COMPDYN 2013
4th ECCOMAS Thematic Conference on
Computational Methods in Structural Dynamics and Earthquake Engineering
M. Papadrakakis, V. Papadopoulos, V. Plevris (eds.)
Kos Island, Greece, 12–14 June 2013

REDUCTION OF OVERTESTING DURING BASE-DRIVE RANDOM VIBRATION TESTS FOR THE EUCLID SPACECRAFT HARDWARE

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Keywords: Random vibration, Overtesting, Notching, Spacecraft Mechanical Loads, Euclid Project.

Abstract. *The purpose of random vibration testing is generally to verify strength and structural life by introducing random vibration through the mechanical interface. Typical applications are electrical components, instruments and small spacecraft for which acoustic testing is ineffective. The base-drive vibration test is conducted with the test item sitting on a moving platform that is driven by a shaker which introduces vibration typically up to 2000 Hz in three single-axis tests. The base-drive configuration is commonly employed to achieve test levels comparable to the launch environment. This conventional approach to testing has been known for decades to potentially overtest the test article at its own resonance frequencies. For this reason "notching" (i.e. reduction) of the specified input spectrum is often necessary.*

Notching on the input spectrum can be considered as far as it does not "jeopardise" the aim of the test, for example the qualification of the test specimen. As a consequence, one of the main issues for the structural engineer is to define a minimum threshold for the input notching. Special attention is paid to the "phenomenology" of secondary notching, which is related to level reduction on critical areas inside the test specimen. This paper provides some guidelines for assessing the notched profile in random vibration testing. In particular the importance and the benefits of performing a vibro-acoustic analysis of the complete spacecraft is illustrated with an example.

The paper is based on some author's contributions to the ECSS Spacecraft Mechanical Loads Analysis Handbook [1], and his experiences with the a number of European Space Agency projects. A critical investigation of the overall approach from the definition of the test specification to the quantification of the notching is shortly presented. The guidelines should be used in the frame of the Euclid spacecraft hardware verification when base-drive random vibration testing will be performed to show compliance to the random vibration and vibro-acoustic environment requirements.

1 INTRODUCTION

Some spacecraft load environments are treated as random phenomena, when the forces involved are controlled by non-deterministic parameters. Examples include high frequency engine thrust oscillation, aerodynamic buffeting of fairing, and sound pressure on the surfaces of the payload.

Random vibration analysis describes the forcing functions and the corresponding structural response statistically. It is generally assumed that the phasing of vibration at different frequencies is statistically uncorrelated. The amplitude of motion at each frequency is described by a power spectral density (PSD) function. In contrast to transient analysis which predicts time histories of response quantities, random vibration analysis generates the power spectral densities of these response quantities. From the power spectral density, the root mean square (rms) amplitude of the response quantity is calculated. The root-mean-square acceleration is the square root of the integral of the acceleration PSD over frequency. Random vibration limit loads are typically taken as the "3-sigma load" (obtained by multiplying the rms load by 3).

Random vibration testing helps demonstrate that space hardware can withstand the broadband high frequency vibration environment. The tests are conducted on an electrodynamic vibration machine or "shaker", which consists of a mounting table for the test item rigidly attached to a drive-coil armature. A control system energizes the shaker to the desired vibration level. Feedback for the control system is provided by a series of accelerometers, which are mounted at the base of the test item. Similarly to sine testing, adequate control approaches and strategies are used to avoid overtesting and to ensure realistic structural responses.

Heritage flight data, test data and analytical methods are used to predict vibration test levels. In most cases the predicted environments are verified later with system-level acoustic tests.

2 THE EUCLID PROJECT

Euclid is the ESA's mission to understand dark energy and dark matter within our universe. The spacecraft will start its journey in 2020 towards “L2”, the second Earth-Sun Lagrange point (Figure 1). It will be launched on a Soyuz rocket from Kourou.

The Euclid spacecraft is composed of a Service Module (SVM) and a Payload Module (PLM). The SVM includes the sunshield. The PLM consists of a large three mirror Korsch telescope feeding two instruments, the VIS visible imager and the NISP near-infrared spectrophotometer. The two instruments deliver Euclid's science, VIS for precise visible-light images of distant galaxies, and NISP for near-infrared spectro-photometry. They lie within Euclid's Payload Module (Figure 2) which provides mechanical and thermal interfaces to the instruments (radiating areas and heating lines). The VIS instrument is delivered in several separate units with dedicated mechanical and thermal interfaces with the payload module: the VIS focal plane assembly including proximity electronics, the readout shutter unit and the calibration unit. The NISP instrument is delivered as a standalone instrument. Both instruments have warm electronics located on the SVM to minimize thermal dissipation on the cold PLM.

