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EXECUTIVE SUMMARY OF THE THESIS

A critical assessment on the mechanical load severity comparison, with focus on aerospace applications

LAUREA MAGISTRALE IN SPACE ENGINEERING - INGEGNERIA SPAZIALE

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1. Introduction

In the operational lifespan of everyday objects, from commonplace items like mobile phones, computers, wristwatches, and cars to advanced aerospace components like aircraft and satellites, exposure to various environmental conditions is inevitable. These conditions include temperature fluctuations, humidity variations, and mechanical stresses. Therefore, proactive measures are crucial during the design and development phases to ensure the durability and resilience of these items to diverse challenges they may face during their operational lifecycle. Qualification testing is a critical step in this process, confirming an item's ability to thrive in anticipated operating conditions. However, discrepancies can arise when the qualification load differs significantly from the actual operative load, necessitating the use of the concept of "severity" to quantitatively assess and compare the risk of failure under various loads. In simpler terms, if the severity of load A exceeds that of another load B, it implies that the risk of failure when subjected to load A is higher. In the context of testing, it becomes essential for the severity of the qualification load to surpass that of the operative load to ensure the item's robustness.

This thesis, in collaboration with *Thales Alenia Space*, has two main objectives focusing on mechanical aspects:

1. **Evaluation and Enhancement of Current Methodologies**: The first objective revolves around a comprehensive investigation of existing methodologies employed for evaluating and comparing the severities of different mechanical loads. This entails scrutinizing their accuracy, computational efficiency, and any possible areas for improvement.
2. **Alternative Approach for Load Severity Comparison**: The second objective involves proposing an innovative approach for comparing the severity of mechanical Multi Degree of Freedom (MDOF) loads. Unlike traditional methods that rely on the absolute acceleration of a Single Degree of Freedom (SDOF) system, this novel approach hinges on analyzing the interface (IF) forces acting on a system subjected to a 6 Degree of Freedom (6DOF) load.

1.1. Illustration of load severity comparison for SDOF cases

Mechanical load severity plays a pivotal role in the realm of engineering, particularly in synthesizing specifications and comparing various loads [3]. To illustrate its practical importance, a compelling example is provided.

In Figure 1, which focuses on a mono-axial direction, the red dashed line shows the Acceleration Power Spectral Density (APSD) from a unit subjected to acoustic loads during qualification, while the black dashed line represents the APSD of the unit's qualification random profile. Some peaks in the measured APSD exceed qualification limits, but APSD alone can't assess their significance. The Extreme Response Spectrum (ERS) is computed for both the measured APSD and the random qualification level.

The figure also displays the severities associated to the APSDs, which are given by the corresponding ERSs, indicated as continuous lines. The ERS is given by the acceleration which is exceeded only once over the duration of the random loading. Narrow peaks in APSD have minimal severity impact. However, there's a peak in the ERS of the measured data exceeding the qualification profile, possibly requiring delta qualification.

The figure also presents the unit's sine and shock qualification profiles, each with its severity, which is the SRS. For each frequency f, the SRS is given by the maximum absolute acceleration calculated by applying a deterministic acceleration time history to a SDOF system with natural frequency $f_n = f$. The SRS of the sine profile is indeed is approximated by multiplying the sine's amplitude by the qualification factor Q, while the shock qualification's severity is represented by the SRS of the shock itself. Remarkably, the ERS of the measured APSD for acoustic loads falls within the unit's shock qualification profile's SRS envelope. This is encouraging in order to believe that no damage nor degradations occurred after the acoustic test.

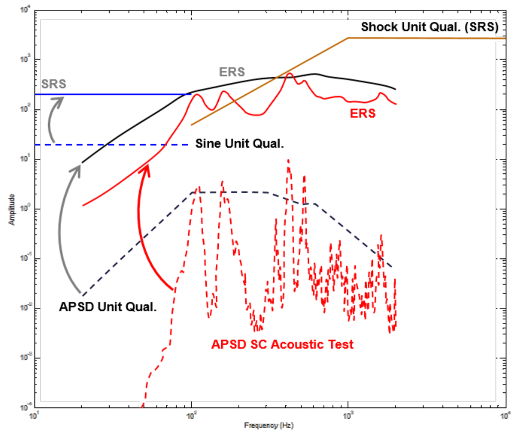

Figure 1: Example of mechanical load severity comparison application.

