

# STRESSING THE DIFFERENCE - Quasi Random vs. Repetitive Shock Vibration.

R C Page (Reliability Plus)  
&  
By R H Weinmann (Screening Systems, Inc.)

This article is based on an original presented by Robert W Weinmann (Screening Systems, Inc.) at the AST 2000 Conference at Boulder, Colorado in October 2000 and was originally published in "Environmental Engineering" Dec 2001

## Introduction

All companies insist that their new products must be developed and thoroughly validated in the shortest possible time and must, of course, exhibit exceptional field reliability and robustness.

The techniques of HALT and HASS have developed from the need to achieve these two objectives. The methods fully complement established design-proving techniques, which although effectively validate the functionality of a product in extreme conditions, often do not thoroughly investigate its true long-term robustness.

To help engineers to develop robust new electronic designs in the shortest possible time, Highly Accelerated Life Testing (HALT) and its production counterpart Highly Accelerated Stress Screening (HASS) are becoming indispensable tools.

HALT and HASS require the careful use of the two most effective stress-screening tools – rapid temperature cycling and omni-axial random vibration. In this article we will examine the importance of using a correctly designed vibration table in order to achieve optimum results during screening.

## Designs of pneumatically actuated vibration tables

Omni-axial vibration tables use compressed air for their motive power, driving mechanical vibrator elements mounted under the table. Currently there are two vastly different types of pneumatic vibration systems available. Unfortunately key differences in the design and performance of these tables are often not fully understood.

The first is a Repetitive Shock (RS) system that includes an "Un-damped, Solid Table" also referred to as a "Rigid Table". The second type of system includes a "Damped, Segmented Table" also referred to as a "Quasi-Random" table. Hereafter, this second design will be referred to as a (QRS) table, for "Quasi-Random System" table.

The two vibration systems that are currently commercially available are totally different, and their probability densities of acceleration are quite different as well.

The commonly accepted definitions on the two systems are:

- (1) The RS system (un-damped, solid) utilizes a *patent pending* complex internal geometry rigid table, which is fundamentally designed to directly transmit the vibrators' shocks to the product under test.
- (2) The Quasi-Random system (damped, segmented) utilizes a *patented* modally rich, highly damped table, which responds to the shocks and, in turn, transmits the table's responses to the product under test.

Any technical comparison between pneumatic vibration QRS and RS tables must include such factors as the spectrum shape, bandwidth, and uniformity of the vibration spectra at the table's top surface along with the time it takes each table to precipitate latent defects in a Unit Under Test (UUT).

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Both the QRS table and the RS table receive their energy as a series of impulse shocks applied to their bases from multiple pneumatically driven reciprocating vibrators. The vibrators used in the QRS systems are illustrated in figure 1.

A single impact of the elastomer-ended piston against the fixed anvil generates a high peak impulse followed by rapidly degenerating ringing energy. The spectrum generated from such a single impact peak is composed of a multitude of frequencies from a low approaching 0 Hz. to a high determined by the design of the elastomer impact end. The desired roll-off frequency is designed into the vibrator by selection of materials and geometry and is a crucial element in the performance of the vibration system.

The roll-off frequency at the high end is determined by the time that the elastomer end remains in contact with the anvil and is calculated by taking the reciprocal of this time. For example, if the contact time is 0.5ms, then the roll-off frequency will be about 2000 Hz.

If a piston repeatedly impacts the anvil at a constant number of impacts per second, the lowest frequency generated changes to the repetition rate of the piston while the high frequency roll-off remains the same as for a single impact. In addition, the energy is focused at frequencies that are multiples of the repetition rate.

For the QRS type table the impact pulses are distributed randomly with time and there are few if any “extended periods of essentially zero G amplitude between the pulses”.

Looking at the frequency domain, the idealized Power Spectral Density (PSD) spectrum generated by a single vibrator running at 25Hz repetition rate and attached to a structure that has no resonance below 3000 Hz and running at a constant air pressure is illustrated in figure 2.

