HOW TO SELECT

A VIBRATION TESTING SYSTEM

Shopping for a controlled vibration test system can appear to be a daunting task. But, as with most technical assignments, the problem can be broken down into a sequence of smaller questions to be answered. In general, you are looking to select a system (shaker and amplifier) mechanically capable of running all of the various tests that you expect to perform on all of the different test specimens you must qualify. In addition, this shaker/amplifier must be controlled by a vibration controller with sufficient dynamic range to assure robust and repeatable control of every test with sufficient measurement channels to provide full protection and understanding of your tested device. The controller must have the necessary repertoire of testing software to cover every anticipated test type and provide for future testing expansion.

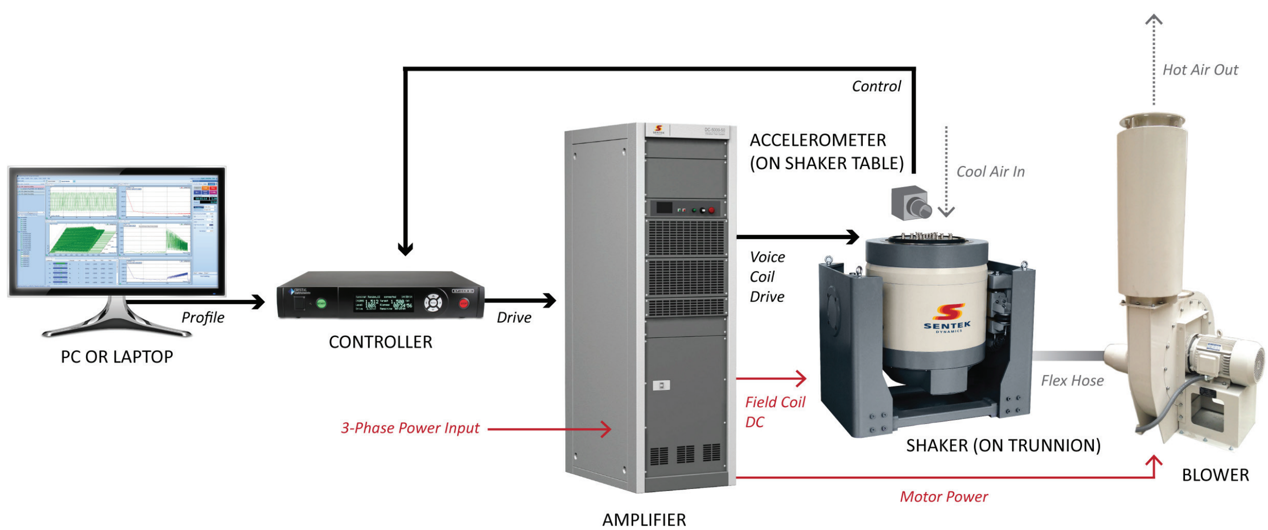


Figure 1: Basic components of an electrodynamic vibration testing system.

Finding a proper system to run a particular test on a particular specimen involves three distinct steps:

1. Determine the *Frequency Range* of the test and the extreme *Peak* motions (*Acceleration, Velocity* and *Displacement*) from the *Test Specification*. Depending upon the type of test (Swept Sine, Random, or Classical Shock for example) you will perform different calculations to accomplish this.

2. From the *Device Specifications* determine the mass of the device under test (DUT) and the dimensions of the mounting surface required to bolt the DUT to a shaker table with its center of gravity (CG) over the table's center. This may require use of a head expander and/or other mounting fixture. Determine the mass of such additional hardware including all mounting fasteners.

3. Look at the *Shaker Specifications* of a trial system. Verify that the mass of the DUT and any mounting fixtures are less than the shaker's rated *Maximum Static Payload*. Verify that the DUT (or its mounting hardware) can be bolted to the shaker's armature or head expander. Add the *Effective Armature Mass* and head expander mass to the DUT mass and the mass of any mounting fixtures and fasteners and any accelerometers mounted to the DUT – this is the *Total Moving Mass* that must be moved by the shaker during the test. Multiply the *Total Moving Mass* by the *Acceleration* determined in step one to calculate the Force required. Verify the determined test *Frequency Range, Force, Peak Acceleration, Peak Velocity* and *Peak Displacement* required by the test and DUT are all less than the corresponding performance ratings of the selected shaker system. If this is true, the selected shaker is acceptable. If not, a more powerful shaker is called for.

Important Background Information

Before we discuss each of these steps in detail, let's review the basic operation of an electrodynamic shaker system and understand what each of its performance specifications tells us. As an example, Figure 2 presents performance specifications for Sentek Dynamics medium force systems. Sentek Dynamics offers smaller and larger systems; all are described by the same set of performance specifications.

Figure 3 provides a schematic representation of a typical shaker. The cylindrical *armature* is the shaker's moving structure. It consists of an *upper load* table to which the DUT is bolted and a lower *coil form*. The armature sits on a soft *air spring* which provides a soft vertical spring rate for the *internal load support system.* Pressure within the air spring may be adjusted to support and vertically center any payload up to the *Maximum Static Payload*. The armature is held concentric with the shaker body by an *Upper Suspension* and *Lower Suspension*. The upper suspension is typically a series of radially dispersed and elastically retained rigid rolling-contact "dog bone" shaped guides. The lower suspension is a set of four rollers bearing on a square shaft. These suspension elements stiffly restrain the armature from rocking, twisting or moving laterally. They allow it to move freely in the axial (vertical) direction against the soft spring rate of the air spring. The *Effective Armature Mass* is the mass of the armature and the additional effective inertia of the suspension elements that move with it. The *Armature Diameter* describes the maximum diameter of the load table.

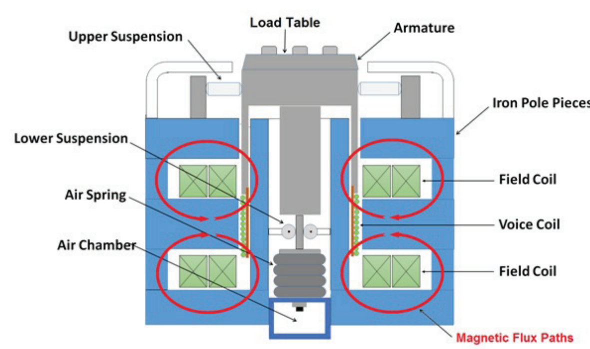


Figure 3: Schematic representation of a Sentek Dynamics electromagnetic shaker.

System Performance	M1528A	M2232A	M3240A	M4040A	M5044A	M6044A	M6544A
Sine Force Peak kN (lbf)	15 (3300)	22 (4840)	32 (7040)	40 (8800)	50 (11,000)	60 (13,200)	65 (14,300)
Random Force rms kN (lbf)	15 (3300)	22 (4840)	32 (7040)	40 (8800)	50 (11,000)	60 (13,200)	65 (14,300)
Shock Force (6 ms) kN (lbf)	30 (6600)	44 (9680)	64 (14,080)	80 (17,600)	100 (22,000)	120 (26,400)	130 (28,600)
Frequency Range (Hz)	5 - 3000	5 - 3000	5 - 2500	5 - 2200	5 - 2500	5 - 2500	5 - 2500
Continuous Displacement mm (in)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)
Shock Displacement mm (in)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)	51 (2.0)
Max Velocity m/s (in/s)	2.0 (78.7)	2.0 (78.7)	2.0 (78.7)	2.0 (78.7)	2.0 (78.7)	2.0 (78.7)	2.0 (78.7)
Max Acceleration Sine Peak m/s2 (g)	784 (80)	980 (100)	980 (100)	980 (100)	980 (100)	980 (100)	980 (100)
Armature Diameter mm (in)	280 (11.0)	320 (12.6)	400 (15.8)	400 (15.8)	445 (17.5)	445 (17.5)	445 (17.5)
Effective Armature Mass kg (lb)	18 (39.6)	22 (48.4)	32 (70.4)	40 (88.0)	49 (107.8)	49 (107.8)	49 (107.8)
Max Static Payload kg (lb)	300 (660)	300 (660)	500 (1100)	500 (1100)	1000 (2200)	1000 (2200)	1000 (2200)

Figure 2: Typical performance specifications for a shaker family.

