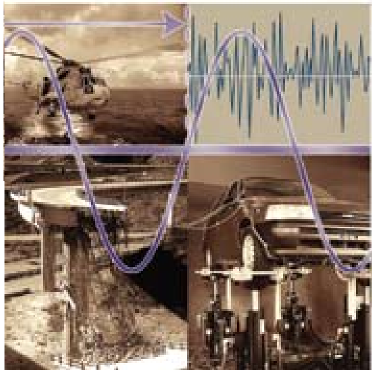

ERI News

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In this issue...

Selecting Isolated Enclosures for COTS Electronics (part 2):

Shock Mounts, Barge Test, Input Levels and Shock Response of Isolated Enclosures

by Herb Lekuch

Introduction

Part 1 of this article was given in the November 2009 issue of ERI News and covered military shock conditions, isolated enclosures and COTS equipment requirements. Part 2 is presented in this issue and describes enclosure layout, high deflection shock mounts, the US Navy's Mil-S-901D barge test specification, typical shock levels and the shock response of isolated enclosures.

Enclosure Layout

Figure 1 shows a typical arrangement of an externally isolated enclosure. Mounts are near the corners of the base and upper rear corners of the rack. Space for the mounts is usually limited and adaptors sometimes have to be provided, for example many racks require special brackets at the rear stabilizers in order to attach the mounts to wall or overhead structure. Isolators are positioned for the widest footprint within the available space.

The simplest, well-balanced arrangement provides the most efficient isolation for shock and vibration reduction and helps to avoid the complexity of pitch and sway particularly of tall racks. In each layout the objective is to bring the elastic center of the isolation system near to the center of gravity (CG) of the populated rack in order to minimize the rotational movement of the rack due to offset loads and oblique shock. To meet these criteria, some mounts may have different stiffness than others depending on the direction of shock and the amount of load that is supported by the isolators.

Working Space for isolation

Military and off-road vehicles are usually difficult packaging conditions with only minimal space for isolators, mounts are often selected to be "stiff" having less than 1 inch of deflection stroke. Navy shipboard installations can normally allow more space and typically 4 inches of deflection stroke is acceptable. Racks with equipment qualified to the Mil-S-901D medium weight shock test (MWSM) commonly used isolators having only 1.5 to 2 inches of deflection stroke capability and many of these have bottomed in the barge test. The greater shock intensity of the barge test showed that some units previously qualified with isolation mounts on land based machines were unsatisfactory when tested on the barge due to the greater relative movement between deck and underside of the rack which exceeded the free space of the isolator.

Voice of the President

810G instead of 810F

Is it possible that your test engineers, design engineers and others concerned about the reliability of your company's products are still mentally back on the venerable (now 11 years old) *Military Standard 810F*? *MIL-STD-810G* appeared in October 2008. I hope that your designers and test lab people have gone to DTC's website and downloaded some or all 800+ pages of "G". And that they have updated your company's design and test procedures. I urge you to read in 810G about some interesting new Test Methods.

My comment was raised by seeing some current ads bragging about how a firm's products meet *Military Standard 810F*. You might want to inform your sales/advertising department that your military customers know about "G" and are probably chuckling every time you brag about "F".

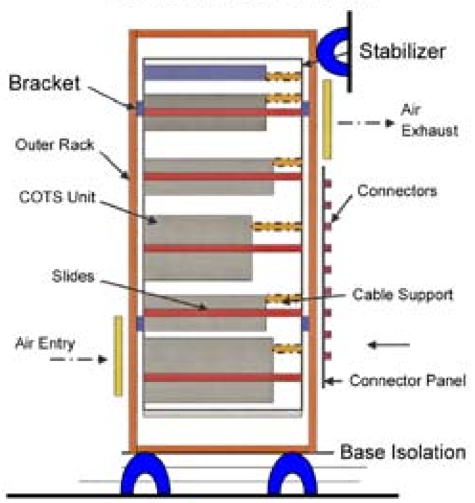


Figure 1 - Schematic: Isolated Rack

Grouped as an assembly of electronic units, the populated enclosure of the kind shown in Figure 1 is a framed structure with the COTS equipment at different locations. The units are secured at the front columns and supported by rear brackets or shock pins. This is an externally isolated rack. Slides enable each unit to be extracted from the front of the rack. Structural elements, chassis, slides, etc., have spring-like properties and minimal damping. To a dynamics engineer, the assembly is a collection of stacked sprung masses, some of which will interact with each other. Shock is transmitted into the enclosure at the base and stabilizer isolators. Measurements show differences at these locations. However in most analyses the input is usually considered to be the same at all mounts.