The vertical lines and spaces indicate the evenly spaced acceleration energy at frequencies that are multiples of the piston’s impact frequency. The pressure of the compressed air supplied to the vibrator is automatically varied by the QRS table controller. As this happens, both the level of the energy imparted to the table and the piston impact frequency change. This frequency change results in a shifting of the acceleration lines to a different set of frequency multiples. Continuously modulating the air supply will continuously change the frequency content of the vibrator’s energy, which acts to fill in the spaces.

In practice, a QRS table will be fitted with a number of pneumatic vibrators. The idealized PSD spectrum for multiple vibrators attached to a structure that has no resonance below 3000 Hz and running at several different modulated air pressures is shown in figure 3.

Notice that there is acceleration energy at all frequencies within the bandwidth from around 5 Hz to above 2000 Hz. The amplitudes of this acceleration energy at each frequency are randomly distributed so the figure is actually a probability function of the acceleration over a short time span.

The low frequencies, down to approximately 5 Hz, are the result of sub-harmonics generated from beat frequencies between the multiple vibrators - running at various continuously changing repetition rates.

### **Frequency Response**

The QRS type table differentiates itself from the RS table starting with its unique design. A multitude of damped resonant structures are inherent in the QRS vibration table. The various layers of structure resonate in response to the vibrators’ excitation while the damping acts to attenuate the peak accelerations and distribute the resonant energy across an increased bandwidth. A pictorial representation of the QRS table’s acceleration response to the vibrators’ input is shown in figure 4.

It is these complex damped table resonances that are responsible for generating the almost infinite number of omni-axial force vectors that are present at the top of the QRS table and that are transmitted to a UUT during a vibration screen.

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In contrast, figure 5 shows how each size RS table acts like a flat square plate which exhibits a few dominate resonant plate bending modes, the frequencies of which are determined by the size of the particular table. Such a plate will exhibit large displacements at the corners for all the modes.

The different methods of energy transfer by the two types of tables yields very different results.

Figures 6a and 6b clearly show the difference in the Power Spectral Density and Bandwidth of the QRS and the RS type table systems respectively. The sine-over-random content in the RS table spectrum is generated by the sinusoidal ringing of the table plates at their dominant plate mode resonances and is therefore predominately only in the Z-axis perpendicular to the plate.

Overlaying these two different spectra (figure 7) the blue highlighted area now shows the increased concentration of energy at the lower frequencies from the QRS table. This low frequency energy is a major contributor to the effectiveness of this table because of the high displacements that are generated.

### Energy Balance

An additional and equally important parameter is the balance between the accelerations in the X, Y, and Z-axis. Early in the development of the QRS table it was decided that the accelerations should balance (i.e. the  $G_{rms}$  levels be equal to each other) at some distance above the table surface. The objective being to excite a product with equal levels of energy in all axes at its centre of mass. This distance was determined by considering the typical products that would be installed in fixtures and mounted on the table. A compromise was struck between large "black box" products and low profile printed circuit boards. As a result the QRS type tables are optimised for X Y Z balance between 1" and 2" above the table top surface.

The QRS table utilises six feedback accelerometers mounted on two widely spaced tri-axial blocks. These accelerometer inputs are used by the closed loop control system to apply consistent and repeatable  $G_{rms}$  levels from test to test. It is standard practice to mount them within the first two inches above the table surface and leave them in the same place for all tests.

### Uniformity of Acceleration

A series of comparative measurements of acceleration uniformity were made across a 38" x 26" QRS table and a 36" x 36" RS table (figures 8 & 9). Small histogram charts with different length bars proportional to the X, Y and Z  $G_{rms}$  show the acceleration levels at different locations across the table top surfaces.

A further key parameter that differentiates the two table designs is evident by inspecting the results. This parameter is the uniformity of the  $G_{rms}$  acceleration across the table surface. The QRS tables do not have the corner flexing and non-uniformity characteristics exhibited by the rigid plate RS tables. Because different sized RS tables are in effect different sized structural plates, each table size has a different resonant frequency ( $F_n$ ) and differing spectral characteristics that will affect a UUT in different ways. It is apparent that the location of the UUT on the table can have a major influence on the vibration energy that it receives.