A copper wire *Voice Coil* is wound about the bottom portion of the armature. This coil sits in a strong (static) radial magnetic flux path produced by two DC-excited *Field Coils*. When a current is passed through the voice coil, a corresponding axial force acts upon the armature. This force is proportional to the voice coil current, to the length of wire in the magnetic field and to the magnetic flux density (in turn proportional to the field coil DC current). The system *Amplifier* drives both coils. It provides a DC current to the field coils, providing a strong static radial magnetic flux. It provides an AC current to the voice coil in proportion to a *Drive* input signal provided by the *Vibration Controller.* The net result is that the armature moves up and down in response to the *Drive* command provided by the controller. The shaker acts like a gigantic ruggedized loudspeaker.

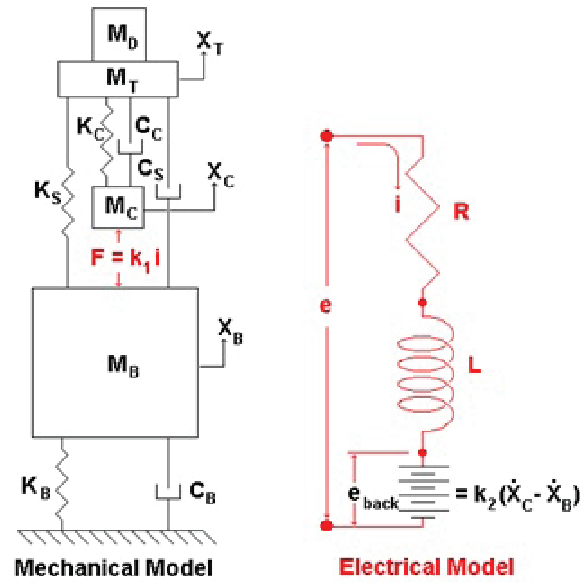


Figure 4: The mechanical and electrical actions of a shaker are cross-coupled.

The electrodynamic shaker reflects strong interplay of electrical and mechanical dynamics. The mechanical system is excited by a force proportional to electrical current, while the electrical circuit is excited by a generated voltage (back-EMF) proportional to the mechanical velocity of the voice coil. The amplifier drives the electrical circuit providing an external voltage, **e**, and driving current, **I**, as shown in Figure 4.

Figure 4 provides some insight into the physics that limit a shaker's performance. In the lumped-mass model on the left, the armature is represented by the two elastically coupled masses $\mathbf{M_T}$ (the load table) and $\mathbf{M_C}$ (the coil form). The DUT is represented by appended mass, $\mathbf{M_D}$. The shaker body is represented by $\mathbf{M_B}$. The spring rate $\mathbf{K_S}$ and the damping rate $\mathbf{C_S}$ represent the air spring supporting the armature and the DUT. The stiffness $\mathbf{K_C}$ and (very low) damping rate $\mathbf{C_C}$ represent the coil-connecting structure of the armature. The stiffness $\mathbf{K_B}$ and damping rate $\mathbf{C_B}$ model the *vibration isolators* that support the body of the shaker. **R** represents the voice coil *resistance* and **L** represents its *inductance*. The constants $\mathbf{k_1}$ and $\mathbf{k_2}$ reflect the electromechanical coupling (current-to-force and velocity-to-voltage).

Three modes of vibration dominate the mechanical signature and reflect in the electrical signature. The mode shapes of these three modes are shown in Figure 5. At very low frequency (2-3 Hz), the compliant isolation mounts allow the entire shaker to translate as a rigid body with almost no relative motion between its components. This deformation shape is termed the *Isolation Mode*. The *Suspension Mode* occurs next, near the lower end of the shaker's *Frequency Range*. In this mode, the table and coil move together as a rigid body relative to the shaker body. Motion in this mode is limited by the design *Stroke* of the machine. When the shaker is properly loaded and centered in its stroke, the frequency of the suspension resonance is (essentially) constant, regardless of the payload attached to the load table. The air spring pressure is adjusted manually against an armature-height alignment gauge or automatically (if equiped with *automatic armature centering*) by adding shop air or venting after mounting a new DUT. Air is added to support the mass of the paylod, that is the air spring takes on a supporting preload to balance the added mass. However the air spring also stiffens in proportion to its internal pressure. Since the suspension stiffness tracks the payload mass, the suspension resonance frequency remains unchanged. At or beyond the high frequency limit of operation, the (undesired) *Armature Resonance* is encountered. Here, the coil moves out-of-phase with the drive table and the elastic armature structure is stressed as a result. Excessive excitation in this mode can seriously damage the armature.

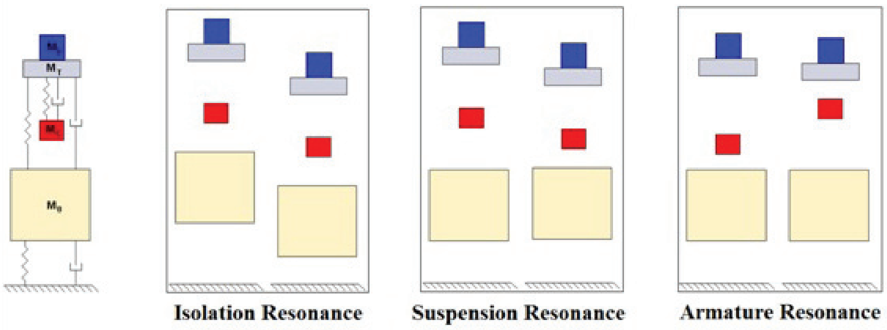


Figure 5: The mode shapes of the three dominant shaker resonances.

The voltage/current (voice coil impedance) and three acceleration/current frequency response functions (FRF) of Figure 6 illustrate typical responses of the model of Figure 4. Note the two sharp resonance spikes. These essentially mark the *Frequency Range* over which the shaker may be used. The lower frequency spike (here at ≈7 Hz) is the *Suspension Resonance* while the upper one (≈5 kHz) is the *Armature Resonance*. The 7 Hz suspension resonance is clearly reflected in the coil impedance; the 5 k Hz armature resonance reflects too, but less clearly. The *Isolation Resonance* is less clearly seen. It produces the slight "kink" evident (between 2 and 3 Hz) in the in the three acceleration/current FRFs. The isolation mode does not reflect into the coil impedance because this mode's shape has no relative motion between the coil and the shaker body

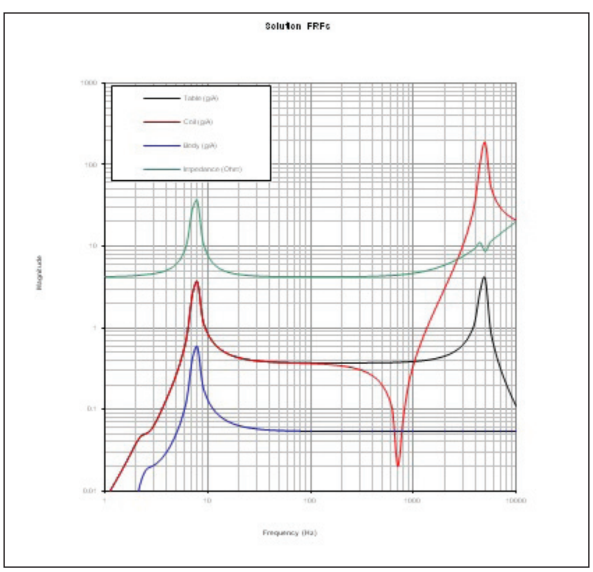


Figure 6: Solution FRFs show voice coil voltage and mass accelerations per ampere of drive current.

Isolation of the shaker is absolutely required to keep the machine from unduly shaking the building that houses it. Standard configuration Sentek Dynamic shakers provide air spring isolators at the trunnion mount or under the feet of a monobase supporting a shaker and slip table. These soft air springs support the shaker body and minimize vibratory forces passed to the laboratory floor. Figure 7 illustrates how successful isolation is. Here, maximum possible acceleration sweeps using the same payload are compared. The tests differ only in the isolation configuration used. The black trace (trunnion isolation) shows an order-of-magnitude reduction in floor reaction forces over a rigidly mounted shaker (red trace). The blue trace illustrates the further reduction of building forces possible by mounting the shaker (rigidly) to a large reaction slab and isolating that slab from the building foundation. This, of course, is a far more expensive solution.