The payload module development relies on Structural and Thermal Model (STM), built at flight standard, used to qualify the PLM with respect to the mechanical environment. After integration on the SVM STM, it is used to check the mechanical and thermal coupling, and thermal model predictions. A PLM Flight Model (FM) undergoes a full proto-qualification programme before delivery to Prime contractor.

The hardware of the Euclid spacecraft has to be qualified with respect to the random vibration and vibro-acoustic environments. For this reason a number of base-drive random vibration tests will be performed. The overtesting should be avoided or limited, without “jeopardizing” the objectives of the test.

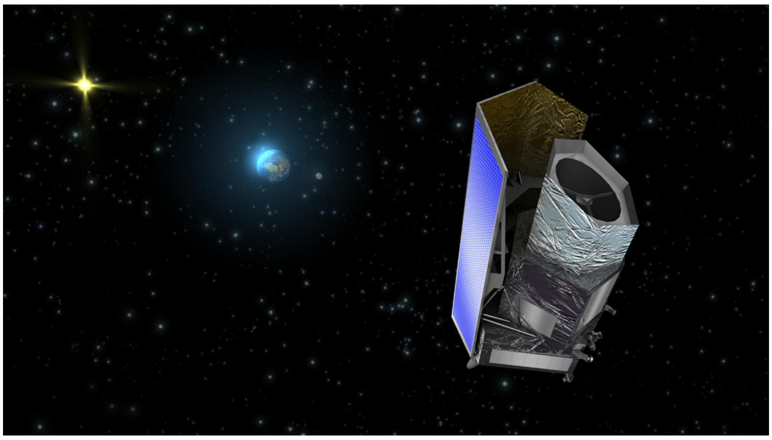

Figure 1: Euclid at L2 – Artist impression (source ESA)

Figure 2: VIS and NISP instruments integrated in the baseplate of the Euclid payload module (source Astrium)

3 CRITERIA FOR RANDOM VIBRATION LOADS AND ENVIRONMENTS

3.1 General aspects

The parameter most commonly used in the industry to define the motion of a mechanical system is the acceleration. The reason is mainly that accelerations are directly related to forces/stresses and “easy” to specify and measure. In practice accelerations are used as a measure of the severity of the mechanical environment. The loads are usually specified in terms of power spectral densities (usually of the acceleration), for the broad band random vibrations.

The main parameters are levels but also durations. The duration is a significant parameter for random vibrations. Of course the damage potential depends both on the load levels and duration, the latter being especially relevant for structural life verification.

3.2 Equivalence criteria

The subject of the equivalence criteria for dynamic environments, which are often crucial for establishing appropriate loads, is very complex, and normally the "equivalence" is limited and it can have some drawbacks. Since in general the equivalence has some limitations, enveloping techniques and conservativeness are often applied. In the following just some basic concepts about equivalence criteria are recalled. Relevant information can also be found in [2].

It should be noted that a very first "equivalence criterion" is implicit in the way the loads are defined. For example it is implicitly assumed that two random vibration environments are "equivalent" if they are represented by the same PSD of the input accelerations and duration, regardless the differences in the time histories. On the other hand, it is a common mistake to use the rms value of the input acceleration as a measure of its severity. The problem with the rms value is that it depends strongly on the values of the PSD at very high frequencies and on the upper frequency limit, which are often irrelevant. The most appropriate measure of the severity of a random vibration test is the PSD value at the resonant frequencies of the item. The maximum PSD value is important only if it is related to the main resonances.

Establishing an equivalence between different environments, or identifying which is the most severe, typically involves the evaluation and comparison of the (expected) structural responses. This is very important since on the basis of environment comparisons, decisions can be made, for example, on the status of compliance or on the most adequate tests which should be performed and included in the structural verification plan.

The equivalence criteria which allow establishing the equivalence between the base-drive random vibration environment with the vibro-acoustic environment are also very important in practical applications. The "equivalence" which can be established is indeed rather limited since the two environments are intrinsically different due to the different physical nature, i.e. purely mechanical and acoustic, of the excitation. In practice, for large spacecraft, it should be noted that the specified levels and duration for the acoustic noise test are, usually, highly conservative. On the basis of that assumption, the base-drive random vibration environments for lower levels of assembly (e.g. instruments and units) can be derived by enveloping the predicted response in terms of acceleration PSDs, when the spacecraft is loaded by the vibro-acoustic environment specified by the launcher authority for the acoustic noise test. For small and compact spacecraft, random tests can be more severe on some areas than the acoustic noise test.