In contrast, the QRS table is made up of several damped layers. Furthermore, the top layer is *segmented* into small irregularly shaped structures. Low frequency corner resonances are significantly reduced and the table  $G_{rms}$  acceleration across the top surface is therefore more uniform. In addition, the segments are designed for all QRS table sizes to transmit the *same* spectra such that any test assembly will receive the same vibration spectra regardless of which size QRS table is used, or where on the table the UUT is fixtured.

A further comparative analysis was made of the uniformity in percentage terms. Results were averaged in each of five equivalent overlapping zones – one central and four corner zones (designated A to E) across each table. Figure 10 shows these results and the remarkable consistency achieved by the QRS table. The top two charts show the variation of the mean % of each zone compared with the mean % of all 5 zones. The lower charts show the worst-case zone maximum % from the mean % of all zones.

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### What the differences mean in practice.

Results from work done by the Ford Motor Company and originally published by Henderson (Ref 1) provide a valuable insight into the real benefits of using the QRS table.

Figure 11 summarises the evaluation performed on identical and similarly defective “computer boxes” when screened on a QRS and an RS table. Superimposed on the measured UUT response spectrum on each table is the approximate “Parts Resonance Span” (frequency bandwidth 200 Hz to 2000 Hz) that will cause nearly all electronic components to resonate.

Also inherent in the design of the QRS table is the *focusing* of the vibration energy into the frequency band (5 Hz to 3,000 Hz) that is most effective in fatiguing defects to failure. In the comparison the QRS type table found the same defects at the same  $10G_{rms}$  intensity *seven times faster* than did the RS table. This is a direct result of the concentrated vibration energy within the effective bandwidth of 5 Hz to 3,000 Hz.

This is important because the purpose of the vibration screening table is to supply vibration energy to a UUT such that its components resonate at their natural frequencies. This component resonance, along with board and other structural bending, is what generates the local stresses at the many junctions in and around the components. And it is these stresses that fatigue the weak or flawed junctions to failure during a product screen.

It should also be noted that most circuit board assemblies have resonant frequencies ( $F_n$ ) below 300 Hz. As a result, these boards will tend to filter out higher input frequencies because of the attenuation of vibration at input to resonant frequency ratios higher than  $\sqrt{2 \cdot F_n}$ . It is, therefore, important for a vibration table to generate significant energy over the component resonance bandwidth to compensate for this attenuation.

### Conclusions

Through a lack of appreciation of their important differentiating characteristics pneumatic vibration tables that utilize damped, segmented technology are often categorised with pneumatic vibration tables called Repetitive Shock (RS) systems.

This is an error because these are two totally different pneumatic table designs that generate vibration spectra in different ways and yield *significantly different* results. Although the pneumatic vibrators in both types of tables contain pistons, which repeatedly shock the bottom of the table, this is where the similarity ends.

The damped, segmented Quasi-Random table (QRS) deserves its own category so as not to be confused with an RS table when performing HALT, HASS or screening efficacy comparisons between shaker technologies.

### Reference

“Get a handle on fatigue to improve results from HALT-HASS stress screening machines”  
by George Henderson (GHI Systems, Inc.) presented at the AST Workshop, Boston, Oct 1999.

