Synthesis of an Engine Vibration Specification and Comparison with Existing Qualification Specifications

Apports de la personnalisation pour synthétiser un environnement vibratoire de type Groupe Moto-Propulseur et le confronter avec des cahiers de charge forfaitaires

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Résumé
Un composant moteur doit satisfaire à des spécifications vibratoires de type forfaitaires données par les différents constructeurs. Ces spécifications, établies pour un même type de moteur, peuvent différer entres-elles de manière notable. Elles peuvent être de nature différente (Sinus Balayés ou DSP), plus ou moins sévères dans différentes gammes de fréquences ou encore couvrir des plages de fréquences plus ou moins larges.
Il a donc été décidé de réaliser des mesures d'environnement vibratoire sur banc moteur afin de constituer un spectre synthétique à comparer aux cahiers des charges de qualification.
On montre que si les spectres de qualification sont largement conservatifs à certaines fréquences, ils peuvent également sous-estimer les niveaux vibratoires dans d'autres.

Abstract
An engine mounted component must satisfy existing sign-off vibration tests that are given by each car manufacturer. These specifications established for the same engine technology may vary dramatically. They can be of different vibration types (swept sines or PSD), more or less severe and cover different frequency ranges.
It has been therefore decided to measure the real vibration environment on an engine test bench and synthesize the measured vibration data into a test specification in order to compare it with the required qualification specifications.
We will show that the required qualification specifications are largely conservative in some frequency ranges.
However, they also under-estimate vibration levels at others.

1. COMPONENT AND TEST DESCRIPTION

1.1. Description of the Engine Mounted Component

The component is an exhaust gas exchanger, mounted on the engine as described in Figure 1 below:

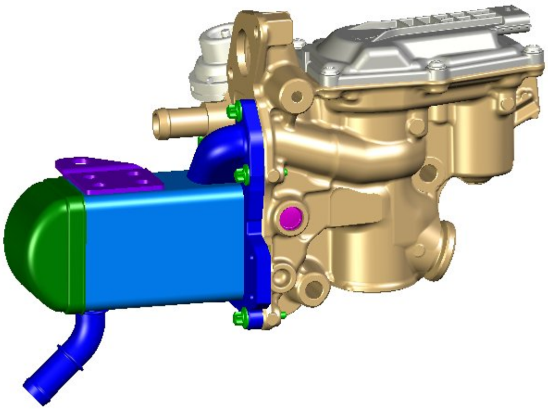

Figure 1: Global view of the system and its environment

1.2. Description of Engine Test

The goal (a representative test specification) requires measurements showing a good statistical content.
Four tests in total were performed on an engine bench.
- Two ramp up tests, where the engine profile is made of 9 constant RPM steps, from idle to maximum engine speed.
- Two resonance tests, where a slow ramped run up is followed by a dwell at max speed and by a ramp down back to idle.
This engine test profile is assumed to be representative of the operational profile of the engine.
Hundreds hours of this profile corresponds to the total life time of the vehicle.
We therefore can derive the number of vibration occurrences necessary for extrapolating to the total life time using the cumulative damage criteria (Miner’s rule).

Remarks:
Two tests of each type allow checking for repeatability. However, we would recommend more tests, 10 for instance, in order to take into account the variability of the environment.
The engine was warmed up before any measurement.

1.3. Description of Measurements Made

Triaxial accelerometers were placed at various positions:

- on the engine housing : to measure the source of vibrations
- next to the main fixing point of the component: to measure the component’s input vibration and derive synthesized PSD and swept sine vibration specification for testing
- on the component: to be used to correlate with the FE simulation

This data was measured and recorded in the time domain. The sampling rate was 25 600 Hz to ensure coverage of a large enough frequency range.

2. SYNTHESIS OF A VIBRATION PROFILE

An accelerated synthesized test signal is required that yields at least the same fatigue damage content as that seen by a component through its whole life.
The Fatigue Damage Spectrum (FDS) represents a plot of the total fatigue damage to occur over a period of time with respect to the natural frequency of the system.
The methodology and the algorithms used to derive the FDS are detailed in GAM-EG-13 [1], Lalanne [2] or AECTP 200 [3].

The aim here is to derive a damage-equivalent synthesized PSD or sine sweep from this engine test.

The Fatigue Damage Spectrum can be derived directly from the time series signal or from its PSD. The first case is called the deterministic approach, whereas the latter is the probabilistic one.

The deterministic approach, done in the time domain, first filters the signal using a set of SDOF frequency responses. It then rainflow cycle counts each filtered signal and computes the damage by using the Basquin equation and the Miner's sum.

The probabilistic approach works in the frequency domain. The PSD is multiplied with a set of SDOF frequency responses to predict component response in the frequency domain. A probabilistic rainflow cycle count is then derived from the stress PSD. This cycle count is then used to calculate the damage using the Basquin equation and the Miner's sum.

The resulting synthesized PSD relies therefore strongly on the FDS, which relies on the measured data.

The various steps listed below are covered in the next sections. These steps start with checking and understanding the measured data. The last ones take care of the synthesis of the environment and the comparison with some existing vibration specifications.

