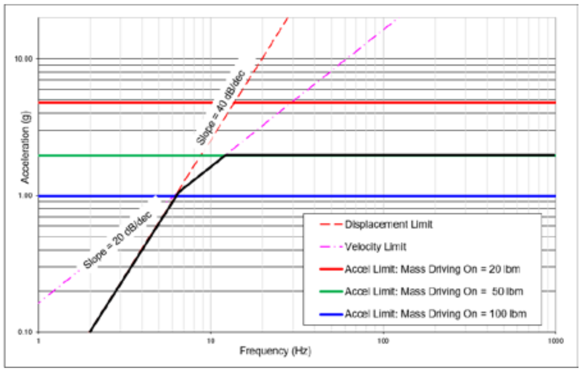

Figure 12. Electrodynamic Modal Shaker Performance Limit Curve.

Hydraulic modal shakers have their performance limits defined by its displacement limit, peak velocity, peak dynamic force, oil column resonance frequency, and servo valve corner frequency[18]. The hydraulic modal shaker's oil column acts as a second order mechanical system that above the oil column resonance frequency has a slope proportional to the inverse of the frequency squared (i.e., -40 dB/dec or -12 dB/oct). The servo valve acts as a first order mechanical system that above its corner frequency has a slope proportional to the inverse of the frequency (i.e., -20 dB/dec or -6 dB/oct). Hydraulic modal shakers displacement limits are typically set to be equal to or slightly less than their dynamic stroke limit in order to avoid engaging the dashpots or accidentally reaching their mechanical stroke limit. Hydraulic power supplies are designed to produce a specified flow rate of hydraulic fluid. The pressurization that results is the due to the resistance to that flow. Therefore, as the hydraulic modal shaker actuator piston velocity increases the hydraulic pressure decreases. Hence the hydraulic modal shakers mid frequency velocity limit is a function of both the hydraulic power supply maximum flow rate and the minimum pressure needed to adequately "float" the hydraulic actuator piston in order to prevent it from rubbing up against the housing and undergoing scoring. The plateau in the high frequency acceleration limit is a function of the peak dynamic force the hydraulic modal shaker can produce and the total amount of mass it is driving, which includes the actuator piston mass. At frequencies above the oil column resonance frequency and servo valve corner frequency, the high frequency acceleration limit curve has a slope proportional to the inverse of the frequency cubed (i.e., -60 dB/dec or -18 dB/oct). Figure 13 shows an example of a theoretical hydraulic modal shaker performance limit curves, which assumes no nonlinear distortion in the shaker force, where the solid black line represents the performance curve when the shaker is pushing on 50 lbm, the oil column resonance frequency is 70 Hz, and the servo valve corner frequency is 150 Hz. Note that when this shaker is pushing on a 100 lbm, the acceleration limit is lowered sufficiently that the hydraulic modal shaker is not velocity limited.

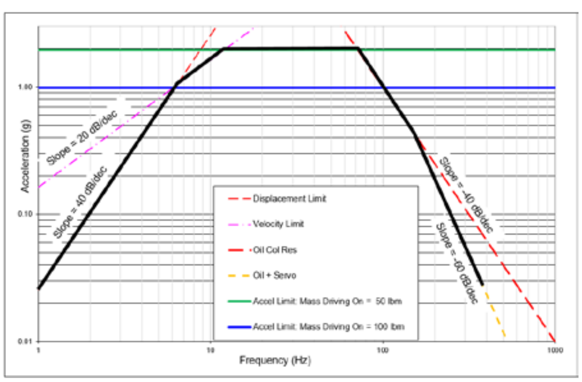

Figure 13. Hydraulic Modal Shaker Performance Limit Curve.

The electrohydraulic servo controller does not limit the hydraulic actuator displacement and therefore does not prevent the hydraulic actuator from exceeding its dynamic stroke limits, reaching its mechanical stroke limit, or exceeding is velocity limit. However, the LVDT displacement signal of the hydraulic piston is available for recording and monitoring during testing to help with this. In order to get an accurate measurement of the hydraulic actuator displacement, it is recommended that an in-situ calibration be performed where the hydraulic actuator piston is fully extended and fully retracted with the LVDT signal recorded at both locations to obtain an accurate LVDT displacement sensitivity.