In a number of situations, for example for primary or secondary notching definitions (see paragraph 4.1), it is crucial to compare the random vibration environment with the quasi-static loads (QSL) specified for the structural item. It is a common mistake, and unfortunately an usual industrial approach, to consider the "3-sigma" output acceleration as equivalent (or in any case comparable) to a quasi-static load. This is in general incorrect since:

- QSL are equivalent accelerations at the CoG, this is not usually the case for the random acceleration considered.
- The rms value of the acceleration depends strongly on the values of the PSD at very high frequencies and on the upper frequency limit, which are often irrelevant.

This can be illustrated by the following considerations.

The response displacement, as well as force and stress, spectral densities have the same "general shape" as the response acceleration spectral densities but their higher-frequency peaks are, in general, significantly lower in proportion to the lower-frequency peaks. In other words the first modes account for nearly all of the total displacement, force and stress. In mathematical terms it can be explained by noting that the displacement PSD is proportional to the acceleration PSD by a factor $1 / \omega^4$. In more "physical" terms, this effect can also be explained by considering that the fundamental modes, e.g. the ones that have the largest effective masses and that establish the "force-link", are usually at lower frequency. Finally a global argument is the following: QSL and random loads do not have the same distribution in the item, depending on the frequency content compared to the eigenmodes. In other words, QSL implies a "quasi-rigid" behaviour which is not far from the first global mode in each direction, but the quasi-rigid behaviour is quite different with respect to the other modes, i.e. upper global modes and local modes.

In industrial applications often two mechanical environments are considered equivalent if they have the same shock response spectrum (SRS) [1], or similarly, for random vibrations, the same random response spectrum (RRS) (also known as vibration response spectrum [3]). In this case the equivalence is established on the base of the single-degree-of-freedom (SDOF) response. This criterion is often applied to structure with base excitation.

Another criterion is based on the evaluation and comparison of the structural response to the mechanical environments. The comparison is often performed in terms of accelerations and interface forces. In this case the equivalence is established on the base of the response of the actual structure.

Both criteria are based on structural responses, either from the SDOF system or from the actual structure. The advantage of the SDOF system is that it is a simple and "standard" structure which can be used not only to characterize the environment but also to estimate the response of the actual structure, albeit with assumptions. The advantage of using the response of the actual structure is that the solution is "exact", but the calculations are more costly and the results are valid only for the given structure. The practice depends on the industrial context, schematically:

- If the structure is "unknown", the SRS approach should be applied. This approach is simple but with assumptions, so with some risk.
- If the structure is "known", it is recommended to compute its actual response. This approach is more costly but also more reliable.

As a conclusive remark on equivalent loads and environments, in general it is always dangerous to replace an environment by another of different type and relevant structural analyses of the item should be performed.

3.3 Verification criteria

The following criteria are normally applied for strength verification of structures exposed to random vibration environments [1]:

- Verification by analysis: margins of safety greater than or equal to zero.
- Verification by test (qualification by acoustic or random vibration testing): test environments are compared with random-vibration environments derived from system-level acoustic testing. In other words this is evidence that the applied random vibration test levels, at lower level of assembly, were high enough, including qualification margins.

3.4 Some inconsistencies of the loads verification process

Some "inconsistencies" and potential issues in the verification process of random vibration loads are here summarized:

• Uni-axial vibration test facilities while the dynamic environments for space vehicle hardware are typically multiple-axis. In practice, tests are performed axis by axis.

• Infinite mechanical impedance of the shaker and the standard practice of specifying the input acceleration as the frequency envelope of the flight interface acceleration (despite the presence of antiresonances or dips in the flight configuration). This is the major cause of over testing in aerospace vibration tests.

• Vibro-acoustic environment often simulated at the subsystem and units assembly level using a random vibration test.

• Test levels largely based on computational analyses. For this reason it is important to validate critical loads analyses.

4 NOTCHING IN RANDOM VIBRATION TESTING

4.1 General aspects

Notching is the reduction of acceleration input levels around resonant frequencies, to avoid over testing. "Notching" can be distinguished in "primary notching" and "secondary notching". The primary notching is performed to limit the shaker-test specimen interface forces to the target values, normally qualification or acceptance loads. This is basically the same as to limit the equivalent accelerations to the centre of gravity (CoG) of the test item. The secondary notching is performed to limit local accelerations inside the test item with the purpose of protecting equipment, instruments or sub-systems.