1/ View the measured data and look for anomalies
2/ Do statistics to assess stationarity
3/ Assess frequency content
4/ Calculate fatigue damage spectra
5/ Derive the synthesized vibration profile as a PSD or swept sine

2.1. Measurements Anomalies : Detection and Correction

This chapter helps us making sure the signals do not contain corrupted data that would lead to incorrect calculations.

The anomaly detection algorithms implemented within HBM's nCode GlyphWorks software detected outliers. These are marked as anomalies after review by the measurement specialist. Some anomalies like spikes were corrected (see figures 2 and 3 below), whereas some others like saturation lead us to mark the signal as unusable.

An example of anomaly correction is shown in the display below:

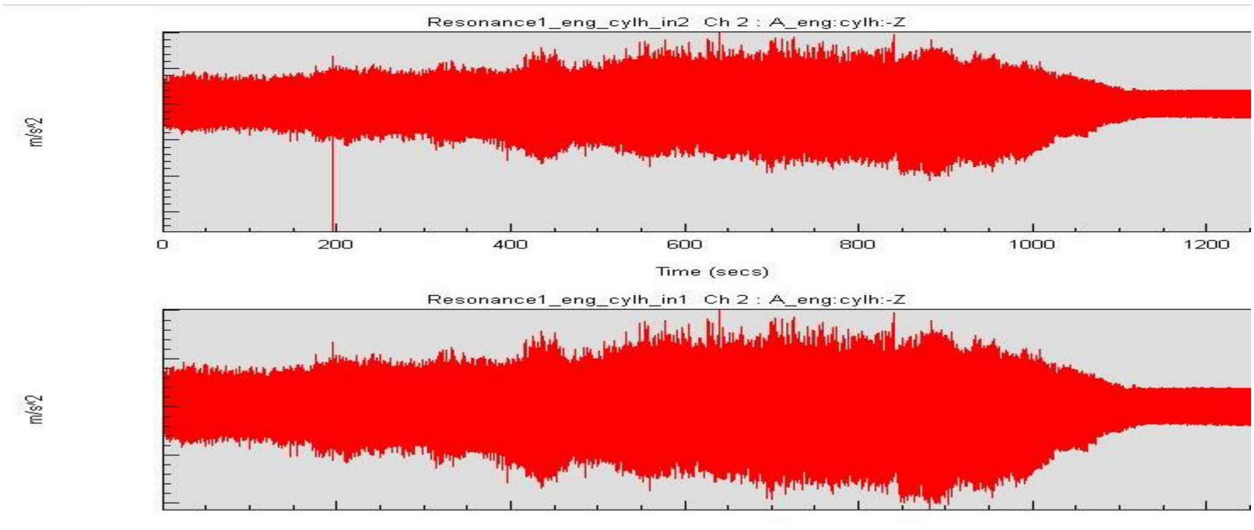

Figure 2 : Example of Spiked data Before and After correction

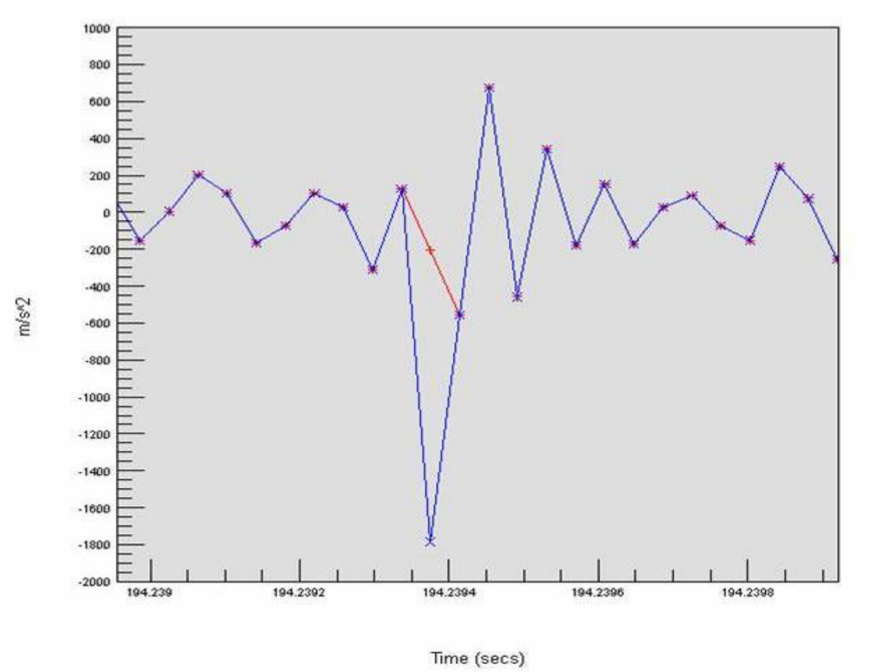

Figure 3: Zoom in to show correction (blue is original, red is corrected)

2.2. Stationarity Assessment

This section helps us deciding which approach (deterministic or stochastic) to use for the FDS calculation.