There are two options for limiting the displacement of a hydraulic actuator. One option is to incorporate an outer nonlinear closed-loop controller. The other option is to perform a series open loop pretest runs to develop a combination of drive signals and servo controller gains that prevent the hydraulic actuator from exceeding its dynamic stroke and velocity limits. In either case it is recommended that test engineers monitor the hydraulic actuator displacements, hydraulic pressure, and hydraulic actuator force and be able to make manual adjustments to the servo controller gains as needed. A computerized monitoring system with video surveillance to provide real-time views of the modal shakers is very helpful in protecting the health of the modal shakers and test article [10].

As already pointed out the relative phasing between the fundamental and superharmonics force components can lead the peak amplitude of the shaker force time history at a particular frequency to be greater than or less than that of its fundamental harmonic. Therefore, the theoretical shaker performance limit curve of a hydraulic modal shaker may not be a conservative upper bound of the actual shaker performance limit curve. To generate an actual hydraulic modal shaker performance limit curve in its test setup, the test engineer can drive the hydraulic modal shaker with a sine sweep with the test engineer closely monitoring it and operating it as close as possible to its displacement, velocity, and acceleration limits. This shaker force time history can then be sine post processed using a bandpass tracking filter to generate the spectrum of the fundamental harmonic of the shaker force, which can then be further adjusted to come up with the actual hydraulic modal shaker performance limit curve. Figure 14 shows an actual hydraulic modal shaker performance limit curve. At low frequencies it is defined by a peak displacement that was set to be 75% of the dynamic stroke limit. The bucket in the knee and the knee itself in Figure 14 are a result of the velocity limit (i.e. maintaining a minimum hydraulic pressure). At high frequencies it is defined by the peak dynamic force and the amount of mass it is driving on. The oil column resonance frequency and servo controller corner frequency lie above the frequency range of interest resulting in no high frequency rolloff in this plot. Accurate actual hydraulic modal shaker performance limit curves are a key part in end-to-end force response simulations that are discussed below.

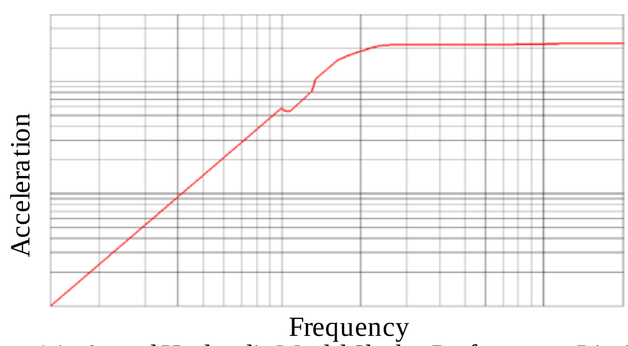

Figure 14. Actual Hydraulic Modal Shaker Performance Limit Curve.

OPERATIONAL AND SAFETY CONCERNS

If for some reason, there is a major failure in the servo controller or with the feedback signals to it (e.g., a broken feedback cable, LVDT goes bad), the hydraulic actuator has the potential to move to full stroke at full force. This failure mode is very similar to the failure mode produced by a major failure in the electrodynamic modal shaker amplifier (e.g., op amp fails). This is why personnel should not be touching or near a hydraulic modal shaker when the servo controller is on regardless of whether the hydraulic power supply is on because there can be significant stored energy in the hydraulic system. Dash pots help in such cases but there will still be a significant transient event. This potential transient event needs to be taken into account when sizing the hydraulic modal shakers, in the selection of the shaker attachment hardware, and in selecting the drive points.

Uninterrupted Power Supply (UPS) backups to the servo controllers are highly recommended to ensure the servo controllers remain under power in the event of a power outage to minimize any transient forces into the test article. This is necessary since there is still a significant amount of potential energy remaining in the hydraulic system because the pressure in the hydraulic fluid supply line does not instantaneously drop to zero when the hydraulic power supply is turned off or loses power. To ensure the UPS system is functioning as intended in the event of a power outage, an instrumented bare head test should be conducted. Simply disconnect line power and ensure that the UPS engages and that there is no transient behavior of the excitation system.