Problems due to shock have involved cable obstruction and extreme electrical motion at upper outer corners of the rack. Sway motion can exceed 3 inches with soft isolation systems. Avoid snubbing and stiff isolators - they can result in local impact at COTS due to inadequate vertical separation between units. There are also examples in severe shock of inserts having pulled out in the side panel of equipment chassis where the slides are attached.

Shock Mounts

The effective shock isolator must be capable of absorbing the impulse energy, dissipating some percentage of energy as strain and heat, while returning the remainder slowly in terms of rebound velocity (kinetic energy) of the movable mass.

Two main types of large deflection mounts commonly used in the United States for Navy equipment are Elastomer Arch (Figure 2) and Helical Cable (Figure 3 - preformed wire rope). Shocktech Inc is a leading manufacturer of both types of mounts. Other manufacturers include IDC, Enidine, Newport News and Barry. In compression, these mounts undergo a form of buckling having high initial stiffness followed by progressive softening over the latter two thirds of allowable stroke. Elastomer mounts also undergo elastic deformation as the rubber stretches. Cable mounts become more oval as they compress and the effective width to height increases. Both types of mounts exhibit non-linear stiffness rates which can yield different frequency response for a system; for instance a mount showing a 6-7 Hz response in shock can exhibit 8-10 Hz in vibration.

To meet the severity of barge shock conditions while also supporting the enclosure and still control the response of the rack to the 15 g's or less, deflection requirements (as shown later on) must allow for up to 4 to 5 inches deflection while having a response frequency of 5 to 7 Hz. If the isolator natural frequency is less than

Figure 2 - Elastomer Arch Mounts - base isolation Courtesy Shocktech Inc. and 901D LLC.

On a different note I'd like to thank our readers who collaborated with Resonance stories for our just released Resonant Ruth column**.** We will continue to publish these stories on our upcoming issues. We would love to keep on receiving them!

Can We Make a Date for a Skype Call?

Nominally, you can call wayne.tustin between 8am and say 8pm, occasionally earlier and occasionally later. Scheduling calls is a good idea --- either a Skype text message or an e-mail.

The times I just mentioned, 8am to 8pm are Pacific time, the same as say Los Angeles. Your time may be different. The World Clock website is useful in calculating a time when we are both reasonably alert.

5 Hz, these mounts can become unstable due to their relatively long column length to width ratio. The amount of damping is an important factor in limiting the shock response and 15 – 17 % C/Cc is a reasonable value for these types of mounts.

Figure 3 - Cable Isolators - base isolation Courtesy Shocktech Inc. and 901D LLC.

Arch elastomers by Shocktech are 7 inches tall, 40 to 70 durometer depending on the model, rated at 13-17 % C/Cc. Capable of 4.8 inches deflection in compression, tension is linear to approximately 150 % elongation. Roll and shear stiffness rates are also nearly linear. Values are approximately 30 to 40 % of compression stiffness depending on the deflection range.

Cable mounts, also 7 inches high, are rated for 4.4 inch compression travel less allowance for incremental set and static deflection. Roll and shear stiffness rates are 30 to 35 % of compression stiffness. Tension is limited by rapid stiffening at almost 2 inches extension. Pre-setting the isolator can be used to adjust the installed height. Allowable loads for cable isolators are generally greater than for elastomer mounts. Stiffness and load rating is a function of the number of cable loops, cable diameter, loop oval and type of preformed stainless steel cable. Damping is in the range 13-17 % C/Cc and partially depends on the amplitude of relative motion across the mount.

Shock Input *vs* Response

The properties of the isolator represent a bridge that is both a barrier and a filter of the input shock applied to one side of the isolation. The output from the isolator is of course the input to the enclosure. The primary objective is to reduce the peak g's to levels thought to be within the capabilities of the equipment. Allowable levels are generally based on manufacturer's specifications although in the absence of data, experience suggests that less than 15 g's is an acceptable level for most COTS electronics.