2. Shock Response Spectrum

SRS serves as a valuable tool to evaluate the severity of transient mechanical forces, encompassing middle to low frequencies and deterministic vibrations. To calculate an SRS, the load in question is applied to a standardized mechanical system, consisting of a support structure and N linear SDOF mass-spring-damper systems. Each system has a unique stiffness (k_i) while sharing the same damping coefficient. Subsequently, the maximum absolute response of each system is determined and plotted against their corresponding natural frequencies. Figure 2 provides a visual representation of this process.

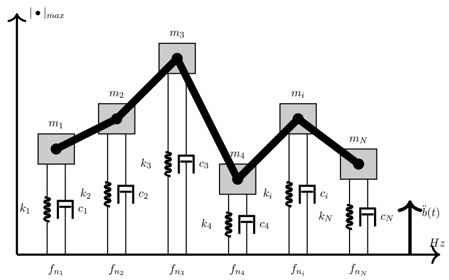

Figure 2: Conceptual scheme for evaluating the SRS.

In this setup, adjusting the k_i and m_i pairs allows for specific natural frequencies. The damping coefficient c_i is selected to achieve a quality factor of $Q = 10$, a convention in the aerospace industry for meaningful severity comparisons. In aerospace, absolute acceleration is the primary output parameter for shock response assessment.

2.1. SRS applications in space engineering

Satellite structures undergo rigorous testing to ensure their robustness under mission conditions. Qualification tests simulate the mechanical stresses during a mission, often utilizing shaker systems. However, current shakers face limitations in replicating 6DOF transient loads effectively. To address this, mono-axial excitation runs are performed using a sinusoidal sweep technique, aligning input test levels with mission conditions. The Equivalent Sine Input (ESI), a crucial parameter which represents the severity of the Launch Vehicle-Spacecraft (LV-SC) Coupled Load Analysis (CLA), guides secondary notching processes to ensure instruments and equipment experience accelerations equivalent to those in the actual mission.

2.2. ESI for a 2DOF system

To validate the ESI concept, a MATLAB-based 2DOF system was developed, considering both modal and transient analyses. Figure 3 depicts the system, with subscripts 1 and 2 representing a SC and a telescope (TS). The base of the system corresponds to the LV-SC IF.

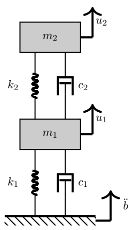

Figure 3: 2DOF Spring-mass-damper system.

The transient analysis provides a dynamic simulation of real-world transient events by incorporating the acceleration time history derived from the CLA. In contrast, the modal analysis emulates satellite tests conducted on a shaker system. Through a systematic parametric analysis, involving variations in the system's masses and frequencies, significant distinctions between the modal and transient analyses become evident. This contrast is especially pronounced as observed in Figure 4. These disparities underscore the inherently conservative nature of the ESI method concerning the capture of the true severity of the CLA, which, in turn, influences the secondary notching decisions made during testing procedures.

3. Extreme Response Spectrum

ERS serves as a tool for assessing the severity of transient mechanical forces associated with non-deterministic vibrations. To conduct random testing effectively, it is imperative to define a random test spectrum. Random vibrations are

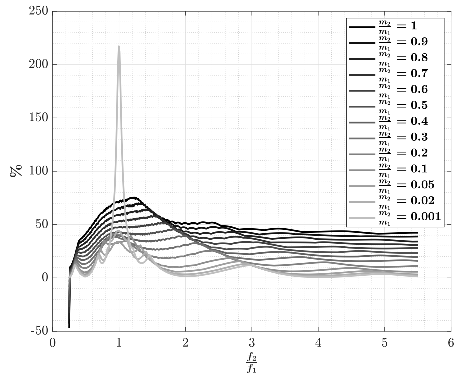

Figure 4: Maximal m_2 absolute response difference between ESI and CLA inputs.

characterized by their Power Spectral Density (PSD), representing the distribution of power across different frequencies. Similar to the SRS, the severity of the input PSD can be visualized through a curve.