Hydraulic modal shakers are stiff compared to electrodynamic modal shakers. If a hydraulic actuator loses the command signal to its servo controller, the servo controller (inner-loop) will use all available force to keep the piston stationary at the user defined neutral position (typically center stroke), unlike an electrodynamic modal shaker whose resistance is only a function of the inertia of its armature and the stiffness of the armature flexures, which are relatively small. Hence if the hydraulic modal shaker is fixed and loses it command signal, its drive point becomes fixed and can develop significantly high loads. This is especially concerning in the case of over-actuated test configurations, where two or more hydraulic modal shakers are driving very close to each other on the test article, This could be the result of wanting to get more force into a particular location on the test article to better excite a particular mode. One solution to this issue is to integrate either active or passive delta-P circuitry into the system design. The delta-P circuitry, either electrical or hydraulic in nature, will sense pressure differences between hydraulic actuators with a common line of action and will cross port the hydraulic fluid in such a manner as to make the hydraulic actuators softer (passive), similar to the way electrodynamic exciters would behave. The effectiveness of delta-P configurations is a design parameter that should be analyzed on a case by case basis and while it will typically not be effective over the maximum operating pressure range, implementation is highly recommended for over-actuated systems. The integration of passive delta-P consists of cross porting internal to the slave valve(s) and while flow requirements to the valve increases, there is a secondary advantage in that THD may be significantly reduced. Again, this potential failure mode needs to be taken into consideration when sizing and choosing the layout of the hydraulic modal shakers.

When using hydraulic modal shakers and their associated hydraulic power supplies and hydraulic hosing, safety procedures should be followed to protect personnel, test article, and facility. Of fundamental importance is that the pressure rating of the actuators, bearings and supply lines are documented and that the hydraulic power supply is not allowed to exceed these maximum pressure limits. Safety issues related to spillage and leakage need to be taken into account in the test planning, preparation and execution. A drip pan to collect dripping/leaking hydraulic fluid from the hydraulic actuator and its hydraulic fittings may be needed. Some amount of residual leakage is to be expected and keeping the work area associated with any servo-hydraulic system clean should be considered mandatory continual maintenance. Safety procedures for inspection and detection of leaking/cracked hydraulic hoses should be implemented to prevent potential contamination of the test article with hydraulic fluid. Personnel safety procedures should also be followed to prevent personnel from being exposed to the hazards from operating hydraulic systems. Personnel safety procedures should cover hazards such as being sprayed by the hot hydraulic fluid, being hit by flailing hydraulic hoses, and when handling hoses the potential of having hydraulic fluid injected under the skin, which can have grave medical complications [19].

When using an electrodynamic modal shaker or a hydraulic shaker modal shaker as an inertial shaker (i.e., pushing against an inertial mass instead of a support stand in order to generate a force against the test article, sometimes referred to as Proof Mass Actuators), the amount of force the shaker can generated at low frequency is a function of both its stroke limit and the total inertial mass. For an electrodynamic modal shaker this total inertial mass includes the mass of its housing and field coils. For a hydraulic modal shaker this total inertial mass includes the mass of the hydraulic actuator housing. If high force levels are needed at very low frequency, the total inertial mass could be well over several hundred pounds.