Navy 901D Barge Test

Photos of Test Setup and Barge Shot

Figure 4 shows a group of racks readied on the barge deck and instrumented with signal and accelerometers cabling. Data is collected and processed off-site from the barge. Fixturing is used to attach the rack as it would be on-board ship. Figure 5 shows the plume almost immediately following the explosion, the shock having already been felt at the barge, the barge is beginning to displace vertically and to the side. The Grade A designation of MIL-S-901D requires

Figures 4 and 5

2010 "Open" Courses

Fundamentals of Random Vibration and Shock Testing, Measurement, Analysis, Calibration, HALT, ESS and HASS taught by Wayne Tustin or Steve Brenner at the following locations:

February 23-25, 2010
Santa Barbara, California

April 5-7, 2010
College Park, Maryland

April 20-22, 2010,
Chatsworth, California

May 10-12, 2010,
Santa Clara, California

Steve Brenner will teach **Military Standard 810G (MIL-STD-810G) Testing - Understanding, Planning and Performing Climatic and Dynamic Tests** on the following dates and locations:

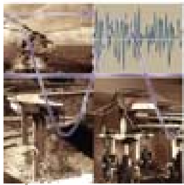
March 15-18, 2010
Montreal, Canada

April 12-15, 2010,
Plano, Texas

May 17-20, 2010,
Cincinnati, Ohio

Larry J. Bogatz will teach **The Fundamentals of Sealing and Fastening**

February 16-18, 2010
Santa Barbara, California



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that the equipment function satisfactorily, or if interrupted, come back to operational mode within a prescribed time limit. Grade B designation indicates that the equipment failed to operate correctly but has remained intact and not impaired personnel safety such as might occur if the door latches failed.

Shock Response
An example of shock response of an isolated rack versus the shock input on the 14 Hz deck in the Navy barge test is shown in Figure 6. Cutoff frequency of the processed data was 250 Hz.

The peak input slightly exceeds 45 g's, the response was held to under 13 g's. The bubble pulse at approximately 0.6 second caused a second significant peak of approximately 15 g's but longer duration than the initial pulse. The response was again nearly 13 g's. The isolation for this unit was nominally 6 Hz. The effects of isolation can also be clearly seen in the SRS plots, Figure 7, where the velocity and acceleration of the response are substantially reduced from the input beginning at 8-9 Hz. when compared to the acceleration and velocity levels for the input SRS. In the isolation frequency vicinity

Test Data - Acceleration vs Time and Shock Response Spectra (SRS)

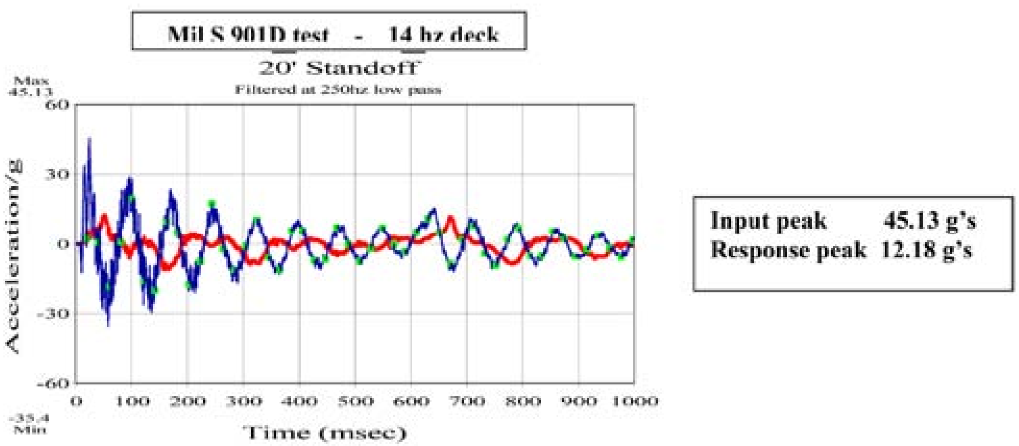

Figure 6

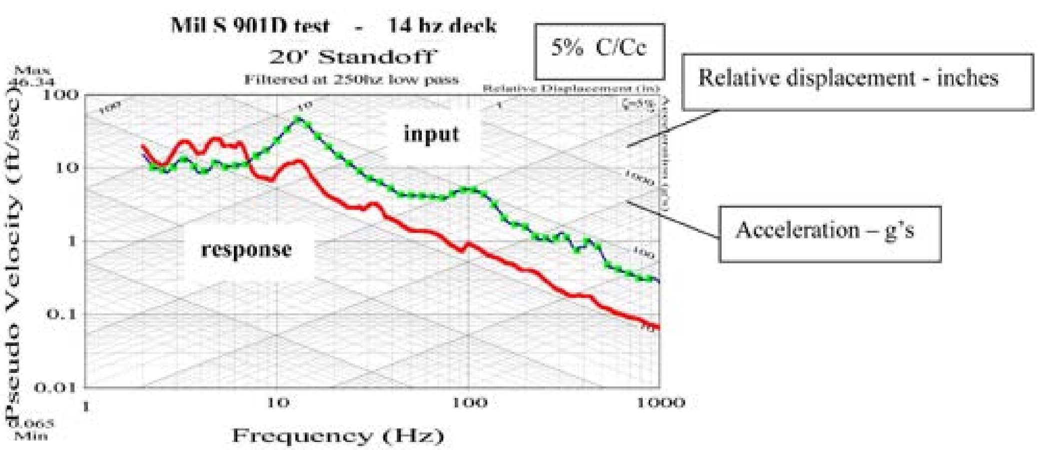

Figure 7

New courses in 2010!