Primary notching in random vibration testing of space hardware is mainly justified by the fact that the real environment in flight is simulated on shaker by an acceleration PSD based on an envelope of the interface levels foreseen in the considered frequency band, typically from 20 to 2000 Hz. This envelope doesn't account for the possible reactions of the test item which can produce level reductions in some frequency bands.

These potential level reductions (with respect to a rigid test item) are due to a high test item dynamic mass at the interface (with the "flight" mounting structure) which reduces the effect of the exciting forces according to the Newton's law. This high dynamic mass is generated by eigenmodes with high effective masses with respect to the interface. As the primary notching is related to interface forces, the unique criterion for mode selection in this case should be based on the modal effective masses [1].

The secondary notching is related to level reduction on critical areas inside the test item. In this case, the frequency response function involved is the test item dynamic transmissibility between the considered area and the test item interface, and the unique criterion for mode selection should be based on the modal effective transmissibilities [1].

4.2 Basic principles

Notching can be considered when it can be demonstrated that an "unreasonable" over testing with respect to the target loads (e.g. qualification loads) occurs if an adequate reduction of the input spectrum in some frequency bands is not applied.

In this context the term "unreasonable" is important. In fact by using random vibration tests the risk of overtesting (as well as undertesting) with respect to the maximum expected flight environment (including margin) cannot be completely removed. This is due to differences between "flight" and "test" conditions which can hardly be removed (e.g. base drive random test used to simulate vibro-acoustic environment, multi-axis excitations vs. single axis test facilities, differences in boundary conditions).

In order to minimize the risk of under testing, and then of mission failure, the general recommendation of minimizing the practice of notching can often be imposed by explicit (or "implicit") requirements, for example by the launcher authority or by the customer. On the other hand it should be noted that, especially in the case of flight hardware, it is normally a "common interest" to limit as much as possible any overtesting and unjustified risk of structural failures.

Other important aspects are the following. The need for primary notching is generally "well understood", i.e. it is due to the specific dynamic behaviour of the test specimen on the shaker produced by the modes with large effective masses. In addition the frequency bands involved are normally narrows and relatively easy to localize. The agreement on the depth and width of the notches is usually rather straightforward.

More complex is the scenario for the definition of the secondary notching. In fact the secondary notching is, in its own essence, a revision of the applicable mechanical environment; it means a change of the applicable test specifications. The arguments are then not only technical, but also contractual, since they involve the sharing of responsibility and risk between customer and contractor. However, within a cooperative scenario, the crucial question remains technical: is the test specimen, or part of it, going to suffer an unreasonable overtesting, if the notching is not applied?

In practice secondary notching opens the door to a systematic reassessment of the test specifications by comparing the predicted dynamic response in the test with the expected "flight" mechanical environment. This is potentially in conflict with the logic of producing the test specifications by "enveloping" the expected mechanical environment at the interface, which has demonstrated during the years to be a robust approach. Moreover the systematic reassessment of the mechanical loads induced by the possibility of performing secondary notching generally induces inefficiencies and complexities in the process of mechanical analyses. This explains why it is often recommended to limit the practice of secondary notching to the critical items which indeed need to be protected.

In short, the notching on the input spectrum can be considered as far as it does not "jeopardise" the aim of the test, for example the qualification of the test specimen. However the application of the basic principles is not straightforward. Adequate approaches and criteria are crucial and some of them are hereafter reported.

4.3 Response and force limiting

To alleviate the overtesting problem and to define the notched spectra, two basic approaches are used:

- methods based on measurement of accelerations, also referred as "response limiting"
- methods based on measurement of forces, also referred as "force limiting"

Response-limiting consists of analytically predicting, usually through coupled loads analysis, the in-flight response at critical locations on the test article, measuring these responses during the test, and reducing or notching the input acceleration at the critical resonance frequencies so that measured responses do not exceed the predicted limits.

The main problems or disadvantages with the response limiting approach are reported for example in [1].

In the force limited vibration approach, response-limiting is replaced by limiting the reaction force at the interface between the test article and the shaker. The force limited vibration notching approach has a number of advantages [1].

In the industrial practice often a hybrid approach which includes both response and force-limiting is implemented.

4.4 Criteria for notching justification

Notching should be accepted when:

- Loads coming from the coupled system are demonstrated (with margins)
- Target loads of the tests would be exceeded

In practice for random vibration tests the assessment is based on the following approach. The random test analytical predictions (and during the test campaign the expected structural response based on lower level test runs) are compared to the results of the system-level vibro-acoustic analysis. Typically the comparison is performed in terms of PSDs of the interface forces, interface accelerations and accelerations at critical locations. Unfortunately reliable and detailed analyses are not always available.