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## FIGURES

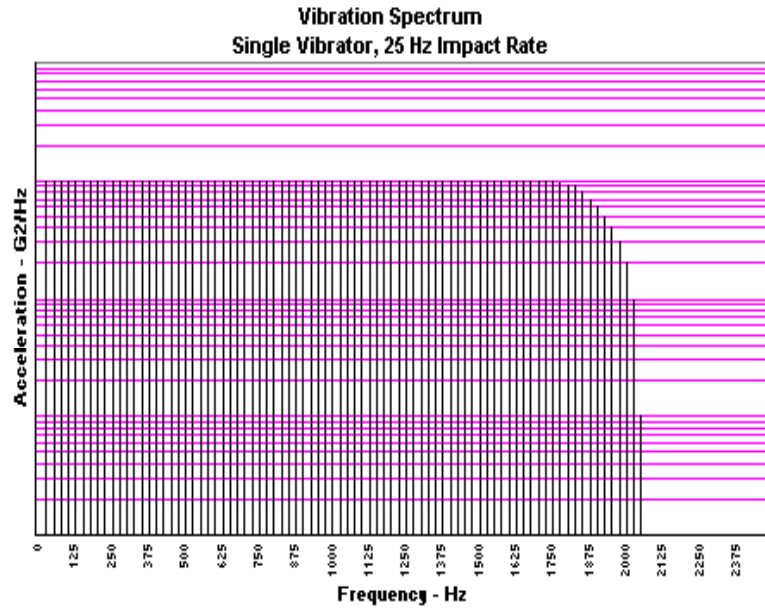


Figure 2 - Frequency domain (Acceleration vs