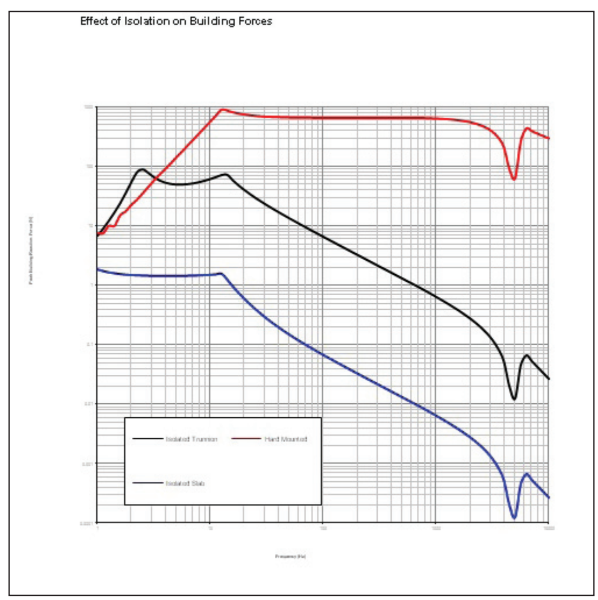


Figure 7: Reducing floor vibration by isolating the shaker.

However, vibration isolation is not without cost. Figure 8 shows the table acceleration achieved in these three tests. (Note the hard-mounted and slab isolated traces are virtually indistinguishable.)

The trunnion isolated test shows a reduction in the maximum possible DUT acceleration from the isolation resonance frequency to the suspension resonance frequency. This reduction occurs because *part of the shaker's stroke is expended to accelerate the shaker body away from the armature.*

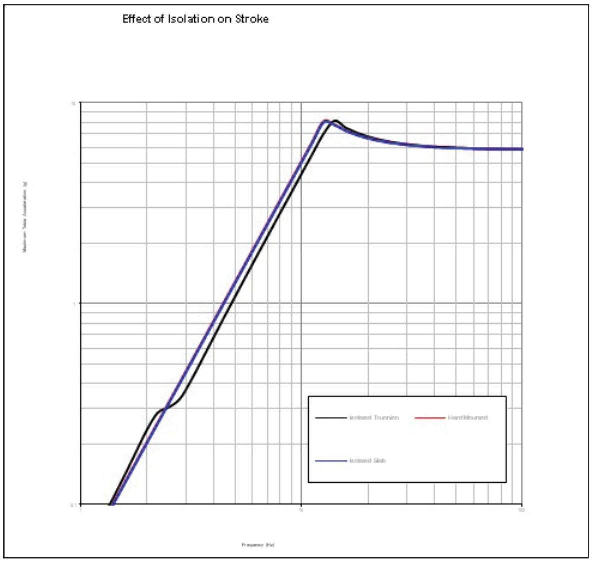


Figure 8: Comparison of maximum acceleration shakes with different isolation configurations.

For this reason, the specified Stroke (maximum peak-to-peak displacement) of an isolated shaker must always be de-rated (reduced). De-rating involves the Total Driven Mass (**M_{td}**) and the Isolated Body Mass (**M_b**) of the shaker; specifically:

$$Stroke_{derated} = \left(\frac{M_b}{M_b + M_{td}}\right) Stroke_{specified}$$

Clearly, the *Continuous Displacement* and *Shock Displacement* can never exceed the shakers peak-to-peak *Stroke*. In examining the feasibility of running a particular test, these displacement parameters should never exceed the *De-rated Stroke* of the shaker. Now let's look at the other limiting parameters.

The system's *Maximum Velocity* specification is actually a reflection of the amplifier's maximum voltage. Recall (from Figure 4) that a back EMF voltage is generated by the voice coil in proportion to its velocity relative to the shaker body. If this generated voltage should exceed the maximum voltage the amplifier can provide, the system cannot maintain control of the armature's motion. Hence the maximum velocity specification is a feedback control issue and not a structural limit. Assuring that the test does not exceed the system's maximum velocity is normally not a significant concern for random tests – it is of some concern for low frequency sine sweeps and it is of paramount concern for shock tests.

The *Maximum Acceleration* (*Sine Peak*) is a structural concern. Exceeding this maximum peak acceleration can result in damage to components of the shaker. This peak acceleration is a reflection of the stress acting within the armature. It is always prudent to resist over accelerating the armature transiently or continuously.

In like manner, the three force ratings (*Sine Force Peak, Random Force RMS* and *Shock Force Peak*) are "do not exceed" levels for the selected test system. They are measured in accordance with ISO 5344 which specifies the payload, random test spectrum and endurance durations. The force ratings are actually reflected levels of the maximum RMS current that may be applied to the voice coil without thermally damaging it. When the shaker is tested at its maximum sine force or to its maximum random force, the RMS current flowing through the voice coil is essentially the same. Hence the voice coil experiences the same 1^2R thermal power for the duration of either continuous signal qualification. The peak shock force level may be considerably larger (2X to 3X) than the peak sine force because it only persists for a short duration. A higher level of transient current (reflecting a higher force) is tolerated by the armature's voice coil.

Step 1 – Swept Sine Testing

The simplest test specification to analyze is that of a Swept Sine test. Figure 9 illustrates the Test Profile of a swept-sine test. The profile is an acceleration-verses-frequency spectrum, normally presented in log-log format. The acceleration amplitude is normally expressed in peak "**g**" units, where 1**g** =9.807 m/s^2 = 386.4 inch/sec^2. The frequency, f, is normally expressed in Hz (Hertz meaning cycle/second), though circular frequency, **ω**, in radian/second (1/s) is sometimes used. (Note **ω**=2π **f** = 6.2832 **f**.)

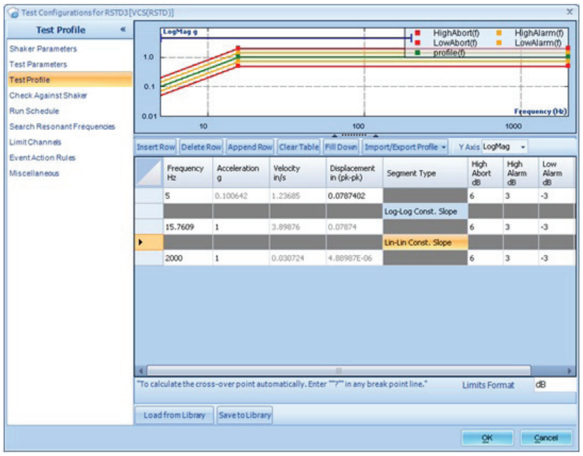


Figure 9: A Swept-Sine Test Profile is a peak acceleration (normally in g's) versus frequency spectrum.

The *Frequency Range* of the test is simply the low-to-high frequency span of this spectrum. The *Peak Acceleration* is simply the largest amplitude of this spectrum.

The test profile is (normally) composed of a series of straight-line segments. Note the frequency, **f**, and acceleration, **g**, of each point defining the end of a line segment. Calculate the corresponding peak velocity and peak-to-peak displacement for each endpoint using the following equations:

1) $Velocity_{Peak} = \frac{9.807}{6.2832}\frac{g}{f} = 1.561\frac{g}{f}\ (ms^{-1})..or..\frac{386.4\,g}{6.2832\,f} = 61.50\frac{g}{f}\ (IPS)$

2) $Displacement_{PTP} = \frac{2\,x\,9.807\,g}{(6.2832\,f)^2}\ 0.4968\frac{g}{f^2}\ (m)..or..\frac{2\,x\,386.4\,g}{(6.2832\,f)^2} = 19.57\frac{g}{f^2}\ (inch)$

Retain the largest result from equation 1) as the *Peak Velocity*. Retain the largest result from equation 2) as the *PTP Displacement*. These actions are summarized for the Test Profile of Figure 9 in the following table:

Frequency	Acceleration	Velocity = 61.5 g/f	Displacement = 19.57 g/f²
Hz	**g_{peak}**	**IPS_{peak}**	**$inch_{peak\text{-}to\text{-}peak}$**
5.	0.100642	1.23685	0.0787402
15.7609	1.0	3.89876	0.07674
2000.	1.0	0.030723	4.88987×10^{-6}
Maxima:	**1.0 g_{peak}**	**1.23685 IPS_{peak}**	**0.0787402 $inch_{peak\text{-}to\text{-}peak}$**
Frequency Span			**5 – 2000 Hz**

Step 1 – Random Testing

Random testing involves generating and controlling a random signal with a specifically shaped spectrum and Gaussian amplitude statistics. The signal is defined by a spectral test profile called a *Power Spectral Density* (PSD) which is always measured and it exhibits an amplitude histogram or *Probability Distribution Function* (PDF) which can also be measured. Figure 10 illustrates these two complimentary views of a random acceleration.