3.1. How to compute the ERS

Calculating the ERS involves applying the PSD under consideration to a standardized mechanical system, such as the one depicted in Figure 2. The next step entails evaluating the Root Mean Square (RMS) of the output for each SDOF system and plotting it against the corresponding natural frequency. However, the RMS represents a mean value and not the largest peak. To provide a more comprehensive characterization of the output, the acceleration RMS must be multiplied by a factor $\sqrt{2\ln(fT)}$ (derived from Lalanne [3]), where T denotes the duration of the random excitation.
Two distinct approaches for computing the RMS are available:

3.2. Integral of the transmissibility approach

In this method, the output acceleration RMS is determined using the formula:

$$\ddot{u}_{rms} = \sqrt{\int_{f\_range} |SDOF\_TR(f)|^2 \cdot APSD(f)\,\mathrm{d}f} \quad (1)$$

Where $SDOF\_TR$ represents the transmissibility of the SDOF system.

3.3. Lyapunov approach

An alternative, computationally efficient approach based on the Lyapunov equation can be employed. However, this method is applicable when the SDOF system's input is a white noise. To adapt it for non-constant acceleration PSD inputs, a stable shape filter is introduced in series with the linear time-invariant system. The shape filter compensates for the non-constant acceleration PSD, ensuring system stability while accommodating the dynamic nature of the input.

3.4. Case study and comparison between the methodologies

As part of a case study, the qualification levels typically used in the space industry for SC units were analyzed. Figure 5 illustrates a comparison between ERS evaluations performed using two methodologies: SinePost, which adopts the integral of the transmissibility approach, and the Lyapunov method. Minor deviations at 100 Hz and 300 Hz are observed, primarily attributed to the limitations of the filter in accurately approximating sharp variations in the Acceleration Power Spectral Density (APSD) specification. Additionally, significant differences in the curve emerge at the extremities, around 20 Hz and beyond 1400 Hz, due to SinePost's integral accounting for the finite domain of the APSD.
Furthermore, a computational cost analysis of the two methodologies is presented, demonstrating that the Lyapunov method is more computationally efficient when looped over natural frequencies and when filter coefficients are known. However, its efficiency diminishes when factoring in the entire filter design process, making it slower compared to the classical methodology in those cases.

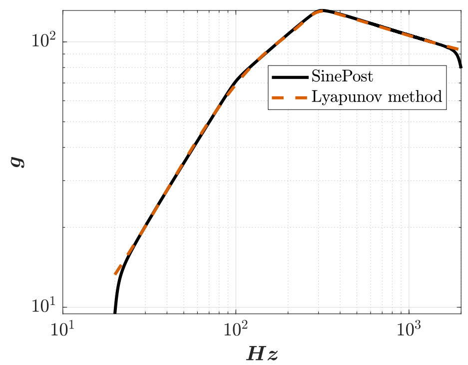

Figure 5: ERS: comparison of the Lyapunov methodology against SinePost.

4. Fatigue Damage Spectrum

The Fatigue Damage Spectrum (FDS) represents a critical tool for understanding how structures can sustain damage even under cyclic loading, even when stress levels remain below the ultimate static strength. This phenomenon is commonly referred to as fatigue. While the SRS and ERS are invaluable for evaluating a structure's peak response to shocks or mechanical vibrations, they fall short in providing insights into the resulting structural damage. The FDS, which builds upon the ERS concept, goes a step further by quantifying fatigue damage based on the frequency of a SDOF system. This framework facilitates the comparison of APSD specifications with varying durations.

