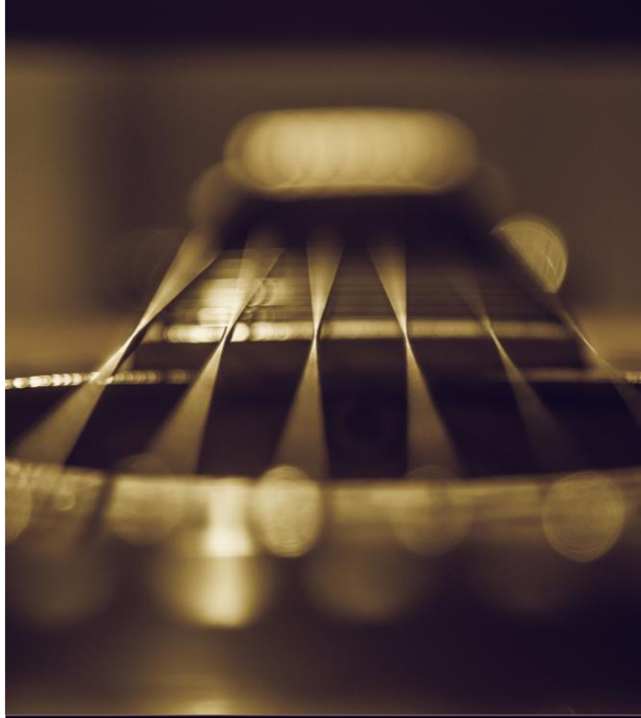
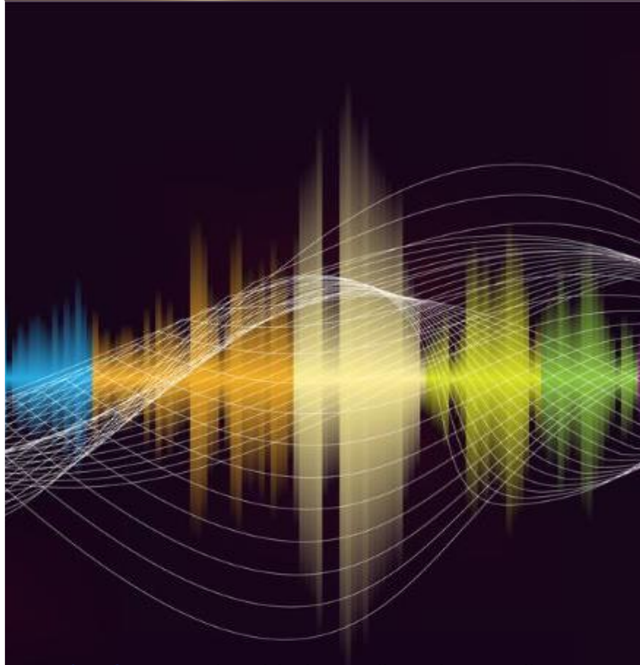
TAILORING OF VIBRATION ENDURANCE REQUIREMENT WITH RESPONSE SPECTRUM

RISE Research Institutes of Sweden

Martin Olofsson

RISE Chemistry & Applied Mechanics





RISE Research Institutes of Sweden

- State-owned research institute with a mission to be a strong innovation partner to corporations and society
- 2700 employees offer unique expertise in a wide range of knowledge and application fields (1/3 with a PhD)
- 100 testbeds and demonstration facilities

Short facts about RISE Applied Mechanics

- 50 researchers, engineers, technicians and admin staff
- Node for solid and structural mechanics inside RISE
- Large experimental & simulation capabilities
- Expertise in shock & vibration integrity and reliability