Two inertial shaker options to consider are suspending the shaker or mounting the shaker on a slide system that is fixed to the test article. Suspending the shaker potentially provides the greatest amount of vibration isolation, but also incurs the additional safety constraints due to having a potentially very heavy suspended weight. In addition, for a hydraulic modal shaker, care needs to be exercised to ensure the hydraulic hoses do not structurally short the shaker (i.e., transmit shaker vibrations back into the test article). If mounting the shaker on a slide system mounted to the test article, two options to consider are roller or ball bearing rails and hydrostatic bearings. While hydrostatic bearings provide the lowest friction and smoothest support, care needs to be taken that they are energized prior to the energizing the electrodynamic or hydraulic modal shaker. For hydraulic modal shakers this typically means the hydrostatic bearing hydraulic power supplies are independent of those for the hydraulic actuators. Also, there should be a mechanical locking device to lock the hydraulic modal shaker when not being used to prevent rubbing occurring in the hydrostatic bearing when it is not energized. It is recommended that a minimal set of accelerometers be mounted on an inertial shaker to monitor its dynamics and motion during testing. For inertial shakers supported on a slide system mounted to the test article, it is recommended that sufficient in-line accelerometers be mounted to the shaker's inertial mass and an in-situ calibration using the acceleration of the inertial mass be used to verify the force being measured by the load cell used to measure the shaker force into the test article. Differences between the load cell force and the acceleration of the inertial mass can indicate stick/slip or significant friction is occurring in the slide system and the load cell force is inaccurate and leading to incorrect modal parameters, in particular mass scaling of the test mode shapes. For inertial shakers supported on a slide system mounted to the test article and pushing on a vertical post that is part of the shaker fixture, the compliance of this vertical post may need to be modeled if it is not sufficiently stiff. To verify the stiffness properties of the vertical post, strain gauges near the line of rotation and in-line accelerometers at the top and bottom of the vertical post could be installed and monitored during testing.

If piezoelectric load cells or load washers are used to attach the inertial shaker to the test article that incorporate a preload bolt in series with the load cell, it is recommended that long time constant signal conditioners be used to measure the bolt preload during installation instead of using the bolt torque. Having the correct preload is important because piezoelectric load cells or load washers need to have the proper amount of compression to work properly. The static compression provided by the preload bolt needs to be high enough that the maximum dynamic tensile loading does not cause the load cell to see less than a minimum amount of dynamic compression loading in order to remain a linear sensor. The static compression provided by the preload bolt also must not be too great so the maximum dynamic compressive loading does not exceed the structural limits of the load cell or load washer (i.e., exceed its crush limit load). In addition because the preload bolt load is in series with the load cell or load washer load, the load being measured by the load cell or load washer can be up to 20% below the true loading (i.e., up to 20% of the load is being taken up by the preload bolt). Therefore, an in-situ calibration should be performed to obtain an accurate load cell or load washer sensitivity.

Best practice is to disconnect the unused shaker from the test article so that it has no effect. For inertial shakers supported by a hydrostatic bearing, this also means the hydrostatic bearings should be energized. However, disconnecting shakers may not always be possible due to access issues. In that case it is recommended a sensitivity analysis be performed with the test article pretest FEM to determine the effect of having unused shakers attached to the test article on the test article FEM modes. This will determine if the test article FEM needs to be test configuration specific, which can potentially make the bookkeeping of the test article FEM correlation more cumbersome.

PRETEST ANALYSIS: END-TO-END FORCE RESPONSE SIMULATION

It is recommended for large scale modal tests that in addition to the standard pretest analysis, which verifies the layout of the modal shakers and sensors (e.g., accelerometers) are sufficient to identify all target modes, an end-to-end force response simulation should be performed to verify the modal shakers have sufficient force capability and the types of modal accelerometers have sufficiently low noise floors and dynamic ranges. This end-to-end simulation should include the sensor, ambient environment (i.e. vibration and electronic noise), and data acquisition noise levels. This end-to-end simulation should also include an accurate presentation of the shaker performance (i.e., shaker forces not exceeding its displacement, velocity, acceleration (peak force) limits, etc.). This will help to inform what excitation types to use; single-point, multi-point, continuous random, burst random, sine sweep, normal mode tuning. Continuous random and burst random have the simplest setup and tend to linearize the response of the test article. For simulations using either continuous random or burst random, a single recorded continuous random shaker force time history of sufficient duration can be manipulated to generate multiple uncorrelated random shaker force time histories using a "slinky approach" [11]. The "slinky approach" takes the recorded random shaker force time history and removes the first time interval, having a duration greater than or equal to the maximum frame length that will be used when processing the time histories into FRF, and appends it to the end, to generate the 2nd modal shaker uncorrelated random shaker force time history. Then this process is repeated, but now operating on the 2nd modal shaker uncorrelated random force time history to generate the 3rd modal shaker uncorrelated random shaker force time history. This process is repeated as needed to generate uncorrelated random shaker force time histories for all modal shakers. Figure 15 shows the "slinky approach" for creating three modal shaker uncorrelated random force time histories, where T_i represents the i^{th} time interval of the original modal shaker random force time history.