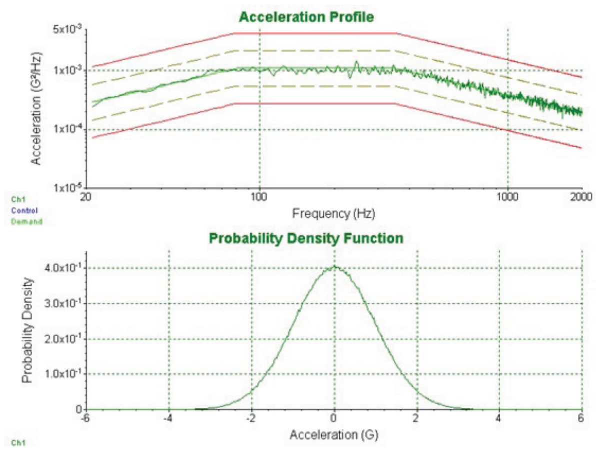


Figure 10: Two views of a random acceleration – PSD upper, PDF lower.

Like a swept-sine test, a *Random* test has an acceleration spectrum as the *Test Profile*. However, the vertical units of the random spectrum are very different; they are in units of **$(g_{RMS})^2$/Hz** conventionally abbreviated to **g^2/Hz**. Such a "squared amplitude" spectrum is called a *Power Spectral Density* or PSD. The area under a PSD is the signal's "power" or mean-square value. Hence the area under a **g^2/Hz** acceleration is the square of the signal's root-mean-square (RMS) value, **g_{rms}**. The "per Hertz" part of the **g^2/Hz** unit label refers to the resolution *Noise Bandwidth* of the instrument that measured the PSD.

A modern vibration controller measures a PSD by using *Fast Fourier Transform* (FFT) processing. The FFT works upon sequentially gathered blocks of **N** successive signal amplitude samples separated equally in time by **Δt** seconds. It mathematically converts these into an **N/2** point spectrum with spectral amplitudes measured every **Δf = 1/NΔt** Hertz. A "power spectrum" is produced by squaring the spectral amplitudes and ensemble averaging many squared spectra.

The averaged *Power Spectrum* becomes a *Power Spectral Density* when the amplitudes are divided by the *Noise Bandwidth*. The noise bandwidth of such a measurement is **kΔf**, where **k** is determined by the "window" or weighting applied to each acquired block of **N** time samples. A typical value of **k** is 1.5 when the (very common) Hanning window is applied. A window functions (such as Hanning) is necessary to suppress spectral distortions resulting from the asynchronism between the acceleration signal and the analyzer.

Because the random testing signal exhibits "bell-shaped" Gaussian amplitude statistics, the peak value, $\mathbf{g_{peak}}$, can be precisely estimated from, $\mathbf{g_{rms}}$. The PDF is an amplitude histogram (counts or occurrence versus signal value) normalized to enclose a unit area. In vibration work it is frequently useful to present a PDF with its amplitude on a log scale as in Figure 11.

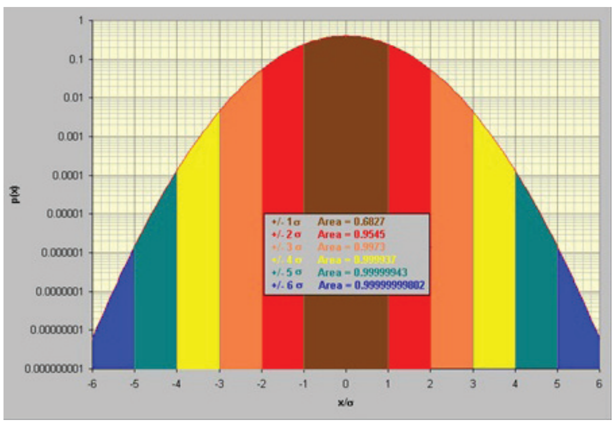


Figure 11: Two views of a random acceleration – PSD upper, PDF lower.

The horizontal (signal amplitude) axis can be scaled to multiples of **σ**, the standard deviation synonymous with $\mathbf{g_{rms}}$ when the signal is of zero *mean value*. Figure 11 illustrates such scaling and shows the fraction of area subtended by ±1, 2, 3, 4, 5 and 6**σ** amplitude bands. Since the total are under the PDF is unity, these areas represent the probability of the signal being within the ± bounds at any time. Figure 11 shows that a Gaussian acceleration signal of RMS value, $\mathbf{g_{rms}}$, will have an instantaneous peak value within ±3 $\mathbf{g_{rms}}$ 99.73% of the time. In other words, if the random signal has an RMS value of $\mathbf{g_{rms}}$, 3 $\mathbf{g_{rms}}$ is a very good estimate of the peak acceleration.

Hence, to determine the *Peak Acceleration* of a random test, we will find the $\mathbf{(g_{rms})^2}$ area under the Power Spectral Density curve, take its square root and multiply it by 3. The required *Peak Velocity* and *Peak Displacement* terms can be found by synthetically integrating and double integrating the Acceleration PSD (dividing the amplitude of that spectrum by **ω=2πf** and by $\mathbf{\omega^2}$) and repeating these actions.

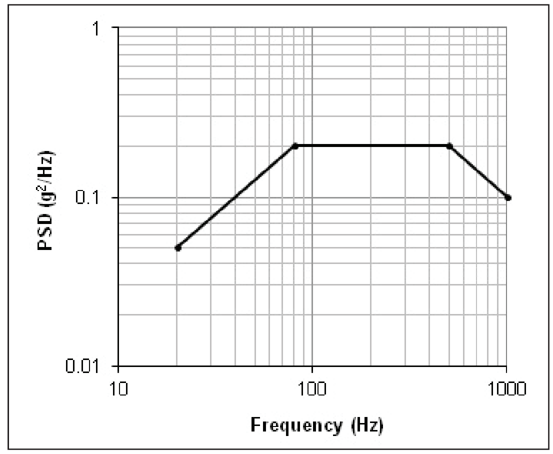


Figure 12: A typical random test profile displayed in log/log format.

At first glance, it would appear that calculating the area beneath the random test's PSD would be a simple matter. Most test specifications contain a profile composed of straight-line segments, so that the required area is made up of triangles, rectangles and trapezoid – shapes with simple area formulas. But those spectra present the $\mathbf{g^2/Hz}$ versus frequency on a *log/log* plot such as Figure 12. When the same spectrum is presented as a linear/linear graph (such as Figure 13), it becomes obvious that the inclined "lines" are actually curves.

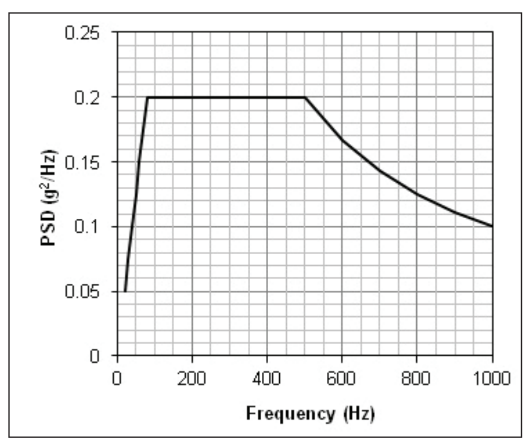


Figure 13: The test profile of Figure 12 displayed in linear/linear format.