Example of load frame for material testing

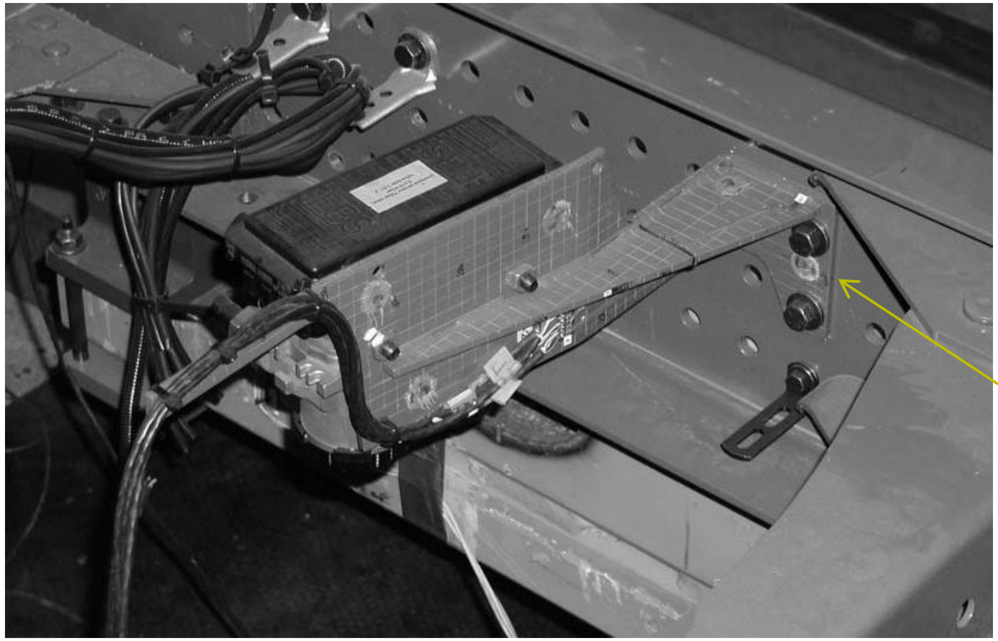
Max force 1.2 MN

Quality assurance of vibration-sensitive equipment

- Many components are mounted to a vibrating structure and can be damaged from the vibration
- The anticipated failure mode involves a resonance that amplifies vibration response so that stress amplitudes is causing fatigue
- Simplified component durability testing on a single-axis vibrator, early in a project, is an important part of the quality assurance work for such components

Vibration simulation on a shaker is difficult

- Vibration type is simplified to a sinusoid, with sweeping frequency, or stationary random vibration – a vibration analysis challenge
- The dynamic stiffness of the mounting fixture is important for correct stress response simulation
- The vibration excitation is applied sequentially in one direction at the time



Field data measurement set-up.
Excitation or response?
Substructure has limited stiffness.
Is a bracket resonance visible in the data?

How do we handle the uncertainty?

- Simplification consequences must be handled
  - Tailoring analysis method according to Lalanne minimises uncertainties from difference in vibration type
  - Simplification with very stiff fixture has many practical advantages during testing, but using a flexible substructure from the true application as fixture reduces uncertainty
  - The consequence from the uncertainty from single axis testing has been explored recently, in a thesis work
- <u>The conclusion that one would like to make from a successful vibration test is that the component will endure a lifetime in the true application</u>, with a certain, high probability
- Hence, because of the uncertainties, we must accept over-testing in order to be able to make the wanted conclusion (and be sure that under-testing is unlikely)

Signal analysis to derive optimal test specification

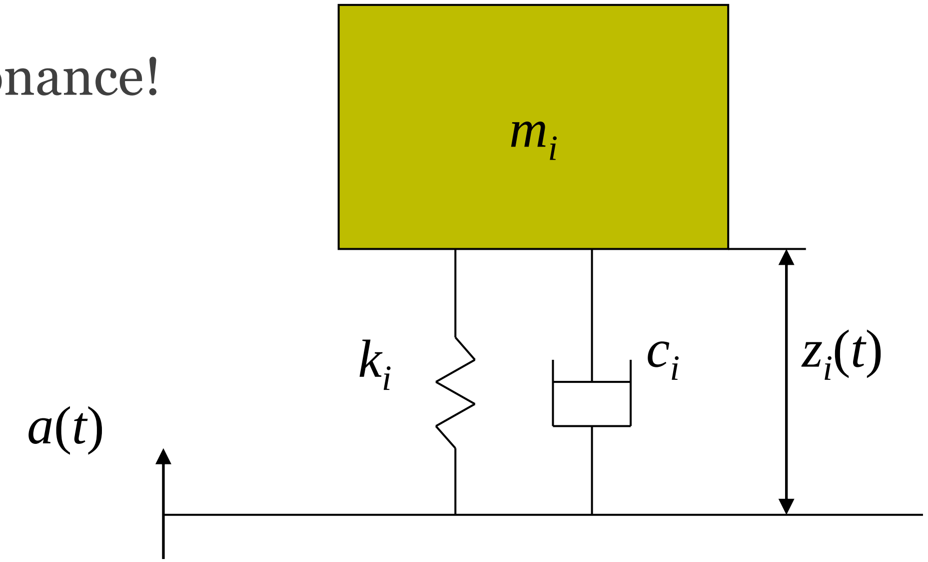
- ’Lalanne’s approach’ using Maximum Response Spectrum (MRS) and Fatigue Damage Spectrum (FDS) is used by default
  - suitable when you do not know the dynamical properties of the component - only the excitation is analysed and described
  - Best way to compare different types of vibration excitations, regarding damaging potential
  - one-dimensional vibration analysis – not a big deal since this limitation is already a fact for single-axis vibration testing
- Option for the ambitious: control of a known critical resonance response
  - this can be done once before the test, with accelerometer or strain gauge, and the result can be used to adjust the amplitudes around the particular resonance frequency