The general criterion of not exceeding the QSL during the vibration test is generally accepted, however the accurate evaluation of the equivalent acceleration at CoG can be a difficult task, especially for secondary notching.

Specific criteria which are generally accepted are hereafter reported.

For primary notching in random vibration test:

- Criterion 1: Primary notching based on measured interface forces can be considered when they are predicted to be higher than the target loads.
- Criterion 2: Primary notching based on measured accelerations can be considered when the adequacy of the I/F forces estimation method can be proved.
- Criterion 3: Width and depth of the notches in the relevant frequency bands should be identified with the goal of minimizing the risk of under testing.

For secondary notching in random vibration test:

- Criterion 1: Secondary notching in random vibration tests can be considered when it can be demonstrated that the severity of the structural response of the test item during the random vibration test is expected to be higher than the acoustic response which is intended to simulate (e.g. satellite acoustic noise qualification test). It means that measured accelerations PSDs in critical locations are expected to be higher than the predicted acceleration PSDs in the system-level vibro-acoustic analysis.
- Criterion 2: Secondary notching in random vibration test should be minimized in order to reduce the risk of under testing within the concerned frequency bands. It means that moderate over testing is acceptable if positive margins can be shown. It should be noted that local dynamic response higher than the “internal item” QSL does not necessarily mean that the QSL (i.e. equivalent CoG accelerations) are exceeded.

4.5 Practical aspects of notching and verification of compliance in random vibration

Qualification of structures in the random environment raises several specific problems such as difficulties in numerical analysis for test prediction due to the extended frequency range, and the choice between mechanical base-excitation and acoustic excitation.

The evaluation (both by structural analysis and test measurements) of the CoG net acceleration of the component is properly performed by means of the external forces. For example, for a pure translation, by Newton’s second law, the CoG net acceleration is simply equal to the measured external force divided by the total mass. Attempts to measure the CoG acceleration with an accelerometer usually overestimate the CoG response at resonances, so limiting these measurements to the CoG criterion (notching in a vibration test) results in an undertest.

The signal processing bandwidth is a common source of misunderstanding in random vibration testing [1]. In the space industry today, an enormous number of samples of random vibration data are taken with many different measurement techniques and processed in many different ways. In particular with the popularity of digital signal processing, the frequency bandwidth can be very narrow and PSD curves often show very high and sharp peaks. These phenomena sometimes create "problems", specifically when comparing PSD curves. Two typical situations are:

- when controlling, within specified tolerances, the test environment at the base of the item with respect to the specified ("target") PSD;
- when processing data from system level acoustic test to arrive at PSDs for comparison with those used to test components.

These "problems" in some cases can be irrelevant since created by the high resolution processing. To be consistent the processing resolution in the vibration test should match the one used to establish test conditions or the processing resolution of the test data should be converted with that used to derive the specified environment.

In case the acoustic test measurement peaks exceed the specification, before taking a decision about retesting the item, the following approaches are used:

- Peak clipping. A commonly used rule is that all narrowband spectral peaks should be clipped by 3 dB [2]. Sharp peaks or peaks with a relatively small bandwidth of exceeding the unit's power spectral density specification might be clipped if the exceedance is not more than 3dB. This approach is based on the following justification. The acoustic test measurements are usually much lower than the unit's power spectral density specification next to the peak. Then it can be assumed that in the relevant frequency band the random input (usually flat for a wide frequency band at left and right of the peak found) is at least equivalent to the corresponding energy input resulting from the higher peak but taking into account the significantly smaller random levels next to the peak (see [1] for the analytical discussion of the approach). However perhaps the best approach "to remove" the narrowband spectral peaks is to compute all spectra with a resolution bandwidth that is proportional to frequency (e.g., a 1/6 octave bandwidth), and then envelope all peaks without clipping [2].
- Application of random response spectrum (RRS). For example [3] shows how the RRS of a typical power spectral density specification is compared with the actual peaky measurements and how it can be then shown that the specification covers the measurements even if there might be large exceedances. An approximate evaluation of the RRS can be performed by applying the Miles' equation with varying the natural frequency from 20 to 2000 Hz, which is the typical random vibration frequency range.
- There could be the chance to compare the unit's internal responses measured during the system acoustic noise test with the ones measured during the random vibration base-drive test at unit's level. In this case the internal responses should be checked first. In fact the incompatibility of the unit's random test excitation with the corresponding acoustic test measurement at the interface of the unit does not exclude a potential compatibility of the respective internal responses, which might depend much on the mounting conditions of the unit on the satellite as compared to the shaker test. In practice, if measurements of unit's internal responses are available from the acoustic test then these responses should be taken into account and, where relevant, the proposed approaches should be adapted accordingly to the comparison of the internal responses. Of course the aim is to verify that the unit's random vibration test has been performed with sufficient excitation alt-

hough an incompatibility with respect to the acoustic test measurements at the unit interface was noted.