frequency) for a single vibrator

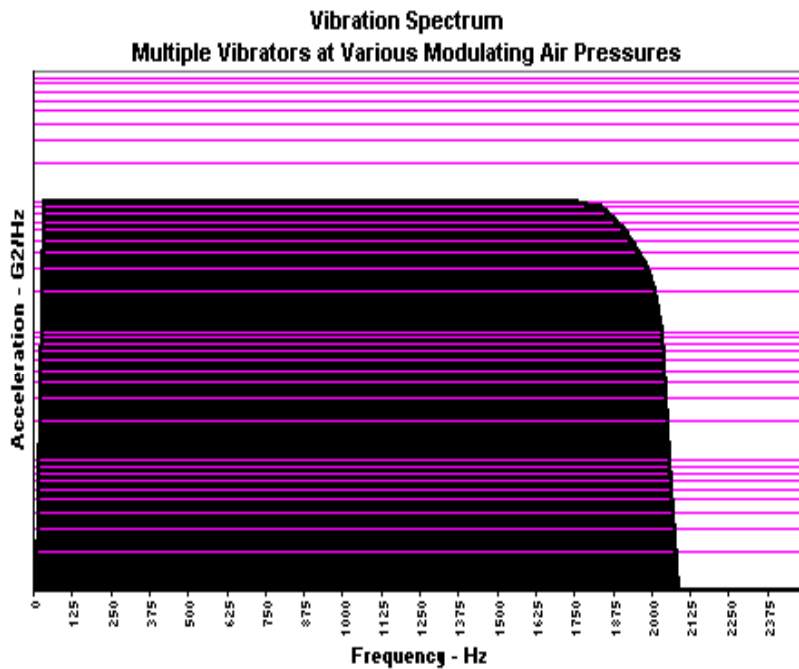
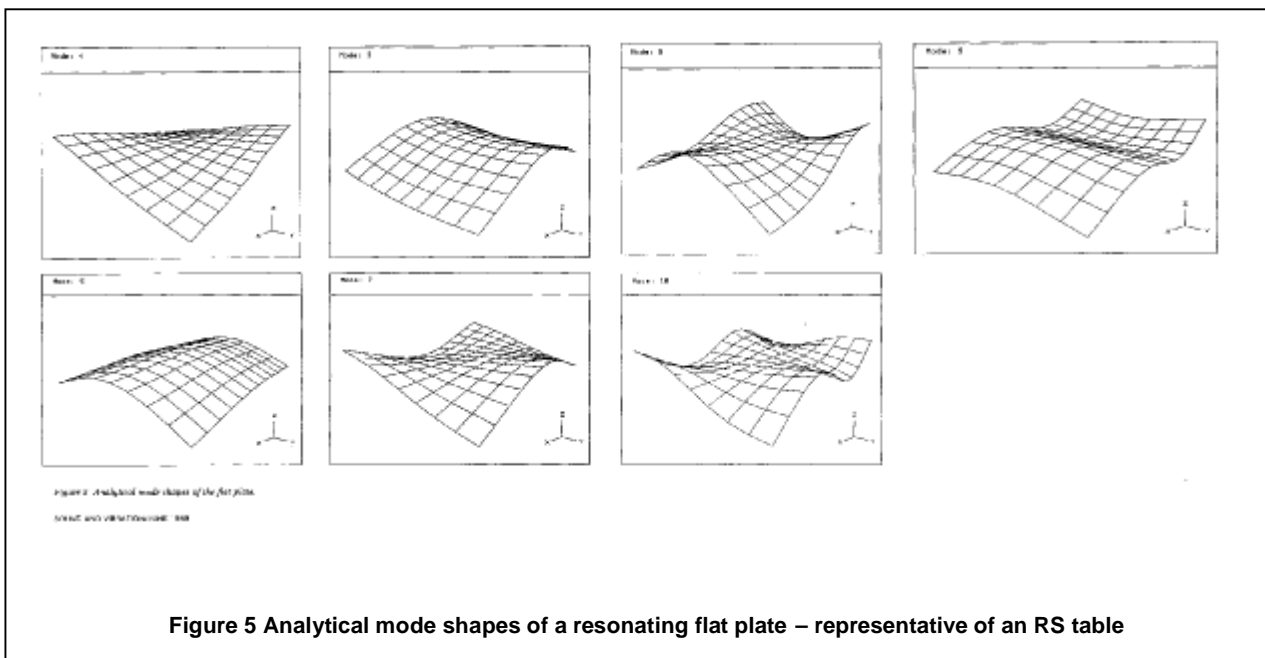
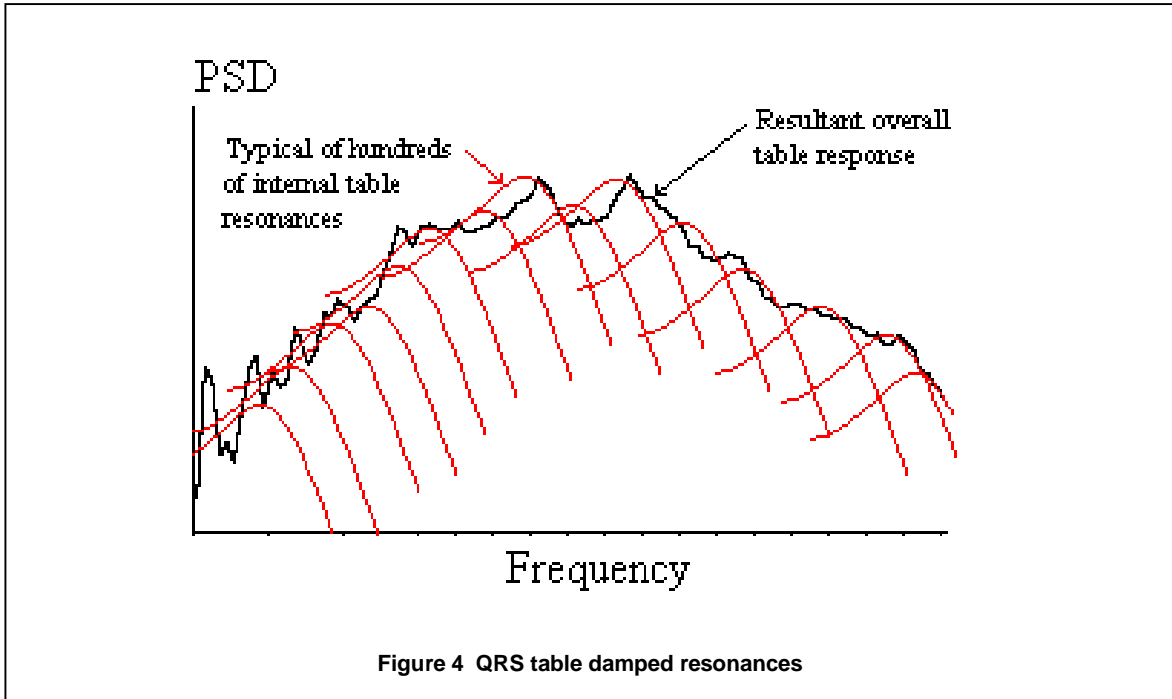