Kevin Howard will teach a new course in **Distribution, Packaging and Testing**

March 2-4, 2010
Santa Barbara, California

Vladimir Valentekovich will teach **Pyrotechnic Shock Testing, Measurement, Analysis and Calibration**

April 13-15, 2010
Santa Clarita, California

Herb Lekuch will teach **Isolating COTS Equipment aboard Military Vehicles**

May 4-6, 2010
Suffern, New York

☞ *Take advantage of our early-bird discounts and save up to $200 when your enrollment and payment reach ERI one month before the course starts. Sign up now!*

Vibration with Minimal Math

Yes, Wayne Tustin's vibration text (picture below) weighs 4+ pounds, but the math is less than 1 ounce. In a 3-day course, Wayne or Steve Brenner or Herb LeKuch spends perhaps 15 minutes using any branch of mathematics.

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just below and through 8 Hz, the response is greater than the input indicating amplification in a range of 1.5 to 2.0 as would be expected for a damped system.

Critical Frequencies and Possible Damage
SRS plots on four coordinate paper shows the frequencies having the highest velocities. The critical frequency region of most electronic equipment is 25-125 Hz. As shown in Figure 7 - with isolation the velocities in the same region are lower because the response shifts to the lower frequencies and decreases the accelerations at the higher frequencies beyond isolation cutoff. More damping produces additional reduction of peak shock.

Damage is understood to occur when stress within a part exceeds its strength. This can happen in a single direction or a combination of loads. Analysis of shock induced stress is mainly based on several factors. These are [1] equipment and components vibrate in mode shapes at their modal natural frequencies, [2] the peak modal stress is proportional to velocity and [3] the unit is considered to accept energy only at its modal frequencies. Calculated velocity is taken as an indicator of the amount of energy at a particular mode. By this model, reducing the velocity by shifting peak g's to lower frequency is thought to reduce the risk of damage because the part undergoes fewer severe velocity cycles and those at critical frequencies are below the amplitude identified with damage.

Acceleration and Relative Displacement vs Frequency
Figure 8 is a plot of acceleration and relative displacement for a multi-mass linear spring model based on typical complex input. For calculation purposes the initial 60 g pulse was followed by damped successively decreasing peaks.

In the isolation frequency range of 5-7 Hz, shock response is 15-20 g's and relative displacement is 3.5 to 3.8 inches. As the stiffness of the isolation system increases, the response also increases (exceeding 20 g's) reaching 40-50 g's at 10-11 Hz. Response increases again (exceeding the 60 g input) over the range of 12-18 Hz Relative deflection is at least 3.0 inches

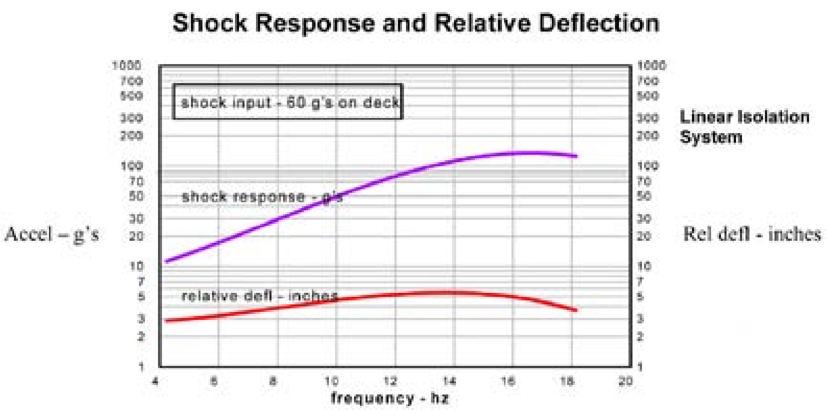
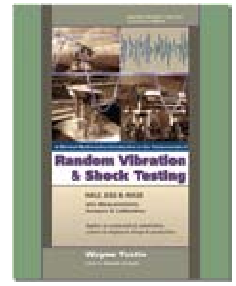


Figure 8

and can bottom the isolator when the isolation is near the predominant frequency of the impulse. Actual results are less because of the greater energy capacity of the softening mounts compared to the linear mounts used in these calculations. Calculated results (and measurements) show the benefits of isolation.