However it should be noted that at the time of taking decisions on the adequacy of the notching , the results of the system level acoustic test are normally not available. On the other hand, recent developments in the area of computational mechanics allow performing vibro-acoustic response analysis of a complete spacecraft. The analysis can be performed by combining the FE method and the BE method with SEA approach, and allows the random levels on units and instruments to be compared to technical specifications or qualification levels. In particular an important advantage of performing a vibro-acoustic analysis at spacecraft level is that the results can be compared to the ones from a base drive random vibration test predictions, for example of an instrument, and to assess if some adjustments of the levels (e.g. secondary notching) are possible, for example before the qualification run.

5 EXAMPLE: SLSTR

5.1 General aspects

In the following some data concerning the qualification test campaign of the structural and thermal model (STM) of the Sea Land Surface Temperature Radiometer (SLSTR) are reported. SLSTR (see Figure 3) is one of the instruments of the ESA satellite Sentinel 3. The test article was instrumented with 21 load cells which allowed to recover the interface forces.

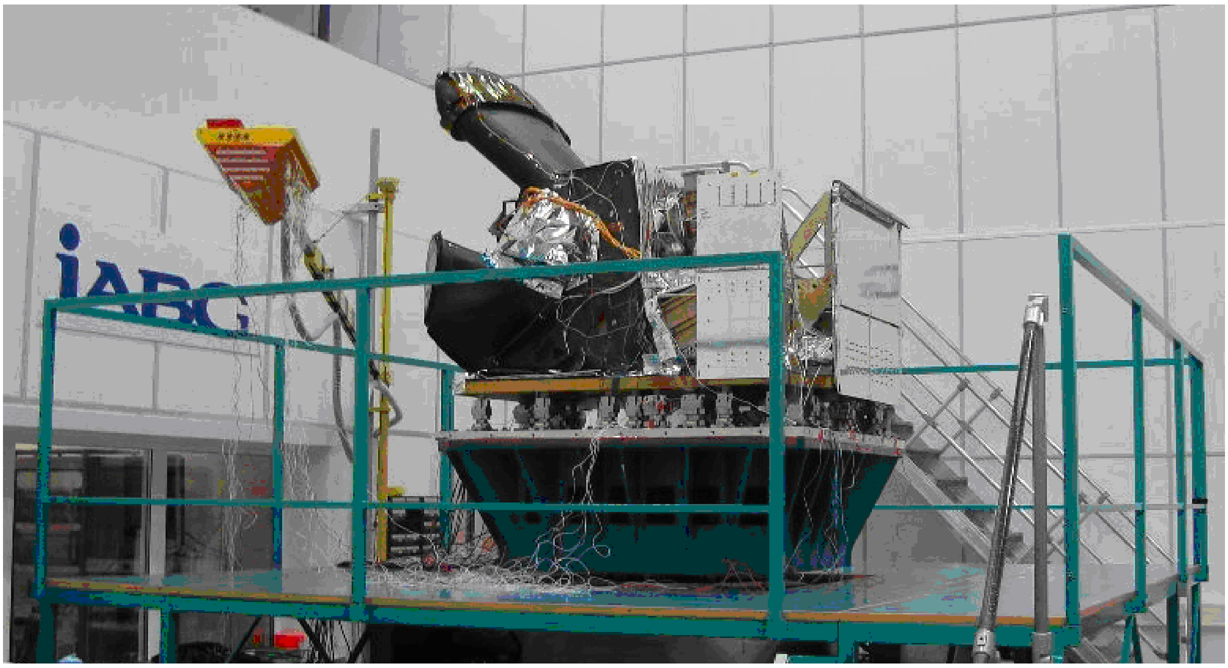

Figure 3: SLSTR STM on the shaker – X axis configuration.

Before starting the test campaign a finite element analysis of the base-drive random vibration was performed to predict the dynamic response of the test item on the shaker and, specifically, to predict if notching of the input spectra would have been necessary. An overview of the finite element model is shown in Figure 4.