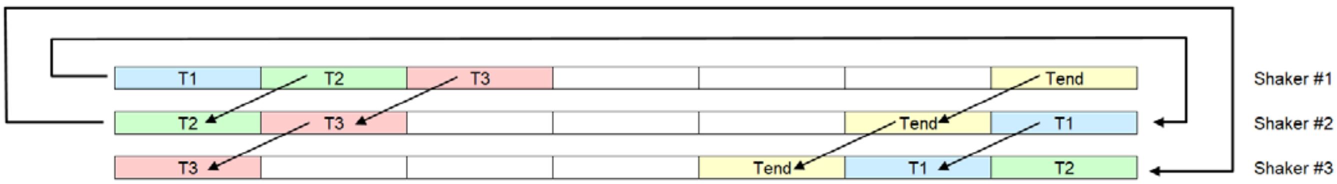

Figure 15. Generating Uncorrelated Random Force Time Histories.

If the time interval chosen has a duration shorter than the data processing frame length, theses random time histories will not be uncorrelated because from the data processing perspective, the time histories in each data block appear to be predominantly time shifted random signals, which have a significantly nonzero coherence and are thus not uncorrelated [20].

If measured modal shaker random force time histories are not available, but the modal shaker force PSD is available, uncorrelated Gaussian random signals can be filtered with a bandpass filter approximating the modal shaker force PSD to generate modal shaker uncorrelated random force time histories. Depending upon the desired crest factor, say 5, one can then set the 5σ level to be equal to the maximum modal shaker force. Then these modal shaker uncorrelated random shaker forces can be used in an end-to-end force response simulation, discussed below, and compute the peak velocities and displacements and check them against the modal shaker displacement and velocity limits, respectively, and make adjustments as necessary to ensure modal shaker limits are not exceeded. This open-loop approach eliminates the need for designing a complicated mathematical model of the modal shaker and the long run times of the associated force response analysis simulations.

Sine dwell and sine sweeps have the advantage of putting more energy into the test article at a particular frequency, which can be helpful if the noise levels are high, and can better identify the nonlinearities of the test article, but are also prone to excite unwanted rattling in the test article. Two approaches to multi-point sine dwell testing are optimal phasing [21] and frequency wrapping [22]. The optimal phasing approach has all shakers driving at the same frequency, but with different phasing. The optimal phasing approach has the advantage of putting the most energy into the test article of these two approaches, but has the disadvantage that closed-loop control of the modal shakers is needed in order to maintain their relative phasing, especially in frequency bands around the test article resonance frequencies. If the modal shakers were operated open-loop, at frequencies close to the test article resonance frequencies the modal shakers would lose their relative phasing producing corrupted FRF from which accurate modal parameters could not be extracted. The optimal phasing approach has another disadvantage in that it requires at least as many phasing combinations as shakers, so that a modal test using three modal shakers would require three sine sweeps. The frequency wrapping approach has each modal shaker driving at a different frequency at any time, and thus has the advantage of not requiring close-loop control to maintain the relative phasing between the modal shakers, but has the disadvantage that it does not as strongly exciting the test article as the optimal phasing approach. Normal mode tuning, which excites the test article at a single frequency requires closed-loop control of the modal shakers for proper scaling of the force magnitudes and phasing. Typically, modal tests use slow sweep rates (e.g., 0.5 oct/min) to give the test article time to reach close to its steady state resonance response at each mode. For a sweep frequency range (i.e., in excess of an octave) a logarithmic sweep rate is preferred instead of a linear sweep rate in order to get the same number of cycles of excitation into both the lowest frequency and highest frequency modes. However, depending on the frequency range of interest a faster sine sweep rate may be required due to the test schedule or the modal shakers thermal cooling issues. The effect of the sine sweep rate on the response of a SDOF oscillator is discussed in [23, 24].