Global statistics were computed on each channel: max, min, mean, standard deviation, RMS, skewness and kurtosis. These global scalar values are a very analytical and useful way of quickly assessing the signal’s behaviour: is it centered around its mean? how “peaky” is it? etc.
We note that the value of the kurtosis exceeds 5, which indicates rather impulsive signals. This is confirmed by the fact that the extrema are often far beyond 3 times standard deviation (up to 6 to 10 times higher).

To complete these results, running statistical analysis were performed, showing whether the mean and standard deviation values remain constant over the sampling period (or vary slightly with RPM).
It is also interesting to calculate the probability distribution of the signal to see if it behaves as a Gaussian process.

All theses analyses can be gathered in a report that one can call a stationary test because the results of these tests give a good idea of the stationarity of a signal.
An example of such a stationarity test is shown in Figure 4 below:

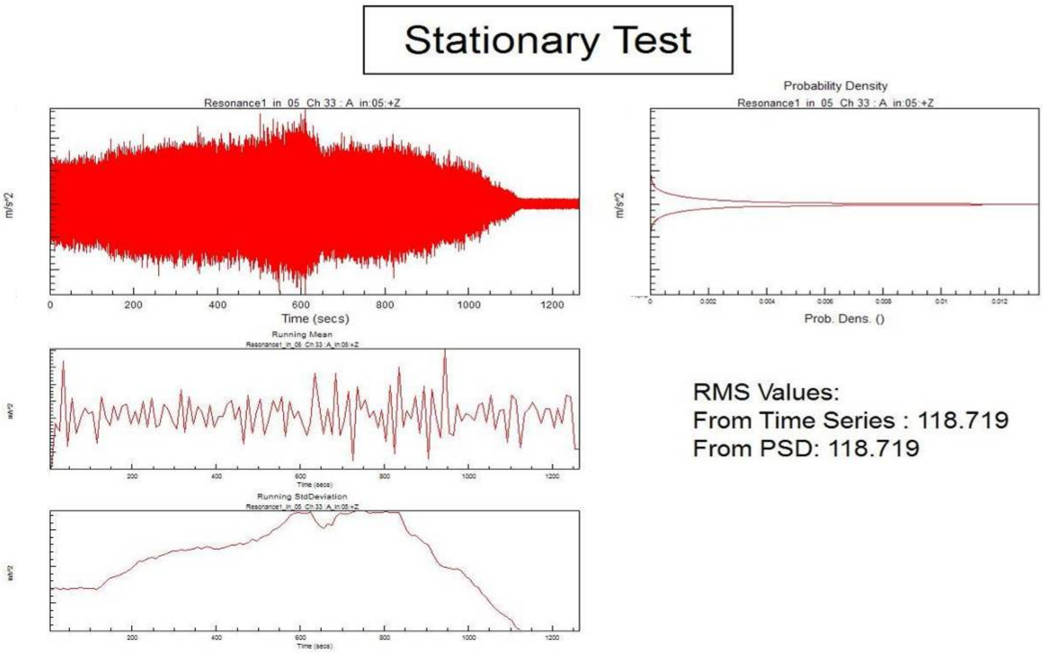

Figure 4: Example of stationarity report containing signal, PDF, running mean and running standard deviation

The probability density function does not exhibit the typical bell-shaped Gaussian distribution. Indeed, overlaying this probability density with a normal distribution defined using the global mean and standard deviation of the signal shows that the distributions are not comparable. This is as expected from the high kurtosis values seen in the time series data.

Moreover, we clearly see how the standard deviation varies with time (its wave form follows the RPM curve). This violates stationarity’s ergodicity requirement.

Therefore, these measurements appear not to be strictly “Ergodic Stationary Gaussian Random” and their PSDs cannot be representative. The Fatigue Damage and Response Spectra will therefore be calculated using the deterministic approach i.e. from time series directly.

2.3. Frequency Content

This section helps deciding which frequency range to excite in the vibration test.

A typical FFT is shown on Figure 5 below.
We clearly see high values at the high frequencies. We will first try to explain where they could come from and then we shall see why they should not have much importance in our case.
The engineering handbook "*Techniques de l'Ingénieur*" [5] contains a chapter dedicated to the engine environment vibration levels. The figure below summarises what the reader can find in the chapter BR2 BM2773.