Figure 3 Vibration spectra from multiple vibrators

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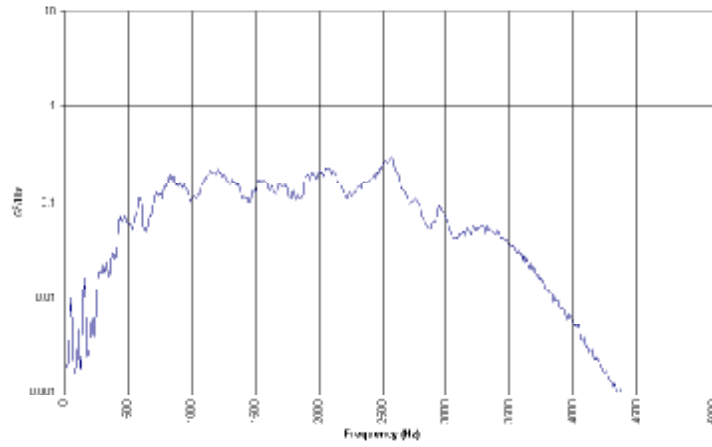


Figure 6a PSD for QRS table

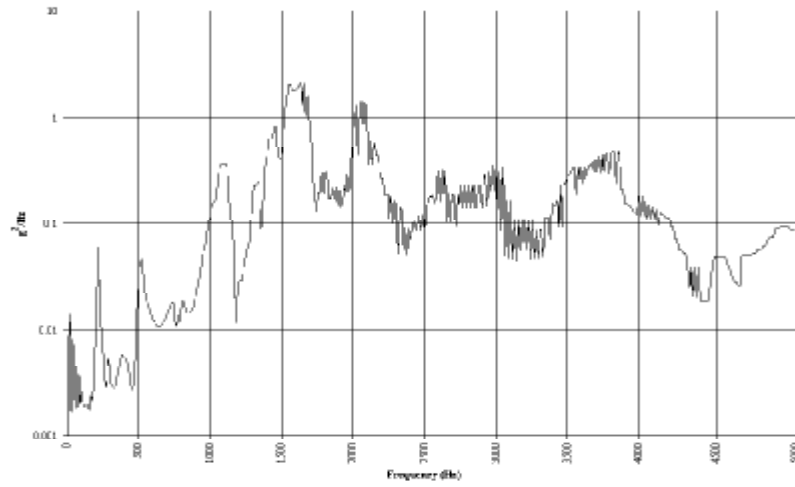


Figure 6b PSD for RS table response

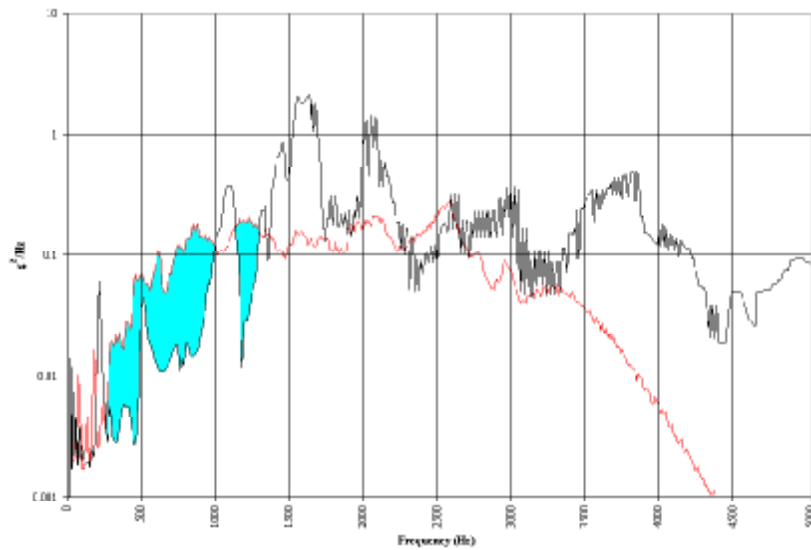


Figure 7 - Unequal energy distribution in the RS table PSD

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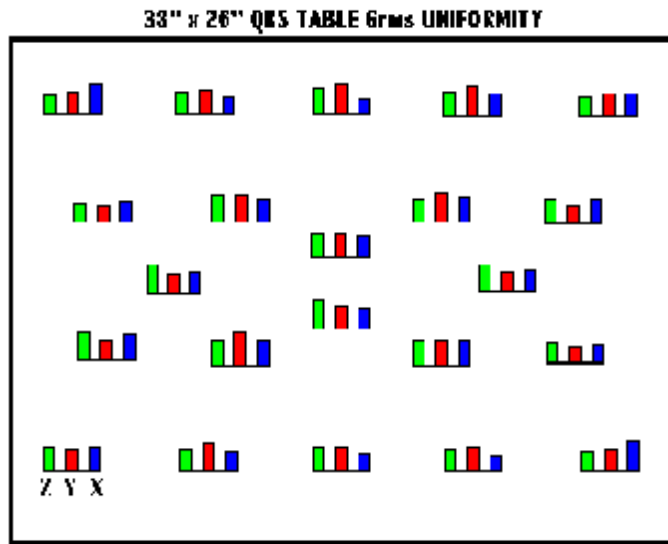