Non-linear Stiffness Characteristics of the Mounts
Arch and cable mounts have similar non-linear stiffness curves in compression. However rated loads and actual transition from relatively 'stiff' to 'softening' are different due to the way in which the isolator deforms. The elastomer arch undergoes shape distortion and elastic stretching while the cable mount changes its oval contour; there is essentially no elasticity in the wire

Their emphasis is on the practical aspects of vibration and shock measurement and testing, along with ESS, HALT and HASS.



OJT is expensive!

OJT or on-the-job training can be very expensive. See figure below. What happened? Well, too much current flowed through that shaker driver coil for too many seconds.

What caused too much current to flow through that shaker driver coil for too many seconds?

The people who repair shakers, in this case winding and installing a new driver coil, tell me that in nearly all cases the shaker user does not know what he did wrong.

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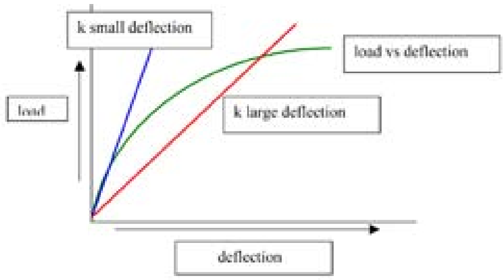

Figure 9

rope. The effective shock response stiffness is averaged at approximately 75 % of the allowable stroke. However, as shown in Figure 9, response at long stroke can be different from response at smaller deflection.

Isolation and Chassis Dynamics

Why increasing the rattle space for isolation may be more important than redesigning chassis construction - the characteristics of the isolation system including the dynamics of the chassis governs the response to a shock input. Vibration response is described by a similar combined isolation and chassis transfer function for that condition.

The transfer function H_f is dependent on the ratio of disturbing frequency to resonant frequency of the system and will be different for shock versus vibration. For example in vibration the general form is $A_{f\ (input)} * H_f = A_{f\ (response)}$ where A_f is the acceleration at a particular frequency.

Figure 10 shows an isolated unit (with rigid chassis and PC board) having nearly 0.1 transmissibility T_r at 25 Hz. Isolation resonance is at 10 Hz, moderately damped at 0.1 C/cc. However if the chassis resonates at 25 Hz, lightly damped and the PC board at 35 Hz also lightly damped, the combined effect is a second mode with a T_r of 2.0 and a third mode at 35 Hz and about 1.0 T_r meaning that the acceleration is at least 1.0 or greater beginning at 5 Hz. The black line curve shows the combined modes.

Summary

Shock reduction to the 15 g criteria for COTS equipment requires relatively soft isolation systems operating in the 5-7 Hz range. This is based on input on the 14 Hz deck. Deck shock conditions will vary depending on the

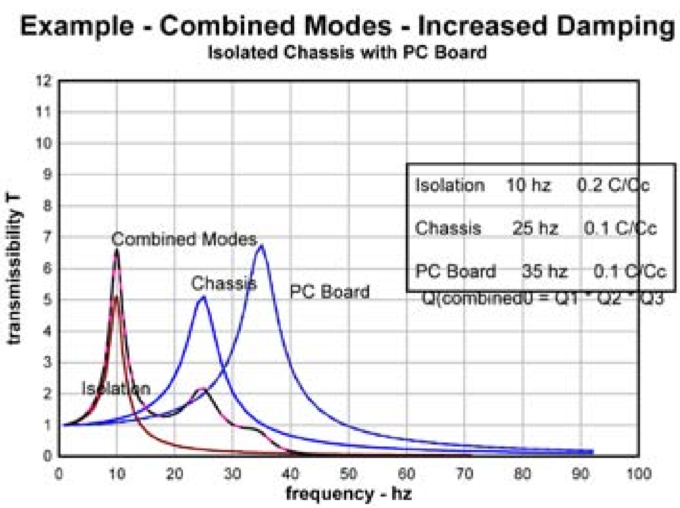

Figure 10

deck frequency and a conservative selection of mounts is recommended. Softening isolators and isolated enclosures have been successfully used. The enclosure requires 3.5 to 3.8 inches of relative deflection to satisfactorily reduce the shock to acceptable COTS levels. Separation of chassis dynamics from the isolation system is important. The benefits of the isolated enclosure isolation include lower velocities at the critical modes and frequencies of the equipment. The risk of damage to equipment is reduced as a result.