Definition of FDS and MRS

- Similar approach as when Shock Response Spectrum was proposed, for description of earthquake severity
- evaluation of effect from excitation, $a(t)$, on SDOF-systems with natural frequencies $f_i = f_1 + i \cdot \Delta f$, $i = 0, 1, 2, \ldots, N$

simple model of a resonance!

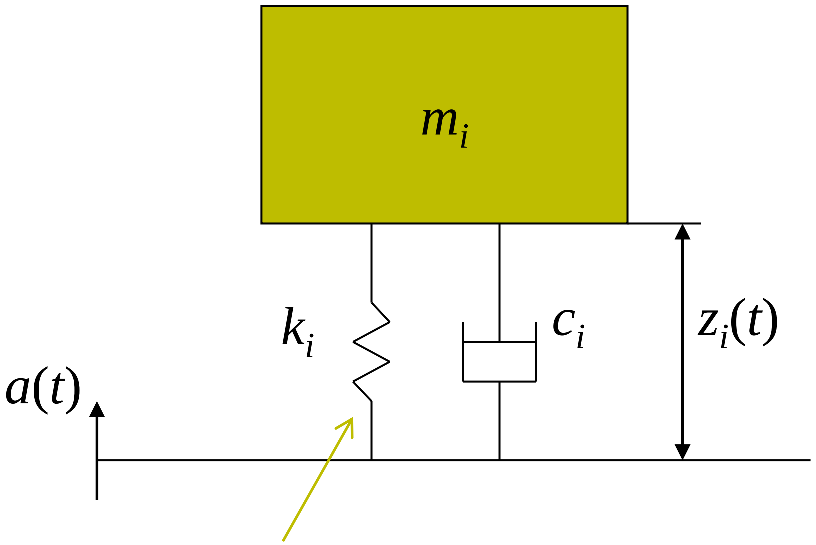
$$f_i = \sqrt{\frac{k_i}{m_i}}$$



Definition of MRS and FDS

Following steps are taken for SDOF system i:

1. Relative displacement response, $z_i(t)$, is calculated
2. MRS
   - $\mathrm{MRS}(f_i) = (2\pi f_i)^2 * \max_{0< t<T}(|\, z_i(t)|)$
3. FDS
   - Extraction of rainflow cycles from $z_i(t)$
   - $\mathrm{FDS}(f_i)$ = fatigue damage based on linear damage accumulation (Palmgren-Miner) – a relative measure