Figure 8 Acceleration uniformity across a QRS table

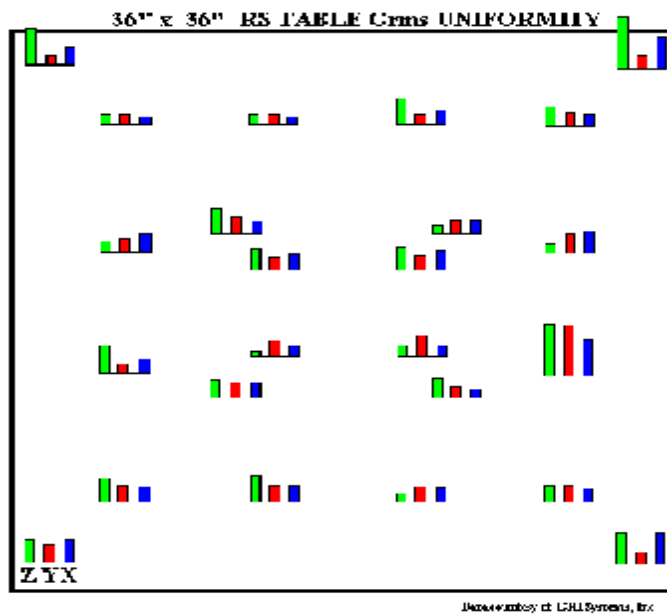


Figure 9 Acceleration uniformity across an RS table

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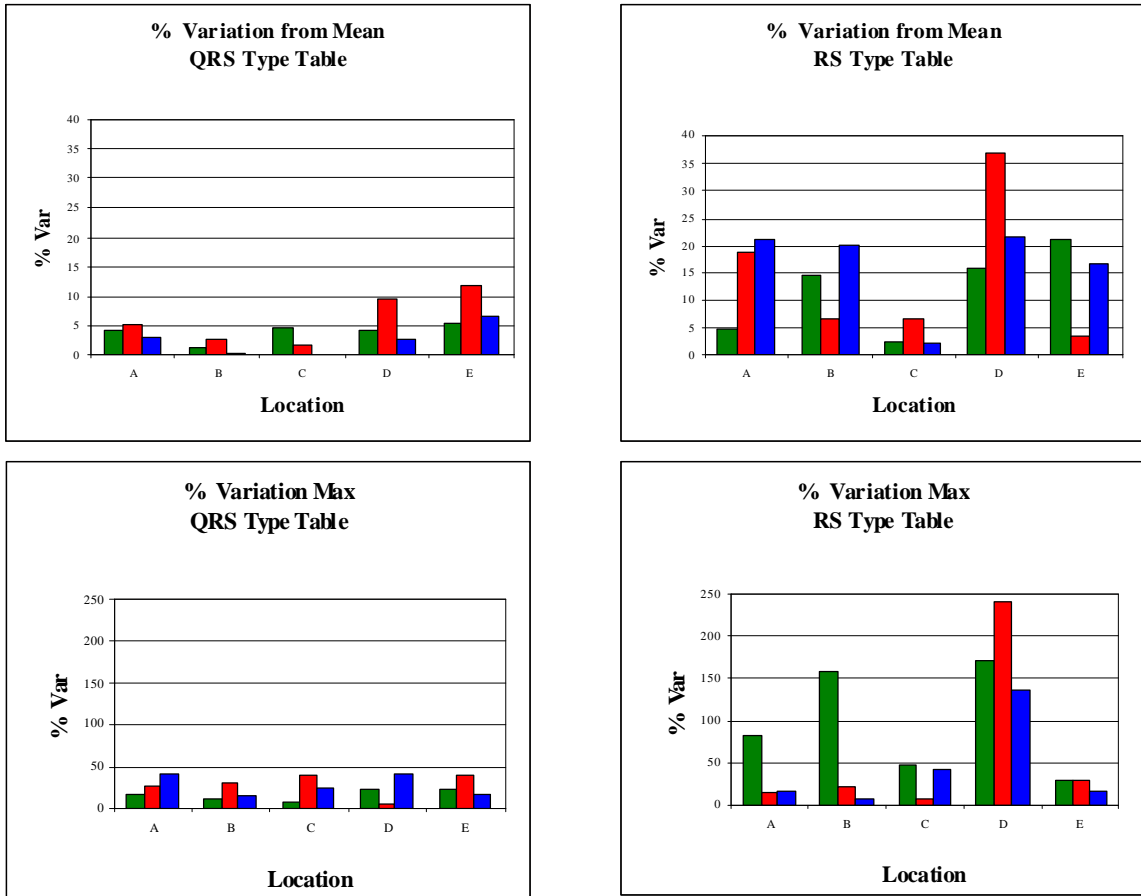
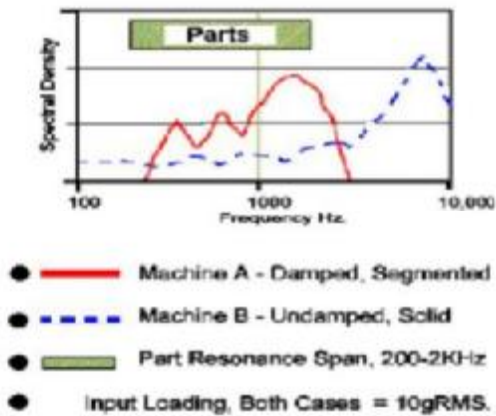


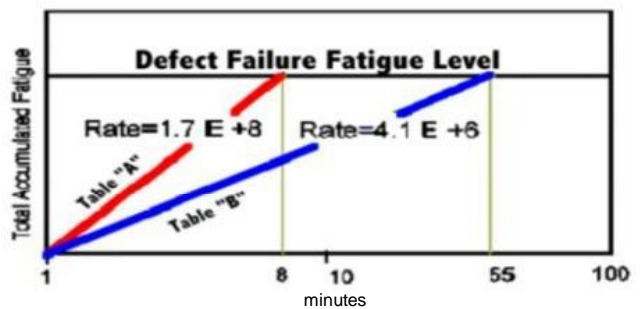
Figure 10 QRS and RS table acceleration uniformity comparison

## PSD Relationships Ford case history\*



\* PSDs from dissimilar machines when screening "computer boxes" with an identical defect.

## Tracking fatigue explains scatter



For identical defects, and the **same g rms intensity**, different fatigue rates will precipitate the defect at **different times**, producing screen scatter.

Presented at 1999 Accelerated Stress Testing Workshop, October 26-28, 1999 Boston Massachusetts

Source: George Henderson, President GHI Systems Inc.

"Get a handle on fatigue to improve results from HALT/HASS stress screening machines".

Figure 11 RS & QRS tables - comparison of results