*Herb Lekuch is an ERI specialist, with an extensive experience in mechanical design, shock and vibration isolation, test and analysis. Herb is teaching **Isolating COTS Equipment aboard Military Vehicles** course* May 4-6, 2010, Suffern, New York.

Typically, the repair costs $10,000. Typically, it shuts down the shaker for a month, forcing the lab to test elsewhere.

I can cite an instance where a lab did this three times and still didn't know the cause of this Upon e-mail request, I'll identify the lab. Upon e-mail request, I'll tell you the most common error leading to burnt-out driver coils.

One partial solution: purchase a spare armature assembly.

A better solution: take one of our short courses or participate in our distance learning. Start at our website and pull down **Training**. Several can attend for $10,000. Learning to avoid this specific shaker failure is one benefit. There are quite hundred more benefits.

Consider joining LinkedIn

Sign up at LinkedIn to exchange ideas, opportunities and information with many other professionals. Search for ERI's Vibration and Shock Group. You can comment on multiaxis vibration testing, on hammers for repetitive shock (as in ESS, HALT & HASS) and other subjects. You can ask questions and expect to get answers.

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Shock Fragility

by Kevin Howard

A test lab in Europe posed the following question: I have a problem with shock fragility determined by the product engineers of one of my clients. They use static calculations without taking into account a minimal pulse time or velocity change. I have the feeling that the amount, or thickness of cushioning being used for the product is far more than is necessary, causing higher material costs, while I am not allowed to perform actual damage boundary tests, as the product is too expensive for destructive testing.

Kevin answers: Your short question actually touches upon several issues. Also, in a subsequent note you mentioned that your client uses only the critical acceleration side of the damage boundary, but only to verify that the product does not break with some minimum input, as opposed to identifying the most fragile components within his product.

I'd like to first say that a shock table is a great tool for identifying the weakest links of a product design. Conducting a full damage boundary test (as described by ASTM D-3332) can be very useful in understanding the most likely parts of a product to fail first in distribution. Using proper methodology, along with appropriate fixture design, allows one to compare a new product's robustness with a previous product's fragility level. Replicating consistent failures found in the field on a shock table allows one to establish a base line for product modifications, or for new products to surpass. This assumes that the distribution system and hazards remain similar to those that caused the initial issues.

Please note that I'm focused here on using the shock table for product design more than for packaging design. The shocks generated on the table, (half sine, 2-3 ms, fairly high acceleration to identify critical velocity change and trapezoidal, long duration, relatively low acceleration to identify critical acceleration) are not inputs that would normally occur to a real product in the real world. It should be noted that when a product is rigidly attached to a shock table, it's like adding a suit of armor to the outside structure of the product, reducing its flexibility compared to what might occur in a cushioned package during impact. As a result, outer plastic case parts may not fracture from shock impacts on a shock table, no matter the level of intensity, while they may well fracture from a moderate impact in package drop. Also, in reading ASTM 3332, one sees that the velocity change impact can only effectively test spring-mass systems with natural frequencies of up to 83 Hz (if the shock duration is 2 ms, but only up 56 Hz if the duration is 3 ms). Even with the above limitations, the basic advantages of the shock table include: repeatable orientations of impact, repeatable inputs, the ability to measure both input and response, and ability to compare one design with another simultaneously if desired. At the same time, there are several reasons to also use free fall impacts, both for bare products and packaged products.

Your client, like many others, uses only the critical acceleration side of the damage boundary, though he doesn't test to failure. You say they don't break products because the products are too expensive. This high value also stops them from testing very many units. Establishing minimum robustness standards for bare products is a great idea. Most products may well experience some level of impact once the packaging is removed, so it behooves the manufacturer to assure himself that the product can withstand a certain amount of abuse. Unfortunately, using only the long duration trapezoidal pulse may not break the same components that a short duration velocity change pulse causes. To

ESTECH 2010 and ITEA Test Instrumentation Workshop

The upcoming IEST's, Institute of Environmental Sciences, ESTECH 2010 event will meet next May 3-6, at Reno, Nevada. Wayne will talk about "Fixturing - The forgotten keystone of a successful vibration/shock test". He will commence by considering the armature of a typical electrodynamic shaker, and will point out that it often has undesirable resonant motions. Why? Because often the armature resonates, even though its development may have spanned years. The test fixture designer may have only hours. So it is no wonder that fixtures often resonate. Wayne will also discuss problems with attachment bolts. He will recommend one simultaneous multiaxis test rather than the current sequential axis testing; one advantage: one fixture instead of three.