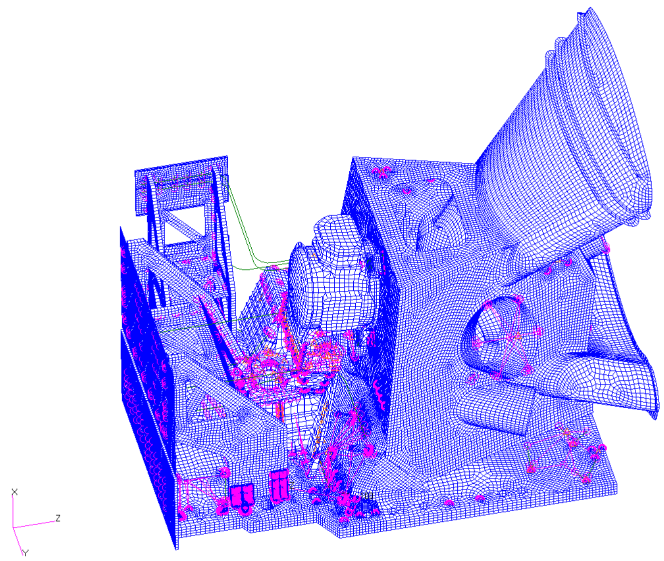

Figure 4: SLSTR STM Finite Element Model (source: SLSTR Industrial Consortium)

The results of the vibro-acoustic analysis at spacecraft level were also available at the time of the test campaign and the structural response was recovered at a number of critical locations of the instrument. In practice a number of comparisons have been possible since during the test the structural response was measured at relevant locations. In particular, the following criteria have been investigated before applying the full qualification levels:

- The interface load levels (expected to be) reached during the qualification test should cover with margins the ones predicted by the vibro-acoustic analysis (acceleration PSDs and forces if available).
- The test item "internal accelerations" (expected to be) reached during the qualification test should cover with margins the ones predicted by the vibro-acoustic analysis, both in terms of PSD levels and, consequently, root mean square.

Figure 5 shows the comparison between the initial test specification level in lateral direction Z (green profile) and the nominal spectrum applied during the test (blue profile). It should be noted that both secondary notchings and a general reduction of the vibration levels have been applied.

Figure 6 reports the nominal qualification levels in Z direction (red profile: "pilot") compared to the levels predicted by the vibro-acoustic analysis at the interface between the instrument and the spacecraft. The comparison is performed in terms of acceleration PSDs at relevant recovery points of the mathematical models, i.e. some restitution nodes distributed around the interface of the SLSTR. It can be concluded that the applied levels should be adequate to qualify the structure with respect to the vibro-acoustic flight environment.

Figure 7 shows an example of structural response in a critical location of the instrument. The extrapolated response at qualification level for the test run in Y direction is compared to the response computed by vibro-acoustic analysis. The root mean square value expected during the test is $5.7g_{rms}$ versus a predicted value equal to $2.2g_{rms}$.

In conclusion the final levels have been agreed mainly on the basis of the results of the vibro-acoustic analysis of the complete spacecraft and the expected structural response at qualification level in the random vibration test run. The latter has been extrapolated on the basis of the structural response measured during the test run at intermediate level.