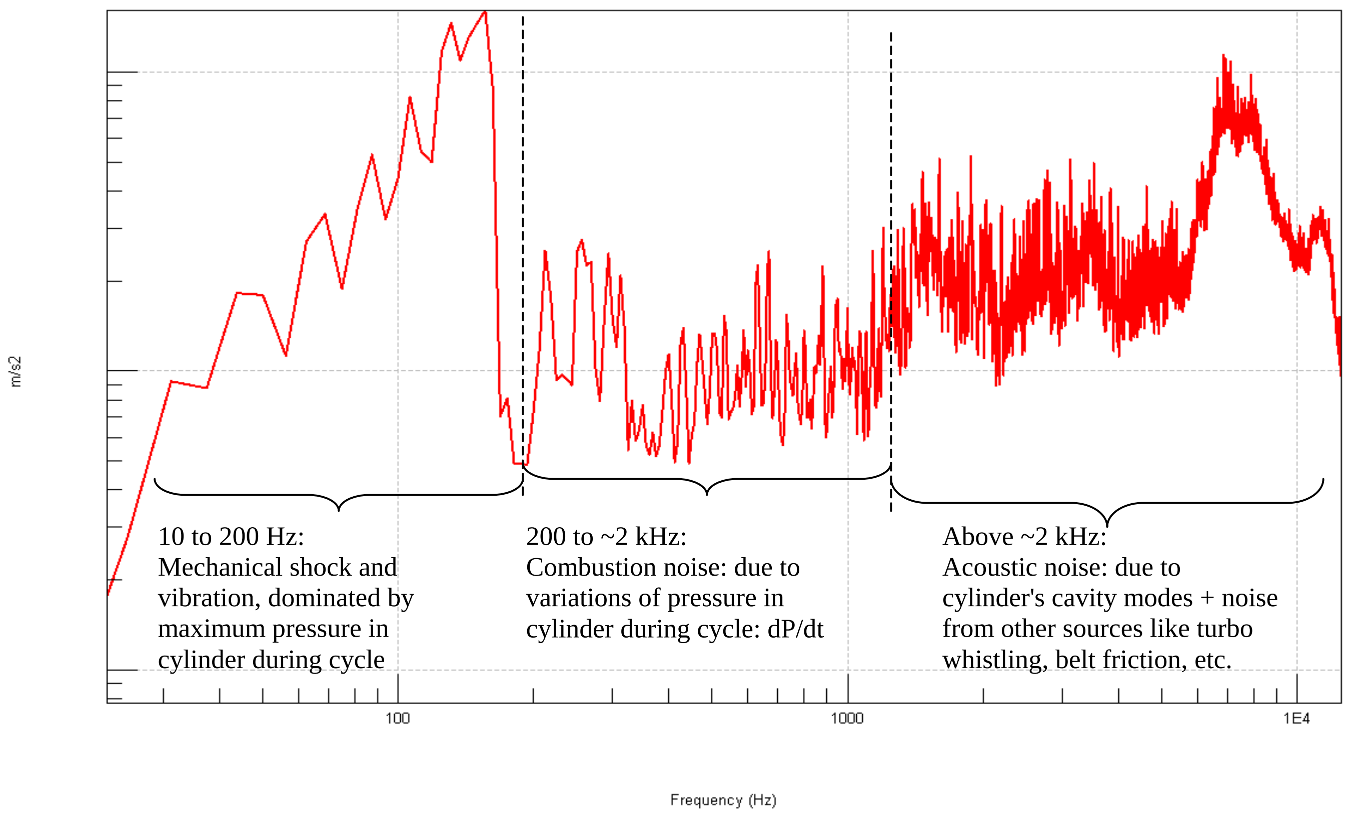

Figure 5 : Sources of noise in different frequency ranges

So, how important is this high-frequency energy for the component? Figure 6 below shows a Fatigue Damage Spectrum realised with parameters typical for Aluminium alloys (b=8, Q=10).

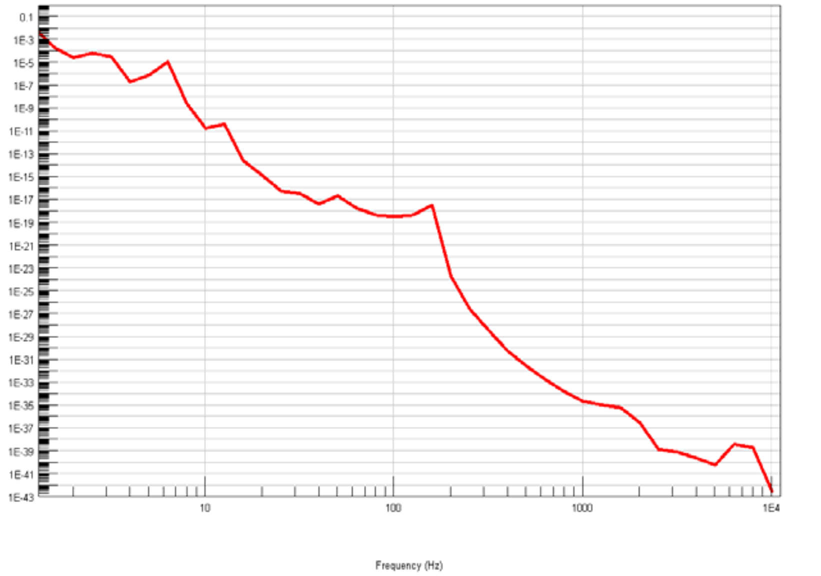

Figure 6: Fatigue Damage Spectrum showing negligible accumulation of damage above 2 kHz

It clearly shows negligible damage contribution by the frequencies above 2 kHz.

This is due to the fact that a constant amplitude acceleration is equivalent to a displacement with amplitude that decreases with the square of the frequency. Therefore, the higher the frequency, the smaller the displacement for constant acceleration. Knowing that fatigue is due to the accumulation of strain cycles in the material, and strain is proportional to displacement, very limited damage should occur at these very high frequencies.