In order to find the area under each (**log A** versus **log f**) PDF segment, we will need to have the **A(f)** equation for each curve. On the log/log plot, each straight-line segment can be described by:

y=mx+b

Where:

$$y = \log_{10}(A)$$

$$x = \log_{10}(f)$$

$$m = \frac{\log_{10}(A_2) - \log_{10}(A_1)}{\log_{10}(f_2) - \log_{10}(f_1)} = \frac{\log_{10}\left(\frac{A_2}{A_1}\right)}{\log_{10}\left(\frac{f_2}{f_1}\right)}$$

$$b = \log_{10}(A_1)$$

Hence:

$\log_{10}(A) = m \log_{10}(f) + \log_{10}(A_1)$

From which:

1) $A = A_1 f^m$ (g^2/Hz)

We will also need the equivalent equations for segments of the velocity and displacement PSDs. These may be obtained by synthetic integration of equation 1. Recall **A** is of dimension **g^2/Hz** where **g** is understood to be an RMS measurement. We require a velocity power spectrum, **V**, of **IPS^2/Hz** amplitude and a displacement power spectrum, **D**, of **$inch^2$/Hz** dimension where both **IPS** and **inch** are understood to be RMS units. The required segment equations are:

2) $$V = \left(\frac{386.4}{2\pi f}\right)^2 A = 3782\, A_1 f^{m-2} \quad (\text{IPS}^2/\text{Hz})$$

and

3) $$D = \left(\frac{386.4}{(2\pi f)^2}\right)^2 A = 95.80\, A_1 f^{m-4} \quad (\text{inch}^2/\text{Hz})$$

The areas under each segment are obtained by integrating equations 1, 2 and 3 with respect to frequency, **f**, over the interval from $\mathbf{f_1}$ to $\mathbf{f_2}$. This results in the mean-square values:

4) $$g^2 = A_1 \int_{f_1}^{f_2} f^m\, df = A_1 \frac{[(f_2^{m+1}) - (f_1^{m+1})]}{(m+1)} \quad \text{when } m \neq -1$$

$$= A_1[\log_e(f_2) - \log_e(f_1)] = A_1 \log_e\left(\frac{f_2}{f_1}\right) = \frac{A_1}{\log_{10} e} \log_{10}\left(\frac{f_2}{f_1}\right) = 2.303\, A_1 \log_{10}\left(\frac{f_2}{f_1}\right)$$
when $m = -1$

5) $$IPS^2 = 3782\, A_1 \int_{f_1}^{f_2} f^{m-2}\, df = 3782\, A_1 \frac{[(f_2^{m-1}) - (f_1^{m-1})]}{(m-1)} \quad \text{when } m \neq 1$$

$$= 8710 \log_{10}\left(\frac{f_2}{f_1}\right) \quad \text{when } m = 1$$

6) $$inch^2 = 95.80\, A_1 \int_{f_1}^{f_2} f^{m-4}\, df = 95.80\, A_1 \frac{[(f_2^{m-3}) - (f_1^{m-3})]}{(m-3)} \quad \text{when } m \neq 3$$

$$= 220.6 \log_{10}\left(\frac{f_2}{f_1}\right) \quad \text{when } m = 3$$

Adding the mean-square for all segments in the PSD yields the total mean-square values of acceleration, velocity and displacement. Taking the square root of each mean-square provides the RMS value. Multiplying the RMS values for **g^2**, **IPS^2** and **$inch^2$** by 3 provides the peak acceleration, velocity and displacement. Multiply the displacement result by 2 to obtain a peak-to-peak value. Let's consider an example.

Note that some specifications will specify the *slope* of rising or falling segments. The slope is normally presented in either *dB/octave* (decibels per frequency doubling) or *dB/decade* (dB change over a ten-to-one frequency span). This allows m to be evaluated as:

$$m = \frac{1}{10}\frac{dB}{decade} = \frac{1}{10\log_{10}(2)}\frac{dB}{octave} = \frac{1}{3.0103}\frac{dB}{octave}.$$

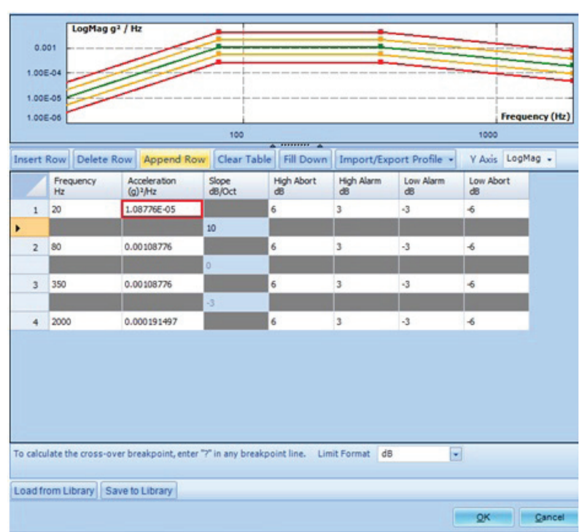


Figure 14: A Random Test Profile is a g2/Hz amplitude Power Spectral Density function.

Figure 14 illustrates a typical Random Test Profile. The frequency, **f**, and **g^2/Hz** of the four points defining this three-segment spectrum were placed in an Excel® spreadsheet. The mean-square values of **g^2**, **IPS^2** and **$inch^2$** for each segment were computed using equations 4, 5 and 6. The results for all segments were summed and the required conversion to peak and peak-to-peak values made. These calculations are summarized in Figure 15.

f (Hz)	**span**	**g^2/Hz**	**m**	**g^2**	**IPS^2**	**$inch^2$**
20		1.09E-05				
	60		3.321928	421.4893	44.59493	0.004776
80		0.001088				
	270		0	0.293695	0.003965	6.7E-08
350		0.001088				
	1650		-0.99658	0.00194	1.66E-06	1.77E-12
2000		0.000191				
sums	**1980**		MS	421.7849	44.59889	0.004776
square roots			RMS	**20.5374**	6.67824	0.069112
3x			Peak	61.61221	**20.03472**	0.207336
2x			Peak-to-Peak			**0.414672**

Figure 15: Calculating the span and peak acceleration, velocity and displacement for the test of Figure 11.

Step 1 – Classical Shock Testing

Controlled electrodynamic shakers have become the preferred test platform for modest shock tests. While drop-test (and other) facilities remain necessary for the simulation of extreme shock pulses, the controlled shaker has proven very cost effective for more routine product qualification and seismic evaluation work. Our modern shaker controllers do an outstanding job of reproducing desired transient pulses safely, reliably and repeatedly. Their use saves the enormous time of designing and iteratively using mechanical drop-targets to provide a required shock profile. One now merely keys in or selects the desired acceleration-versus-time shock profile and runs the test.

However, an electrodynamic shaker presents some physical barriers to shock testing. These machines have a limited range of displacement (stroke) and exhibit velocity limits that cannot be exceeded without loss of control. The shaker controller compensates for these shortcomings by employing a process termed compensation.

Shock test profiles are typically described by an acceleration-versus-time history of a few milliseconds duration. Certain shapes Illustrated in Figure 16 have evolved as the Classical Shock library.

These basic pulse forms stem from prior drop-testing where the test object starts in free-fall and collides with a target whose elastic and crush properties determine the test acceleration profile. The resulting pulse shapes are almost invariably unipolar as displacement and velocity were not considerations in their development. When these same profiles are run on a shaker, velocity and displacement are the primary concern.