For both large electrodynamic and hydraulic modal shakers, it is recommended a few accelerometers be mounted on them and/or their support structure to capture their dynamic motion during testing. This can help to identify unexpected vibration levels in them and their support structure that can potentially lead to off-axis misalignment issues, which in turn produce unmeasured forces into the test article that will corrupt the test data (i.e., incorrect FRF). It is recommended a mini-modal hammer tap test be performed on the modal shaker setups themselves to provide insights into whether modifications to the modal shaker setups are needed prior to when the modal test begins (e.g., better securing the modal shakers to their support stands, adding a thin foam pad between the shaker and its support stand to increase damping, etc.). Also, for large electrodynamic and hydraulic modal shakers, the shaker attachment hardware needs to be strong enough to withstand the shaker forces, while at the same time minimizing off-axis excitation into the test article. The design of the hardware connecting the actuator to the test article can be challenging and require significant design consideration. Commensurate with this, the test article needs hard-points at which these high forces can be applied. This may require analyzing the test article to ensure it can either withstand the maximum force the modal shakers can produce or determining the force limits to be set for the modal shakers.

HYDRAULIC MULTI-AXIS SHAKER VIBRATION TEST FACILITIES & BEYOND

Civil engineers have used single-axis and multi-axis shaker vibration test facilities utilizing hydraulic shakers to seismically test civil engineering structures for many decades because of the high displacement, velocity, and force requirements [15, 25 – 28]. Both the European Space Agency (ESA) and NASA have built their own multi-axis shaker vibration test facilities utilizing hydraulic shakers to test large scale aerospace flight hardware. The European Space Agency (ESA) European Space Research and Technology Center (ESTEC) HYDRA 6-DOF multi-axis hydraulic shaker vibration test facility has 8 hydraulic actuators and a custom built nonlinear shaker controller to significantly reduce the nonlinear distortion [29 – 36]. The NASA Mechanical Vibration Facility (MVF), which has 16 vertical and 4 horizontal hydraulic actuators, was designed to test NASA Multi-Purpose Crew Vehicle (MPCV) Orion class spacecraft having a total mass of 75,000 lb, center of gravity (cg) height above the MVF Table of 284 in., and a diameter of 18 ft [37 -41]. However, as structures become increasingly larger, it simply may not be practical to use either electrodynamic or hydraulic modal shakers due to the high level of the ambient vibration environment and/or due to operational constraints. Civil engineers have turned to using Operational Modal Analysis (OMA), which uses the ambient vibration environment to identify the modal parameters of civil structures [42 – 54]. Similarly, aerospace engineers have also used OMA to identify the modal parameters of large scale ground based structures [55] and of launch vehicles during flight from their flight data [56 – 58]. For aircraft and spacecraft, their modal parameters have been identified from the analysis of their response to attitude control inputs [59] and for the Hubble Space Telescope from its response to attitude control inputs and jitter [60, 61].

Even if the test engineer will be using electrodynamic or hydraulic modal shakers to excite the test article (i.e., traditional Experimental Modal Analysis), they are encouraged to expand the standard data quality checks performed on recorded ambient data to include OMA techniques. This can provide a quick look at the test article modes and in particular which modes are well excited by the ambient environment and help to determine the minimum modal shaker forces needed to obtain an adequate signal to noise ratio.

CONCLUSIONS

Hydraulic modal shakers become more attractive as test articles become dimensionally larger, more complex, and massive in weight, combined with the need to excite them to higher than traditional levels to identify their nonlinear characteristics. Hydraulic modal shakers can generate higher force levels, have longer stroke lengths, and higher velocity limits, at lower unit cost with a smaller spatial footprint compared to electrodynamic modal shakers. However, they present some unique challenges, which the test engineer needs to be aware of to successfully utilize them in a modal test. This paper has covered the basics of electrohydraulic servo valves and hydraulic actuators, what causes the nonlinear distortion in the hydraulic modal shaker force and how it manifests itself in the response of the test article, operational limits that need to be respected, and safety issues and best practices that need to be followed when operating and maintaining a high pressure hydraulic system. Keeping these things in mind, the test engineer can find hydraulic modal shakers to be a valuable tool in performing high force large scale modal tests. However, the test engineer may find that even with hydraulic modal shakers, the test article cannot be sufficiently excited or for large scale test articles that they "naturally ring" due to the ambient environment. If so, they are encouraged to consider adding Operational Modal Analysis techniques to their tool set.

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