A waterfall analysis shows that the 2nd order dominates largely the vibrations: Figure 7 below illustrates this.

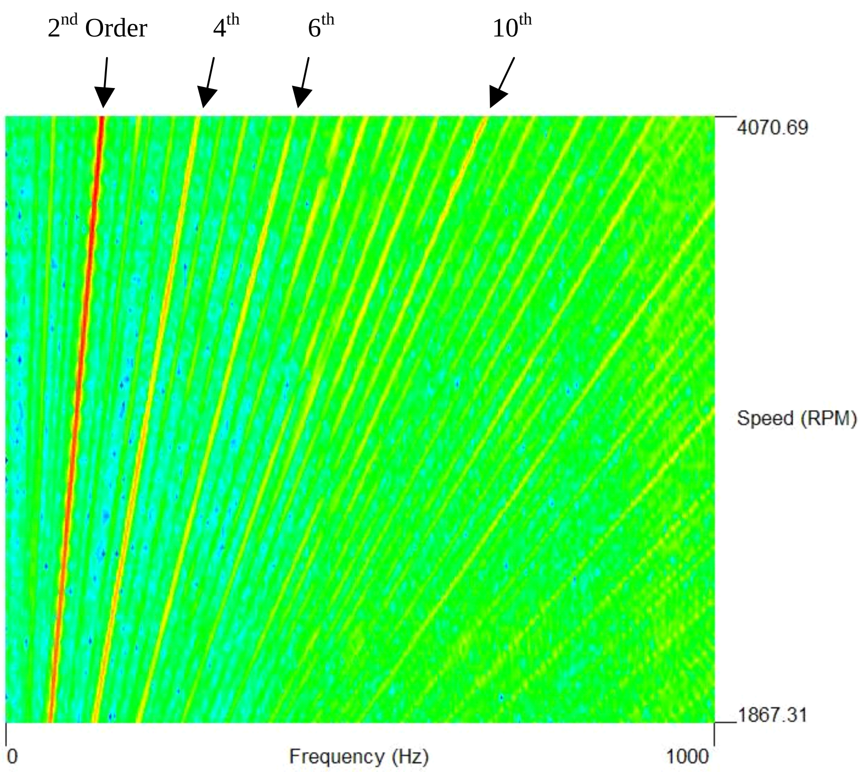

Figure 7 : Waterfall analysis with order lines

Above 2 kHz we saw that the contributions are rather due to engine combustion, cavity modes and are limited in amplitude (typically 2 orders of magnitude less than the energies within the frequencies where the 2nd order dominates). Moreover, the component is made of metallic materials and we know from simulation that its main mechanical resonant frequencies should not exceed ~500 Hz.

Our analysis will therefore be limited to the frequency range of DC to 2 kHz.

However, in the case of electronic circuits and components, very stiff and brittle, frequencies beyond 2 kHz should be considered and treated as they could excite resonant modes and therefore have a significant impact on their durability.

2.4. Calculation of FDS

The signals coming from the accelerometers close to the main fixing point were considered. The deterministic approach was used here because of the non-stationary nature of the signals.
The fatigue parameters are typical values for a metallic material:

- Basquin slope b = 8
- Dynamic amplification factor: Q = 10

Figure 8 below shows an example of FDS calculation using HBM's nCode GlyphWorks software [6].

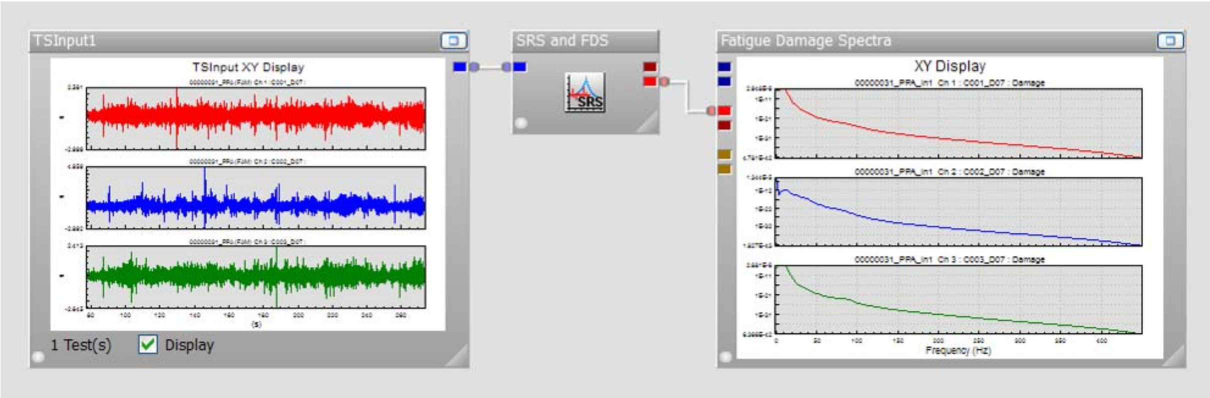

Figure 8 : FDS Calculation using GlyphWorks

2.5. Derivation of synthesised Spectra

This section discusses how to obtain an equivalent PSD or swept sine that is equivalent to the obtained combined FDS.