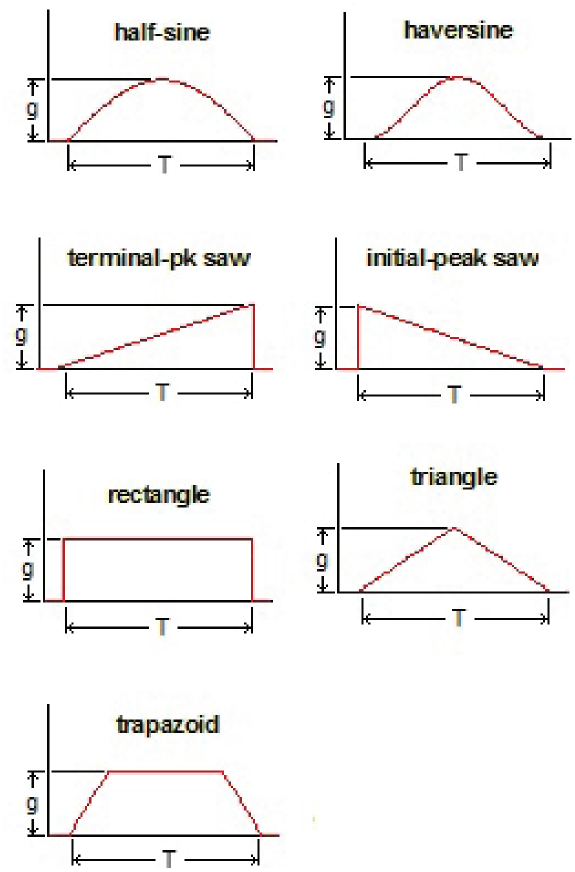


Figure 16: The Classic Shock pulse shapes.

Consider the classic half-sine acceleration shock pulse shown in Figure 17. Since the acceleration is solely in the positive direction, the velocity at the pulse's conclusion is positive and the test object continues to move at this velocity even though the acceleration has returned to zero. The test object has displaced during the shock pulse and will continue to displace at constant velocity until arrested by some barrier. Without smarter intervention, the stroke limits of a shaker could provide this barrier with likely expensive and dangerous results.

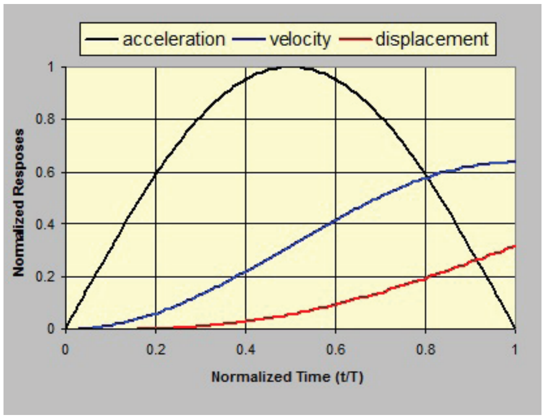


Figure 17: Normalized responses to a half-sine acceleration pulse. Black - acceleration/A, blue - velocity/AT, red – displacement

You will note that the motional responses of Figure 17 were presented in *normalized* form. Time was divided by the pulse duration, **T**, acceleration was divided by the peak acceleration, **A**, velocity was divided by **AT** and displacement by **AT²**. These normalizations are industry-standards, allowing an acceleration pulse and its integrals to be easily over-plotted within a reasonable graphic range, without concern for the peak value of the pulse or its duration or the system of measurement units employed.

A shaker can only operate over a limited range of displacement, its *stroke*. Within this range it can only operate in a controlled fashion if its *velocity limit* is not exceeded. Companion amplifier voltage limits determine the maximum controlled velocity of an electrodynamic shaker. Hence, a shock-test can only be run if the resulting displacement falls within the shaker's stroke range with peak velocity within system limits. Further, as a matter of practicality, the test must start from conditions of zero acceleration, velocity and displacement and return to this state at the conclusion. This is accomplished by adding additional *compensation pulses* of small-amplitude to the desired test profile.

The compensation pulses can be applied before or after the desired test pulse. In general, two pulses of opposite sign are required to cause the displacement, velocity and acceleration to start from and return to zero amplitude. The compensation pulses may be of the same shape as the test pulse or they can be of a very different shape (or shapes).

Compensation pulse height is normally chosen to be a low value within the allowable tolerances of the chosen classical pulse. The two compensation pulses may be of different amplitudes. In general, a lower amplitude compensation pulse must be of longer duration. Figure 18 illustrates compensating a half-sine pulse with two rectangular pulses, each with amplitude equal to 20% of the half-sine's amplitude.

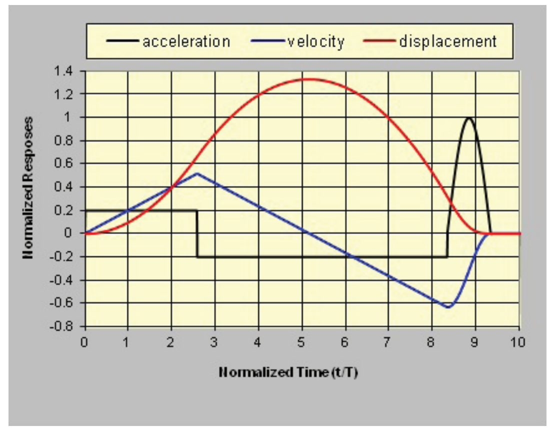


Figure 18: Pre-pulse compensation of half-sine using two 20% rectangular pulses.

Figure 18 presents *pre-pulse compensation* of the half-sine pulse. A test of this type might be used on a disk drive while monitoring read/write operation through and after the shock pulse. Pre-pulse compensation provides a clean after-pulse environment free of any "aftershocks". This allows unambiguous diagnosis of after-shock read/write operational health.

In contrast, consider qualifying an automotive air-bag deployment sensor. One of the fundamental goals of such a test is to measure the acceleration g-level at which the sensor switch closed, causing detonation. Such a test is perfectly served by *post-pulse compensation*. Figure 19 illustrates the same half-sine test pulse compensated by the same 20% amplitude rectangular pulses.

(continued on next page)

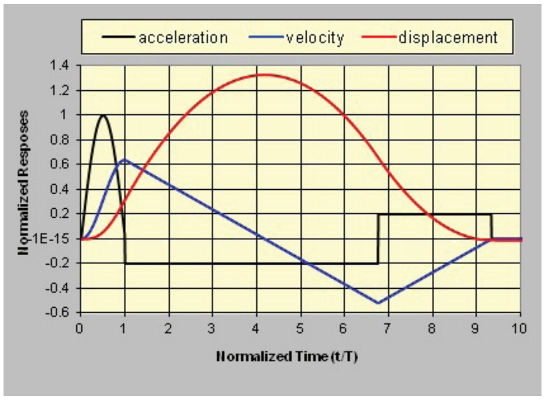


Figure 19: Post-pulse compensation of half-sine using two 20% rectangular pulses.

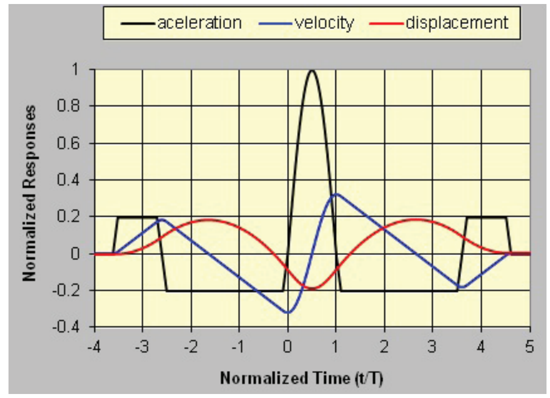


Figure 20: Pre & post-pulse compensation of half-sine using two 20% trapazoidal pulses.

Note that the duration of the compensation pulses is not arbitrary. Once the shape and height (of each) compensation pulse is selected, the duration of the two pulses needed to compensate the test pulse are unique. In Figures 18 and 19, the duration of the +.2 pulse is 2.580 time the duration of the half-sine. The duration of the -.2 pulse is 5.763 times the half-sine width. Hence, the compensated pulse is 9.343 times as long as the prescribed half-sine.

Note that pre-pulse or post-pulse compensation results in a one-sided or unipolar displacement. That is, only half of the shaker's stroke is used. In order to run high-g pulse tests, it is necessary to employ the shaker's entire stroke. This requires pre & post-pulse compensation, involving two compensation pulses of opposite sign before the test pulse and two after it. Figure 20 illustrates combined pre and post-pulse compensation of a half-sine using equal amplitude trapezoidal compensation pulses of 20% amplitude. Compare this with Figures 18 and 19 that illustrate the same test-pulse and note:

1. The peak displacement values are centered about the shaker's mid stroke, doubling the available displacement range.
2. The total peak-to-peak stroke used is significantly less than that of a pre or post-pulse (only) compensated test.
3. The peak velocities are significantly lower for the pre and post-compensated test.
4. The total compensated pulse times are all about equal (≈9.5 times the half-sine duration).