4.1. How to compute the FDS

Calculating the FDS can vary depending on whether the acceleration data is available as a time history or represented by its Power Spectral Density (PSD). Regardless of the form, the process involves imposing the acceleration at the base of an array of SDOF systems. The formulation considers the stress in each SDOF system, which is directly proportional to a specific parameter, such as pseudo velocity (PV), relative displacement (RD), or absolute acceleration (AA). The choice of parameter can vary depending on the study's context.

4.2. Case study and comparison

For a practical case study, let's consider a component subjected to two distinct specifications: Specification A and Specification B. The goal is to determine whether the component needs to undergo testing again with the new specification. Details of both specifications are sourced from [1].
To conduct a comprehensive comparison of the severities between the two specifications, the analysis is performed under three different stress-proportional cases: PV, AA, and RD. The corresponding graph displaying these three cases can be observed in Figure 6. It's essential to note that although the crossing frequencies seem quite similar across the three cases, they are not precisely identical. For instance, the crossing frequency for stress proportional to PV and RD is 71.6 Hz, while for stress proportional to AA, it's 71.5 Hz. This subtle difference can be at-

tributed to the distinct nature of PV, RD, and AA in relation to the relative motion of the SDOF system and the mass's response to the base motion.
In summary, the choice of approach for evaluating the FDS for comparison purposes has a minimal impact on the conclusions drawn, as all cases lead to the same overarching conclusion.

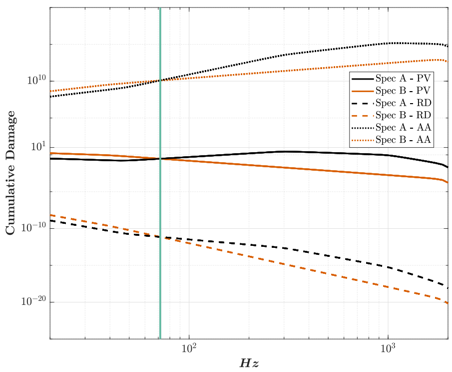

Figure 6: Comparison of the comparison between FDS evaluated considering the stress proportional to PV, RD and AA.

5. Proposed 6DOF approach for mechanical severity comparison

In the context of structural qualification dynamic tests, particularly in the low-frequency range, monoaxial shakers are typically used, with each axis tested individually using a sine sweep. However, operational loads in real-life scenarios are often multiaxial. This discrepancy between testing conditions and actual loads necessitates a method to evaluate the severity of both the qualification profile and the operative load profile. Additionally, a criterion for comparison is required to ensure that the qualification load's severity exceeds that of the operative load.
The existing criterion, which is based on SDOF systems, is effective when internal structural mode shapes exhibit minimal coupling. However, this approach primarily relies on SDOF systems for evaluating load severity. To address the complexities of multiaxial loads, especially in the presence of rotations, a refined criterion is needed.

5.1. Criterion for comparing different severities

In the realm of vibration analysis, comparing different severities is fundamental for assessing structural responses. The traditional SDOF criterion quantifies the severity of a base-enforced acceleration by the maximum absolute acceleration experienced by the mass of the SDOF system. However, this approach falls short in capturing moments when dealing with 6DOF systems. In this proposed approach, the structure is approximated by a mass that possesses six Degrees of Freedom (DOFs): three for translation and three for rotation. The mass is connected through six springs and six dampers to a base that is subjected to a 6DOF acceleration. The system is solved considering only the first natural frequency for each DOF. The relationship between the forces and moments at the base and the accelerations at the base is assumed to be represented by the Rigid Body Mass matrix of the structure, while the actual damping of the structure is also taken into consideration.
To formulate an appropriate criterion for comparing 6DOF severities, two distinct cases are proposed:

1. When base rotational accelerations are negligible compared to translational ones, the criterion can be expressed in terms of forces: For two 3DOF base-enforced translational accelerations (A and B), the mechanical severity of A surpasses that of B if the following conditions are met:

$$|F_x^A(\bar{f})|_{max} > |F_x^B(\bar{f})|_{max} \tag{2}$$
$$|F_y^A(\bar{f})|_{max} > |F_y^B(\bar{f})|_{max} \tag{3}$$
$$|F_z^A(\bar{f})|_{max} > |F_z^B(\bar{f})|_{max} \tag{4}$$