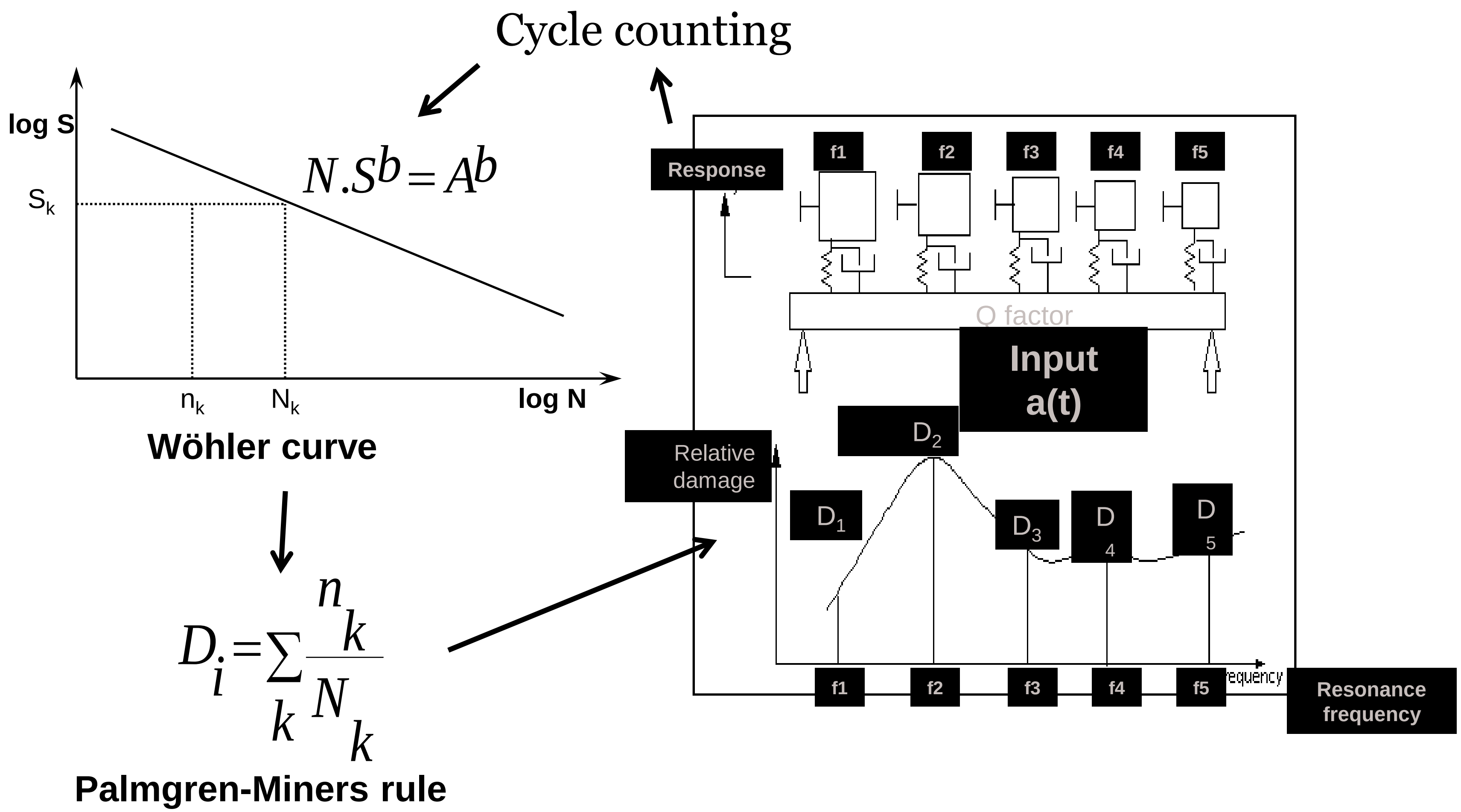
Spring stress is proportional to $z_i(t)$

Other parameters involved in calculation of MRS and FDS

- The only parameter for MRS calculation that needs a choice or assumption is the Q-factor, the damping of each SDOF-system
  - Q=10 is used almost always (= 5% relative critical damping)
- For FDS calculation we also need to choose a ’Wöhler slope’, which is the exponent in the damage accumulation formula and tells you how much (exponentially) more fatigue damage a large stress cycle causes compared to a smaller one
  - b = 5 is used almost always

  Other parameters are set to 1 as the choice does not matter for comparative analysis (positioning of Wöhler slope and relation between SDOF displacement and material stress)

Fatigue Damage Spectrum



MRS calculated from frequency domain excitations

- Random vibration, with specified Power Spectral Density, $G_{aa}(f)$

$$M(f) = \omega^2 \sigma_z \sqrt{2 \ln n_{0+} T}$$ expected value of extremum

$$\sigma_z{}^2 = \int G_{zz}(f) df = \int H_z{}^2(f) G_{aa}(f)\, df$$

  - n_{0+} is the expected number of zero crossings (+ slope) per unit of time, also derived from (area moments of) $G_{zz}(f)$
  - T is the exposure time (test duration)

- Swept sine vibration, with specified amplitude spectrum, $A(f)$

$$M(f) = QA(f) = 10A(f)$$ Extremum within the range of $A(f)$

FDS calculated from frequency domain excitations

- Random vibration, with specified Power Spectral Density, $G_{aa}(f)$

$$D(f) = n_{0^+}T\left(\sigma_z\sqrt{2}\right)^b \Gamma\left(1 + {}^b/_2\right)$$

expected value of damage

  - assuming narrowband response -> peaks have Rayleigh distribution and also that the 'other' parameters are set to 1
  - b is the Wöhler exponent (typically 5) and Γ is the gamma function

- Swept sine vibration, with <u>logarithmic sweep rate </u>(use it always!)
  - Complicated – Lalanne has a formula
  - …or create a time series and calculate MRS/FDS from that