By using two preceding and two following pulses, the controller can position the shaker armature to a negative displacement prior to the test-pulse with a negative velocity equal to about half of the *velocity change* induced by the desired test waveform. It will select these pulses so that the stroke used is centered in the shaker's range. Thus, a modern controller allows the shaker system to deliver the most aggressive test-pulse possible within its physically limited stroke and velocity capabilities. At first blush, this all seems too good to be true. However, it is the natural result of using *shorter* compensation pulses.

Figure 21 provides pre-pulse, post-pulse and pre & post-pulse compensation results for the most common Classical Shock pulses. All compensations tabulated in Figure 21 assume 3% pre-pulse amplitude and 15% post-pulse compensation pulses of half-sine shape. Select a pulse shape, acceleration amplitude, **g**, and pulse duration, **T**. Read the peak velocity as a multiple of **gT**, and the stroke as a multiple of **gT²**.

	Pre-Pulse Compensation				Post-Pulse Compensation				Pre & Post-Pulse Compensation			
	Peak Velocity/gT		PTP Displacemet/gT²		Peak Velocity/gT		PTP Displacemet/gT²		Peak Velocity/gT		PTP Displacemet/gT²	
	m/s	IPS	m	inch	m/s	IPS	m	inch	m/s	IPS	m	inch
Half-Sine	3.055	120.9	14441	568.6	5.4	213.2	23803	937.2	4.2	165.4	8102	320.5
Haversine	2.418	95.46	11208	441.3	4.255	167.8	10831	426.5	3.455	136.29	5374	211.6
Terminal Peak Sawtooth	2.527	99.76	11418	449.6	4.4	173.2	14610	575.2	3.673	144.93	5289	208.3
Initial Peak Sawtooth	2.509	98.95	11838	466.1	4.4	173.2	16499	649.6	3.436	135.6	5709	224.8
Rectangle	4.873	191.9	21327	839.7	6.036	238.2	33040	1301	6.746	265.6	15575	697.5
Triangle	2.436	96.53	11293	444.6	4.309	169.76	15197	598.4	3.509	138.47	5415	213.2
Trapazoid	5.255	207.4	23175	912.4	6.22	245.4	35644	1403	7.291	287.2	20656	813.2

Figure 21: Compensation results for various Classical Shock pulses.

For example, if you desired to run a **5 g_{pk}** half-sine pulse of **11 ms** duration, using **Pre & Post-Pulse compensation** read (in English units) Peak Velocity/**gT** = 165.4 and PTP Displacement/**gT²** = 320.5. Then calculate:

1. Peak Velocity = 165.4 **gT** = 165.4*5*11x10^{-3} = 9.097 IPS
2. PTP Stroke = 320.5 **gT²** mm = 320.5*5*(11x10^{-3})^2 = 0.1939 inch

The solutions presented in Figure 21 are not general. They may be inappropriate for a specific test, but they do provide a convenient means of comparing the utility of specific shaker systems. Modern vibration controllers can combine a myriad of pulse shapes to provide optimization for stroke, velocity and *the energy imposed on the test object*.

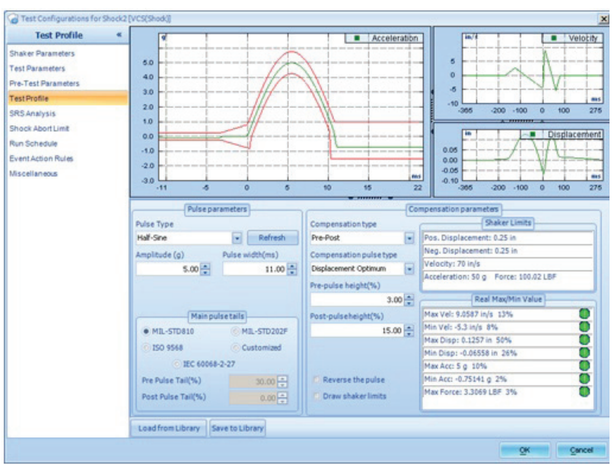


Figure 22: Classical Shock Test Profile is an acceleration-versus-time history.

Figure 22 illustrates a Classical Shock Test Profile as seen on a Crystal Instruments vibration controller. The selected half-sine test pulse (and its allowable tolerance band) are shown in the main display panel with smaller panels presenting the velocity and displacement time histories. Computed extreme values of the required acceleration, velocity, displacement and force are tabulated in the lower right pane.

Figure 22 should be viewed as extremely good news for any reader feeling burdened by the arithmetic discussed in this section. Firstly, we are through the "tough part". Secondly, all Crystal Instruments controllers automatically do these calculations (and a lot more) for you whenever you specify a test profile. In fact, the test requirements can simply be recalled from the test library and melded with the shaker specifications stored in the shaker library. The controller then performs a *feasibility analysis* before attempting to conduct the test and will stop impractical application without risking the shaker system in any way. But to enjoy such convenience, you must first own a controller. Assuming you don't currently have such an instrument, we will continue with the selection process.

Step 2 – Device Specifications

Each candidate test object presents a new series of practical mechanical problems. First, the DUT must be firmly bolted to the shaker. However, you must assure that the center-of-gravity (CG) of the payload is directly over the center of the shaker to avoid stressing and damaging the shaker's suspension. In some cases this results in offsetting the DUT's mounting base from the shaker table's center. In all cases you will need to position tapped holes to receive the hold-down bolts. This often translates into an adaptor plate with tapped holes to mount the DUT and through holes aligning with the tapped holes on the shaker's load table. The adaptor is bolted to the shaker and then the DUT is bolted to the adaptor. Never drill and tap the load table of your shaker!

While a collection of clamps, hold-down bars and threaded rods may serve to mount a specimen for a one-off or experimental shake, anything that is routinely tested deserves its own adaptor. Proper mounting adapters assure the DUT is secured to the shaker in exactly the same way every time the test is run. Permanently mark or label each adaptor and bag and store the necessary mounting fasteners with it. Know the exact weight of each adapter set.

Sometimes the mounting base of a test specimen may too big to sit securely on the shaker with its CG centered. The solution to this problem may be a *head expander* such as that shown in Figure 23. Sentek Dynamics offers a wide range of standard head expanders to fit any of our shaker armatures. Aluminum and magnesium head expanders are available with square or round platforms in sizes from 300 mm (11.8 inch) to 1500 mm (59.1 inch).

Custom head expanders (and custom adaptors and fixtures) can also be provided. Know the weight of each head expander and its table-mounting fasteners. Store these components together.

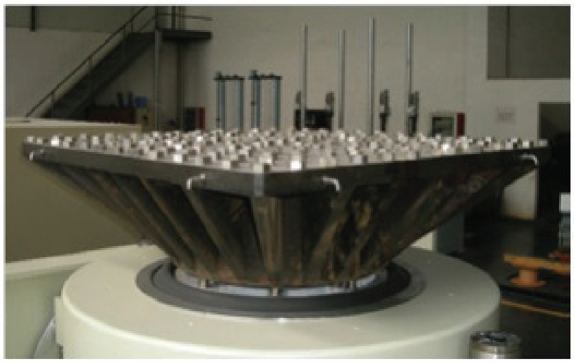


Figure 23: A head expander fitted to the armature's load table increases the mounting area.

A device is often tested in multiple directions (multi-axis testing). If the device is small and light enough, a custom fixture may be built to hold and test multiple DUTs simultaneously. If the specification calls for *the same test profile in each axis*, the fixture can be designed to hold (multiples of) one DUT with its X-axis up, one with its Y-axis up and one with its Z-axis up. Should the specification call for *a different profile in each axis*, a different fixture holding multiple specimens may be required for each axis. Again, permanently mark each adaptor and store it with its required mounting fasteners – know the mass of each such fixture set.