2.5.1. Fatigue Parameters

The same parameters are used as for the forward analysis (i.e. from time series to FDS). These are described in the previous chapter.

2.5.2. Combinations of FDS

The accelerometers close to the main fixing point are all positioned on a rigid part of the engine. Therefore the FDS obtained are enveloped.
Example: $FDS_{FixingPoint} = MAX \{FDS_{Pos1} ; FDS_{Pos2} ; FDS_{Pos3}\}$

2.5.3. Inversion to PSD or swept sine

The objective is now to find a PSD or swept sine that is equivalent to the obtained combined FDS. In other words, the FDS of the equivalent spectrum should match the combined FDS.
This equivalency is obtained using an iterative approach: we adapt the PSD until its FDS overlays with the target FDS, within a certain margin.

The synthesised equivalent PSDs are calculated to have equivalent damage in an exposure duration of 30 hours per axis. The synthesized equivalent swept sine is calculated with a sweep rate of 1 octave per minute, for 30 hours per axis, which represents 191 sweeps in total.

2.5.4. Safety and Test Factors

Any synthesized test specification should ideally include a safety factor on fatigue damage. First, a statistical safety factor can account for variations in applied loading and the fatigue strength of the component. Second, an additional test factor is used to account for the limited number of durability tests to actually be undertaken. The total safety factor employed is therefore taken as the product of these two safety factors.
The total coefficient applied on the PSD is: $K_{PSD} = 1.5$
For the sine sweeps, the total coefficient applied is $K_{SineSweep} = 1.25$

3. COMPARISON OF SYNTHESIZED VIBRATION SPECTRUM WITH GENERIC SPECIFICATIONS

The obtained synthesized specification can be compared with the generic spectra given by the car manufacturers.
These qualification spectra are often of different natures: they can be either random (PSD) or harmonic (swept sine) and have to be applied for different exposure durations e.g. typically 75 or 96 hours per axis.

We decided to compare the PSD and swept sine specifications separately, all spectra were scaled to 30 hours peraxis. Only the vertical axis is shown here for clarity.
The HBM's nCode GlyphWorks software was used in these comparisons, as illustrated in Figure 9 below:

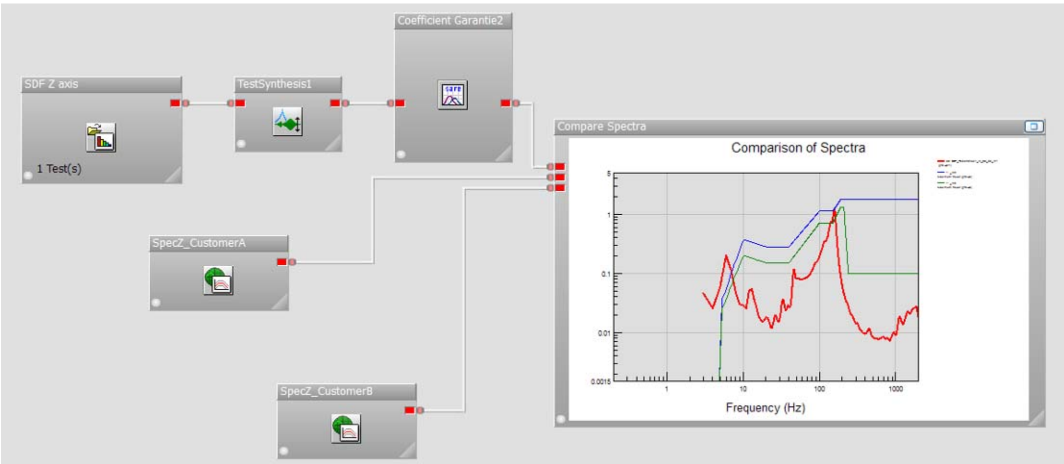

Figure 9: Process for Analysis and Comparison of resulting Spectra

In the Figure 9 above, the existing vibration specification is defined as a table of 2 columns {Frequency; PSD Value} for the PSD and {Frequency; Value} for the swept sine.
Now, let's have a look at the obtained synthesized PSD, compared with the existing random specifications coming from the car manufacturers: Figure 10 below overlays them.