Right after ESTECH 2010, Wayne will also participate in the 'Transducer Workshop session of the ITEA Test Instrumentation Workshop, May 10-13, at the Tuscany Suites and Casino in Las Vegas, Nevada. Wayne's tentative title is "Vibration and Shock are Multiaxis". He will start with triax accelerometers and cables that have three connectors.

rectify this, I recommend that companies establish minimums for both velocity change and acceleration. However, due to the limitations mentioned above for the 2-3 ms pulse, I also recommend small free fall impacts onto a very stiff surface, like the shock table surface. Dropping products onto their bottom faces and edges can represent the impacts the products will likely experience in the "real world."

Believe it or not, your client can benefit financially from conducting tests to failure! What if their minimum quality standard called for a 30 G level robustness, but the product didn't actually break until 60 G's? Would this not mean they could use half the amount of protective packaging materials they currently use? Or .... what if the product breaks at 31 G's…wouldn't your client like to know the most likely component to break, before customers begin complaining? Wouldn't your client want to have more design margin than 1 G? Manufacturing and assembly tolerances might well allow such a component to degrade below their 30 G minimum.

Companies that use only the acceleration side of the damage boundary often believe they should then provide cushioning that will not allow any more than the defined critical acceleration level to pass through the cushioning. This is extremely conservative. A trapezoidal input causes all components to react to their maximum levels. Any other type of input (for instance a half sine pulse during package drop) will only affect some components; none will respond as much as they do on the shock table. The amount of response to a shock input is regulated by the ratio of the component's natural frequency to the shock input frequency. As a result, the longer the duration of the input pulse, the greater the shock amplification, even though the input shock may be simultaneously mitigated/ reduced with the cushion. As a result, if a product breaks on the shock table at 50 G's, then it will probably not break in a cushioned package until the half sine pulse reaches 85 to 100 G's. Also, a shock table normally tests only flat impacts, whereas a product in a package is far more likely to experience edge and corner impacts rather than flat impacts. Box edges deflect, further buffering the product as long as it doesn't bottom out, causing a direct hit on the corner of the product. Misunderstanding the conservative nature of a shock table has led many companies to believe that they need resilient, thick foam cushioning when actually they only needed inexpensive expanded polystyrene (EPS) or molded pulp and less deflection space.

More progressive companies realize that focusing on the velocity change side of the boundary allows them to reduce protective packaging. Consumer products with critical velocity changes greater than 120 in/sec (the higher the better), along with critical acceleration levels greater than 50 G's, often need only good blocking and bracing to protect plastic case parts from impact. If a product's critical velocity change exceeds the input velocity changes found in handling and distribution, there is no need for shock mitigation. Also, more cost-conscious companies see the benefits of testing, and breaking, more products rather than fewer products. Yes, prototype products can be expensive, but suffering field damages as a result of incomplete testing, or producing excessive packaging for the life of the product and obtaining no real value for it, both reduce profits. It is often possible to minimize the costs of breaking many products by simply understanding early in the program the most likely components to break and then re-building products with modifications to those specific components and/or their mounting features. Some companies have been very progressive in also utilizing dummy products instead of real products to help fine tune packaging designs prior to the first products being tested. This method allows the most cost effective package to be developed first, and then to require the product designers to provide sufficient

Then he will ask why do we still use ordinary single-axis electrodynamic shakers. Note that the automotive folks multiaxis shake, using multiple servohydraulic shakers. Wayne will review what the Army has done with multiaxis at Adelphi and at White Sands. Also the Air Force at Ogden, Utah, Hill AFB. Also the Navy at Keyport, Washington. One commercial test lab at Redford, Michigan has imported a 3-axis system from IMV in Japan. Wayne's presentation will include video clips from the labs.

New Shock and Vibration Handbook

Many of our readers are familiar with the McGraw-Hill Harris' Shock and Vibration Handbook, which appeared in 1961 (Wayne has an autographed first edition), with subsequent editions in 1976, 1988, 1996 and 2002. Edition 6 has just been launched, including innovative techniques and technologies, such as the use of waveform replication, wavelets, and temporal moments. In late 2007 everyone's dear friend, the late Allan G. Piersol, having devoted years of effort to Edition 6, asked soon-to-retire-from-Sandia Dr. Thomas L. Paez to become co-editor.

robustness to withstand typical hazards that the packaging can not protect from.