In most cases, a small DUT can be rotated to align a desired test axis with the vertical. However, this may not always be true. If the DUT must be subjected to multi-axis shaking, but it cannot be laid on its side, you require a *slip table*. As shown in Figure 24, this is simply a horizontal table floated on an oil film excited by a shaker rotated to the horizontal and fitted with a *driver bar*. This rig is used to perform the X and Y-axis horizontal shakes. The Z-Axis is most probably done by remounting the DUT on the shaker after it is re-oriented to the vertical.

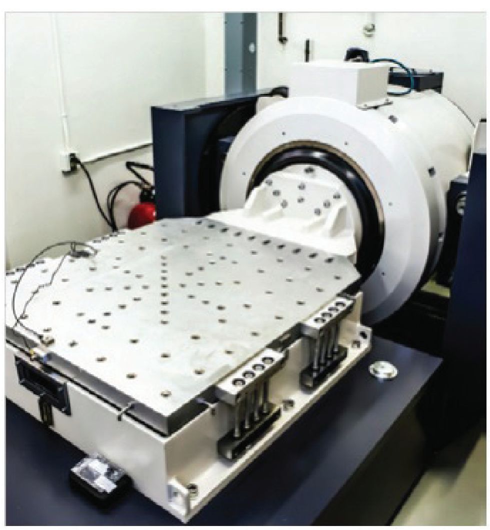


Figure 24: A slip table driven by a shaker through a driver bar is used for horizontal shake tests.

Know the moving-mass of your slip table and the mass of your driver bar and all its fasteners.

In short, your analysis of the device under test (DUT) should reveal the mass that must be driven by the shaker in every test configuration required. This includes any adaptors, head expanders, slip tables, driver bars and fasteners required. For every axis (or other variation that must be driven), know the *total mass* that will be driven by the shaker.

Step 3 – Shaker Specifications

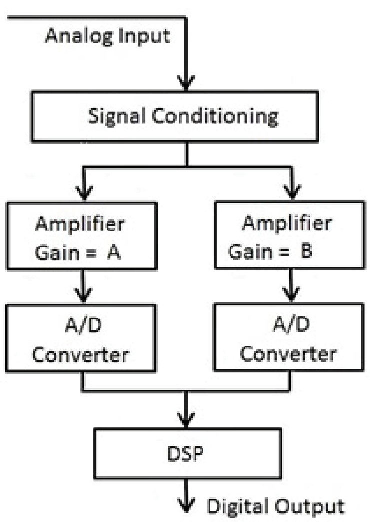
Verify that the *total moving mass* (for a vertical shake) identified in step 2 is less than the *maximum static payload* of a trial shaker. Add the shaker's *effective armature mass* to the total moving mass determined in step 2. Multiply this total by the acceleration determined in step 1. The resulting peak force (**F=ma**) must be less than the *Peak Sine Force* rating of the shaker for a sine test and less than the *Peak Shock force* rating for a classical shock test. For a random test multiply the total moving mass by the computed *RMS acceleration* and verify the result is less than the shaker's RMS *Random Force* rating. Verify that the shaker's *Frequency Range* exceeds that determined for the test in step 1. Verify that the peak velocity required for the test is less than the *Peak Velocity* of the shaker. Be absolutely certain that the peak-to-peak displacement determined necessary for the test is less than the <u>de-rated</u> *Maximum Displacement* (Stroke) of the shaker. If all of these requirements are met, the shaker is suitable to run the test in question with the specified DUT and fixture payload.

If multiple benchmark tests must be met, some shaker iteration will likely be required. It will be necessary to select a shaker with the highest force rating and longest stroke required by all tests investigated. With an eye to your future, respect the old saw, "You can never own too much stroke!" Be aware that stroke and force ratings affect system price directly. Also be aware that owning insufficient shaker can cause you no end of grief.

Selecting Your Controller

Crystal Instruments (CI) offers more choices of vibration controllers than any other manufacturer. You can choose a very dedicated controller or a broad-application modular hardware platform. CI offers controllers with 2 to 512 input channels and 1 to 128 output channels. Every CI controller uses the same powerful *Engineering Data Management* (EDM) software providing exactly the same user-friendly control interface regardless of the selected hardware.

Each analog input is serviced by <u>two</u> 24-bit ADCs and a DSP implementing the cross-path calibration technology of our US Patent number 7,302,354 B2 to achieve better than 150 dB dynamic range, simultaneously measuring signals as small as 600 nV and as large as ±20 V, with time data retained in 32-bit single precision format (per IEEE 754-2008). 54 sample rates from 0.48 Hz to 102.4 kHz are provided with better than 150 dB of alias-free data from DC to 45% of any selected sample rate protected by steep 160 dB/Octave anti-aliasing filters. Successful control is all about dynamic range and no competitor can exceed ours.



All Crystal Instruments controllers are designed to function as networked servers. An Ethernet connection is used between the PC or laptop and the controller because it uses only *differential signals* and cannot cause a ground loop. In addition, Crystal Instruments employs Ethernet with IEEE 1588 Precision Time Protocol (PTP) time synchronization technology. Spider controllers and modules on the same network can be synchronized within 50 ns accuracy, even when hundreds of meters apart. This guarantees ±1° cross-channel phase match up to 20 kHz across the complete system. With this unique technology and high-speed Ethernet data transfer, the distributed components on the network truly act as one integrated system.

With the wide variety of Crystal Instruments controllers, you are sure to find one that is exactly right for you. Our most popular model is the **Spider-81**, a full-featured dedicated controller with eight inputs and four outputs. When more channels are a "must" consider the 16 input **Spider-81A**. If you are testing complex prototypes and need the utmost in flexibility, consider a **Spider-80X-A35** dedicated enclosure and multiple **Spider-80X** modules. For a minimum-cost system ideal for academic applications look to the 4 channel **Spider-81B**. If your work centers on transportation *Squeak and Rattle* resolution, consider the **Spider-81C**, which allows you to control the shaker with your iPad® as you walk around and interact with the test specimen.

Spider-80X Chassis and Module

Spider-81A

Spider-81

Spider-81B

Spider-81C

STANDARD CRYSTAL INSTRUMENTS CONTROLLER MODELS AT A GLANCE

Model	Analog Inputs	Charge Mode	Analog Outputs	Digital I/O pairs	LCD Panel & Controls	Special Features
SPIDER-81	8	yes	4	4	yes	expandable to 512 inputs using SPIDER-HUB
SPIDER -81A	16	yes	4	4	yes	built-in SPIDER-HUB with 7 IEEE 1588 Ethernet inputs
SPIDER -81B	4		2	4		high-quality low-cost basic controller
SPIDER -81C	2		1	4		built in wireless modem (no Ethernet)
SPIDER -H	4		1	dedicated		controls Halt/Hass chamber temperature & humidity
SPIDER-80X	8		2	4		Dual-function DSA and VC S module
SPIDER-80X-A35	8N		2N	4N		dedicated rack with N SPIDER-80X modules ($1 \leq N \leq 8$)

Crystal Instruments offers a full suite of control software. Choose exactly what you need and know that you can expand your test repertoire as required. Please visit http://www.crystalinstruments.com/ for far more detailed information about our Vibration Control System (VCS) hardware, software and application information.

VCS-20	**Random Control**
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VCS-20-03	Data Recorder Function
VCS-20-04	High Resolution
VCS-20-05	Drive Notching/Limiting
VCS-20-06	Kurtosis Control
VCS-20-08	Sine on Random
VCS-20-09	Random on Random
VCS-20-10	Dual Shaker Control for Random
VCS-20-11	Displacement Optimization for Random
VCS-20-15	iPad Application for Random
VCS-40	**Swept Sine Control**
VCS-40-01	Resonance Search, Track and Dwell (RSTD)
VCS-40-03	High Frequency Control
VCS-40-05	Total Harmonic Distortion (THD) Measurement
VCS-40-06	Data Recorder Function
VCS-40-07	Drive Notching/Limiting
VCS-40-09	Low-frequency Control
VCS-40-10	Dual-Shaker Control for Sine
VCS-40-15	iPad Application for Sine

VCS-60	**Classical Shock Control**
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VCS-60-03	SRS Synthesis
VCS-60-04	Data Recorder Function for Shock and TTH
VCS-60-06	Large Block Size for Shock and TTH
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VCS-80-04	Data Recorder Function
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