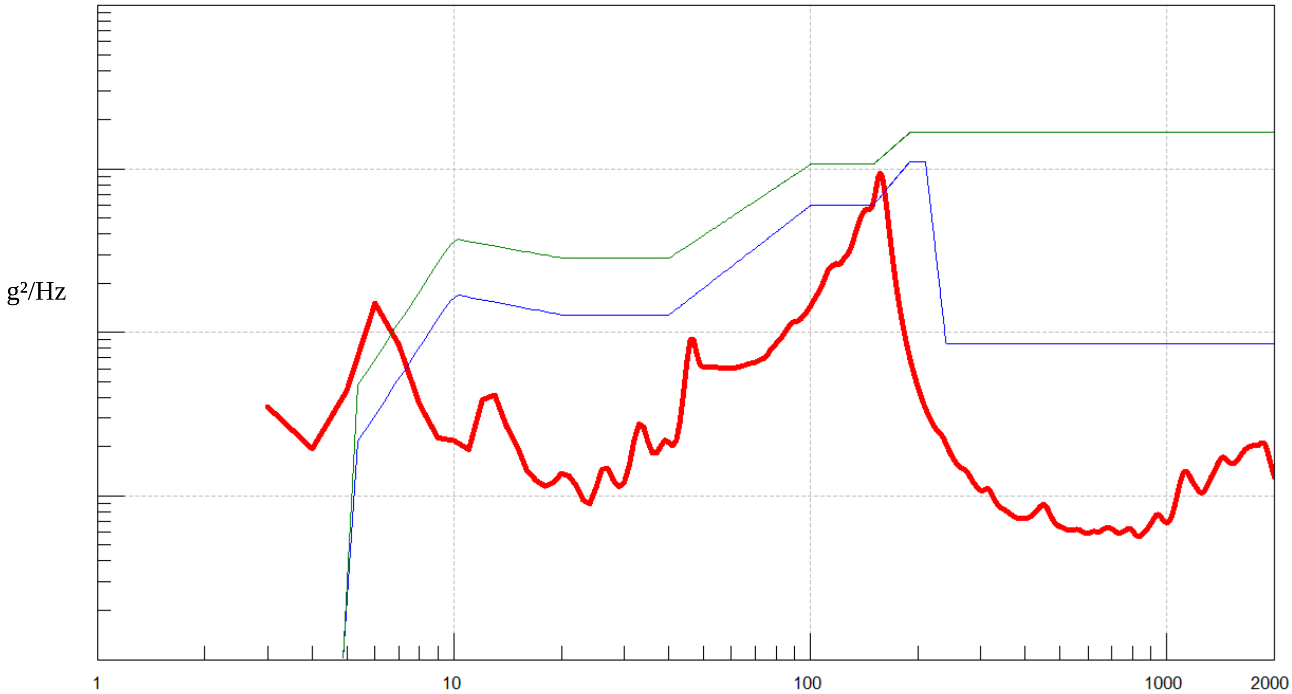

Figure 10 : Overlay of obtained PSD (red) vs. existing test specifications (blue and green)

The existing specifications seem to cover well the synthesized spectrum, especially at high frequencies (above 200 Hz). Note that at ~150 Hz the synthesized spectrum exceeds one of the existing specifications.

Similarly, Figure 11 below overlays the synthesized swept sine with the existing harmonic specifications.

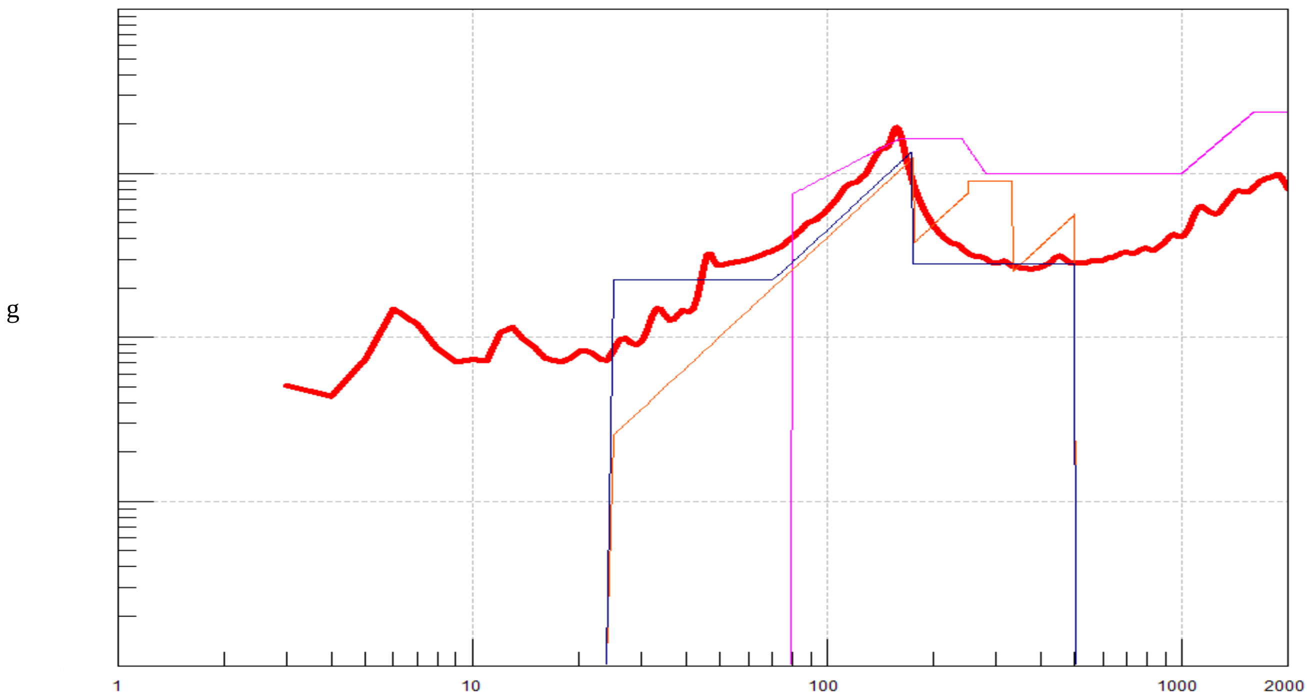

Figure 11 : Overlay of obtained swept sines (red) vs. existing test specifications (magenta, orange and dark blue)

We see here that the synthesized specifications may exceed the existing requirements under 200 Hz i.e. typically where the engine orders are important contributors to vibrations!