   Here, F represents the forces at the IF, and $\bar{f}$ denotes the frequencies considered.
2. When rotations at the base cannot be ignored, and the base-enforced acceleration exhibits 6DOFs, the criterion is adapted to consider forces and moments:
   For two 6DOF base-enforced translational accelerations (A and B), the mechanical severity of A outweighs that of B if the following conditions are met:

$$|F_x^A(\underline{f})|_{max} > |F_x^B(\underline{f})|_{max} \tag{5}$$
$$|F_y^A(\underline{f})|_{max} > |F_y^B(\underline{f})|_{max} \tag{6}$$
$$|F_z^A(\underline{f})|_{max} > |F_z^B(\underline{f})|_{max} \tag{7}$$
$$|M_x^A(\underline{f})|_{max} > |M_x^B(\underline{f})|_{max} \tag{8}$$
$$|M_y^A(\underline{f})|_{max} > |M_y^B(\underline{f})|_{max} \tag{9}$$
$$|M_z^A(\underline{f})|_{max} > |M_z^B(\underline{f})|_{max} \tag{10}$$

Once again, F and M represent the forces and the moments at the IF, respectively; instead $\underline{f}$ indicates the vector of the first natural frequencies for each DOF of the system.

This criterion offers a comprehensive framework for assessing and comparing structural severities in complex vibration scenarios, accounting for the multidimensional nature of 6DOF base-enforced accelerations.

5.2. Approximation made and applicability domain of the approach

The approach provides a simplified approximation of the global behavior of real systems and should be used for relative comparisons rather than estimating precise forces and moments. More specifically it's added value is expressed when assessing the severity of shaker qualification tests compared to expected operative loads.

5.3. Case study and comparative analysis: MATLAB vs. MSC Nastran

A case study involved comparing the mechanical load severity for two different load cases. The first case represents a 6DOF spacecraft base dynamic acceleration during launch, while the second case is the same, but with all rotations constrained to zero. The comparison focused on longitudinal base force (F_z) and lateral base moments (M_x, M_y).

The results obtained through the proposed approach in MATLAB, were compared with consistent results obtained from Finite Element (FE) direct transient analysis. The severity comparison is performed through the ratio between the force or moment calculated for the load case 2 and the force or moment calculated for the load case 1. As a criterion, positive and negative values are treated separately and the maximal of the two ratios is thus retained. The results are visually presented in Figure 7. This analysis effectively underscores the considerable influence of base rotations on load severity and affirms the efficacy of the proposed approach in furnishing valuable quantitative estimations, particularly for relative comparisons.

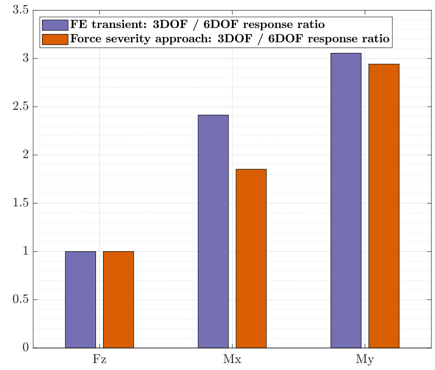

Figure 7: Histogram representing the largest ratios of the IF forces for the 3DOF and 6DOF cases.

6. Conclusions

In this thesis, a thorough examination was conducted on existing methodologies for assessing and comparing mechanical load severities. It was observed that these methods, originally designed for simpler systems, tend to lose accuracy as systems grow more complex, particularly when considering local accelerations within structures. This limitation, combined with the growing body of research on 6DOF shakers [2], highlights the necessity for a more consistent use of the severity comparison, as outlined in Section 5, which exhibited good capabilities in approximating real-world scenarios. Future research opportunities encompass the validation of this approach across diverse case studies, including scenarios involving random vibrations and fatigue assessment when dealing with 6DOF loads.

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