Combining vibration environments in a life mission

- Assume that A and B are two vibration environments that a component is exposed to and that α and β are the number of repetitions for each environment, respectively, corresponding to a full product life mission
- Then,
- total MRS = envelop of MRS(A) and MRS(B)
- total FDS = α*FDS(A) + β*FDS(B)
- Hence, if you would take one measurement of A and one of B and glue α number of copies of A and β copies of B together, the resulting MRS and FDS would be the same as above

Advantages from using MRS/FDS

- Analysis of measured data done in time domain - no need for data classification (required in frequency domain analysis) and no issue with non-stationary data
- Easy to compare damage potential between measured data and vibration defined in frequency domain and to calculate spectrum or PSD for a damage-equivalent swept sine or random vibration, respectively
- Efficient data reduction and easy to merge results from different environments
- Effective reduction of vibration testing time, with control on max stress, when tailoring of test vibration is based on MRS and FDS
- Resulting vibration environment description with MRS and FDS can be recycled and is valid independently of component upgrades (or for all components mounted in the surrounding)

Pre-requisites and limitations

- Assumptions only valid for linear structure with light or moderate damping
  - true modal response assumed to be proportional to $z_i(t)$ (or stress in SDOF-system spring)
- Damage equivalence is based on Palmgren-Miner linear fatigue damage accumulation rule
- True Damping and Wöhler slope is generally not known
  - Deviations are of less importance when comparing environments, especially when vibration types are similar (only important to use the same parameter values consistently)
- Information about different frequencies acting simultaneously is lost (if two different modes are excited and both contribute to hot spot stress)

Motivation for safety margin

- A relevant vibration test should have MRS-values that exceed MRS-values from operating data with a safety factor, for all frequencies
- Even when measured data reflects maximum operating levels, uncertainties like
  - how well a vibration environment can be simulated when using only one excitation direction at a time,
  - influence from mismatch in dynamic stiffness in substructure and fixture,
  - limited amount of measured data from limited number of product individuals,
- … call for an extra margin
- Before quantifying the safety factor, one also needs to take into account
  - how the vibration test result is to be interpreted. Is it a verification test, for which the probability of under-testing must be low (or development test)?
  - the consequence of possible over-testing. How expensive is it to build in extra robustness in the design of the component?