Reducing the direct material costs of packaging is good, but reducing box size and getting more boxes on each pallet and in each vehicle ... there are the real savings! Logistics costs for moving packaged products can be 5 to 10 times greater than packaging direct material costs. Many companies benefit from turning their design process around. Instead of testing products to see how much packaging should be added, they should establish the smallest package possible. Maximize densities throughout distribution. Minimal protective packaging forces responsibility onto product designers. Profits increase. This topic is further covered in all my Design for Distribution courses.

Kevin Howard - Kevin, an ERI Specialist, is a well known leading expert in the field of packaging engineering. Kevin specialized in distribution packaging and testing in school and has applied this knowledge over the past 25 years to generate some of the largest cost savings in packaging history.

Kevin will be teaching a new course in "Distribution Packaging and Testing - Design for Distribution - Proven Methods for Reducing Costs and Damages" on March 2-4, 2010, Santa Barbara, California

Resonance – Useful in musical instruments (but potentially harmful in vehicles)

From Wikipedia, the free encyclopedia
Page Version ID: 299493431

A wind instrument is a musical instrument that contains some type of resonator (usually a tube), in which a column of air is set into vibration by the player blowing into (or over) a mouthpiece set at the end of the resonator. The pitch of the vibration is determined by the length of the tube and by manual modifications of the effective length of the vibrating column of air. In the case of some wind instruments, sound is produced by blowing through a reed; others require buzzing into a metal mouthpiece.

How does resonance affect that explanation? Relatively weak sounds at many frequencies are produced simultaneously at the mouthpiece. If the player places his fingers properly to produce, for example, a note called “middle C”, those relatively weak sounds at 256 and its harmonics at 512, 768, 1024 Hz, etc. are greatly magnified by resonance. Resonance? Yes. The length of the resonant air column is the distance from the mouthpiece to the furthest away closed key. The player has closed all the closer keys.

I hope we agree that with musical instruments, resonance is useful.

Seeking New Specialists

In these difficult times, some who have been laid off have decided to consult for many companies (rather than seek another job). If that sentence applies to you, please consider associating with ERI.

As you know, ERI is a specialized engineering school, focusing on enhancing the reliability of electronic and other equipment. Is this the general area in which you have worked - Europe or North America - for many years?

May we hear from you?

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(Resonant Ruth continues here)

But the manufacturers of your automobile strove to avoid structural resonances. They avoid "stacking" their resonances. The engineer designing a speedometer, for example, asks the instrument cluster designer, "What are the instrument cluster natural frequencies? I want to avoid them."

The instrument cluster designer asks the instrument panel (IP) designer, "Hey, what are the IP natural frequencies? I want to avoid them."

The instrument panel (IP) designer asks the body designer, "Hey, what are the body natural frequencies? I want to avoid them."

The body designer asks the suspension designer, "Hey, what are the suspension natural frequencies? I want to avoid them."

I teach students in my short courses the "Golden Rule" of successful design: **Thou Shalt Not Stack Thy Resonances.**

This rule applies not only in avoiding noises emanating from automobile parts, but also in preventing early structural failures in land, sea. air and space vehicles.

Wayne Tustin, ERI's president, can be reached by e-mail or phone (805) 564-1260.

Thanks much to our readers who are collaborating with our new column "Resonant Ruth"! We will continue to post your contributions on our upcoming issues. Stay tuned!

Resonance can be useful in applications where you use vibration to screen, compact or dose material. The fascinating thing about vibration is, that it can be most harmful in most applications but also be used in a positive way for others.

By Jesper Boesen Nielsen, Denmark

In answer to your question about a "useful" resonance story I would suggest that most any (non-electronic*) musical instrument is a productive use of resonance. From Helmholtz resonance of the air column of a wind instrument, to the plucked string of a banjo, resonance is at play.

On a more "engineering" level, the tuned vibration absorber is a good example of a useful resonance. They've been used to tame the destructive resonances in high-tech aerospace applications, and even to reduce wind-vortex induced vibration in the Alaska pipeline and street light poles.

*(*) Of course I am sticking to the mechanical resonant domain with these examples. As an electrical engineer, I would be hard-pressed to find any modern electronic device without some useful resonant circuit, from the crystal oscillator** in a CPU-based device, to the tuned circuits in an RF receiver.*

(**) I know – the crystal is really a mechanical resonant device, but its output is electrical.

By Kevin C. Cooney, Pennsylvania

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