Note that some specifications only cover a frequency range up to 500 Hz, while others go up to 2000 Hz. It can be beneficial to sweep frequencies up to 2000 Hz, especially if component includes electronic equipment.

An interesting analysis would be to compare all existing specifications (Random and Harmonic) with the synthesized PSD in terms of potential damage. Figure 12 shows an overlay of Fatigue Damage Spectra, obtained considering either stochastic approach (for the PSDs) or the deterministic approach (for the swept sines). The same colour code was kept for clarity.

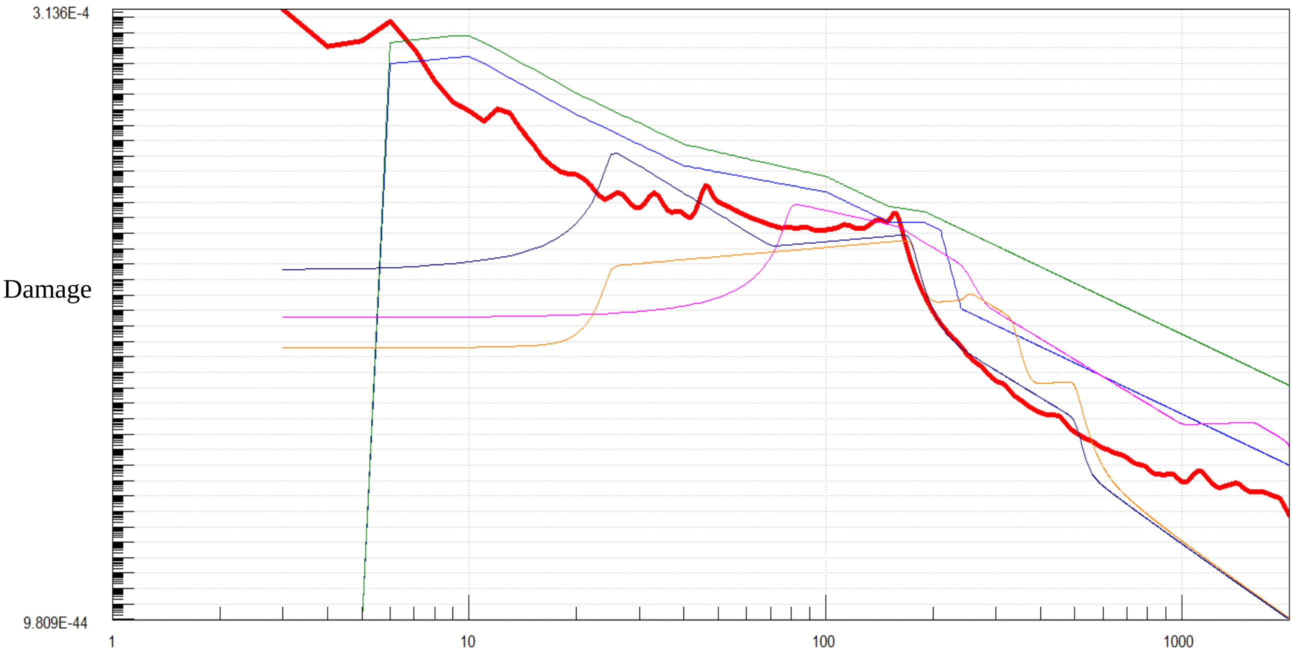

Figure 12 : Comparison of damage spectra between the synthesised PSD (red) and all existing specifications

We see that the existing specifications defined as PSDs (blue and green) are more severe than the swept sines (magenta, orange and dark blue). We find again that the synthesised specification exceeds the swept sines under 200 Hz.

4. CONCLUSION

This study has shown that it is possible to use measured vibration data and service usage data to create an equivalent PSD or a swept sine test specification. Further, this can be compared with the existing vibration specifications and requirements.

As a global result, we see that the synthesized specifications have higher vibration magnitudes than the existing vibration profile up to 200 Hz. The synthesized specification's magnitudes are lower than the existing profile in the high frequencies. In other words, the existing specifications may lead to overdesign at high frequencies and underdesign in the low frequency range (typically where the engine orders are important contributors to vibration).

Finally, the methodology using the Fatigue Damage Spectrum is also very interesting for comparing severities of existing vibration specifications or for finding iso-damage equivalencies between random and harmonic loadings.

5. BIBLIOGRAPHY

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