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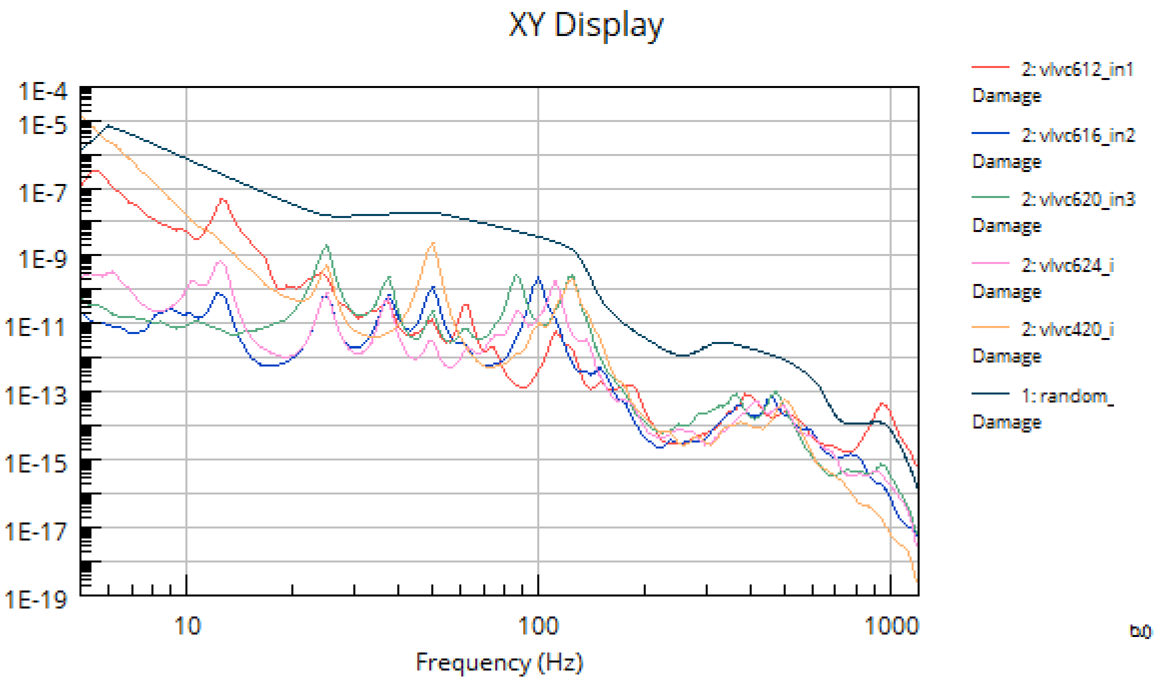
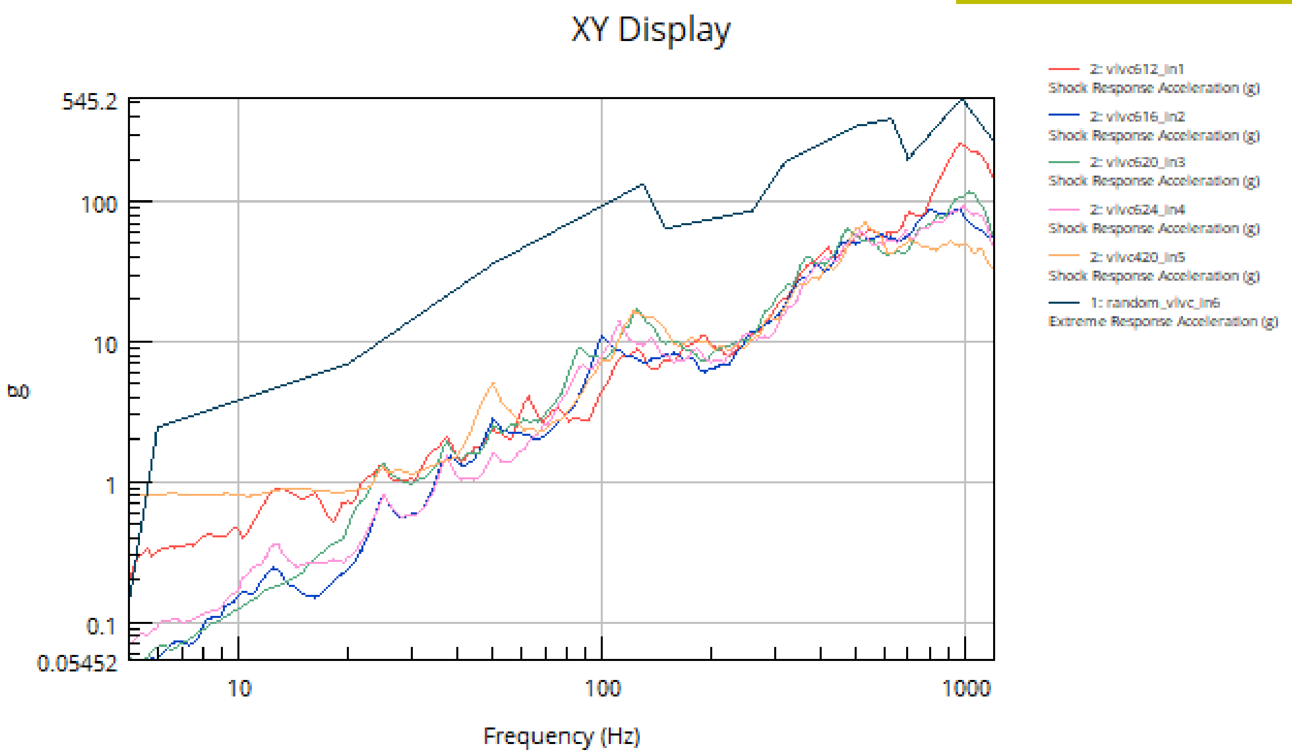
Tailoring of random vibration test in practice

- Start from a MRS target from operating data. Make more than one target if vibration types differ between different operating conditions, for separate simulation.
- Find test vibration PSD that gives a MRS that is at least 2 times higher than target
- Find test duration that makes FDS stay above target FDS with a factor of 15
  - Iteration process – repeated adjustment of PSD breakpoints are needed
  - Make a smooth PSD – accept compromised margin at peaks and high margin at valleys
  - If test duration is too long, test acceleration can be applied by increased PSD (and MRS)
  - Accept lower margin than 2 for high frequencies, when FDS margin is more than enough, and accept higher margin than 2 at low frequencies for sufficiency in FDS

Example of field data vs vibration test





Thank you!

- More questions are welcome!