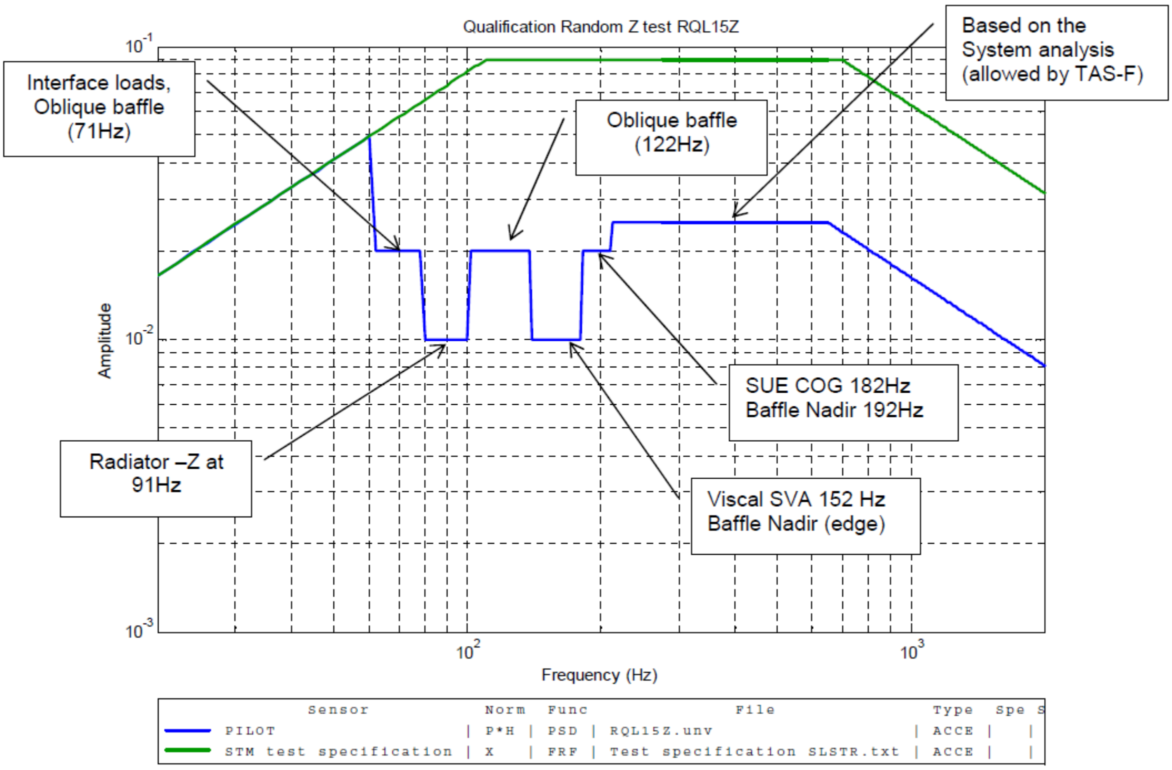

Figure 5: Random vibration test in Z: nominal input profile compared to specification (source TAS-F)

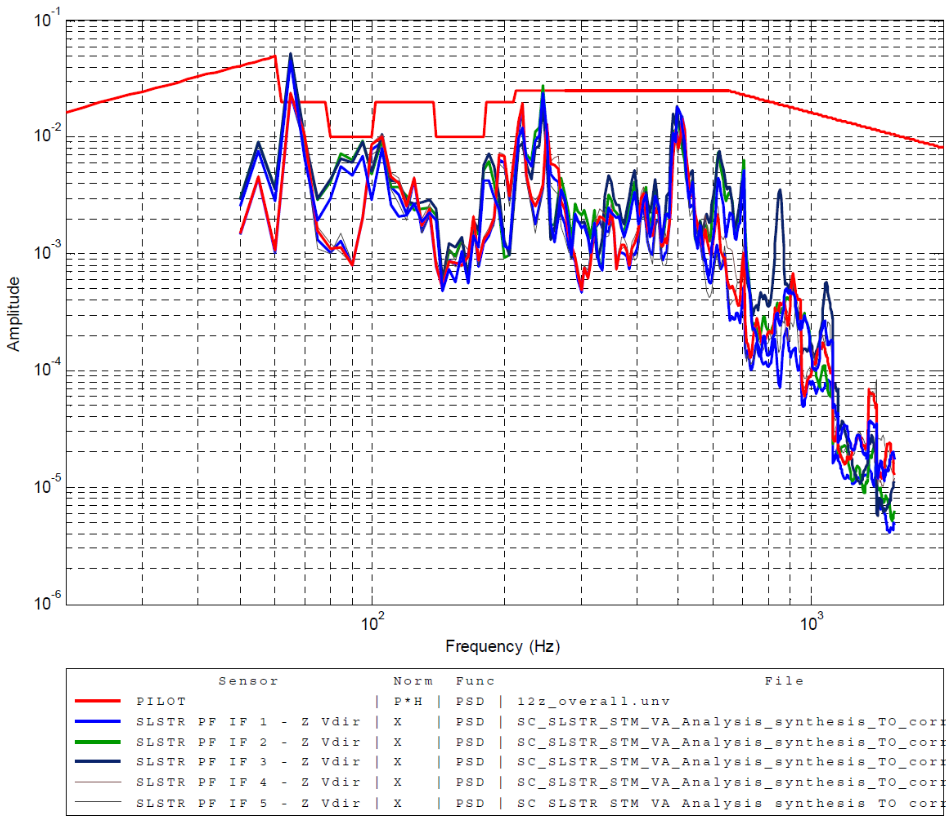

Figure 6: Nominal qualification levels (Z direction) compared to PSD levels calculated by vibroacoustic analysis (source TAS-F)

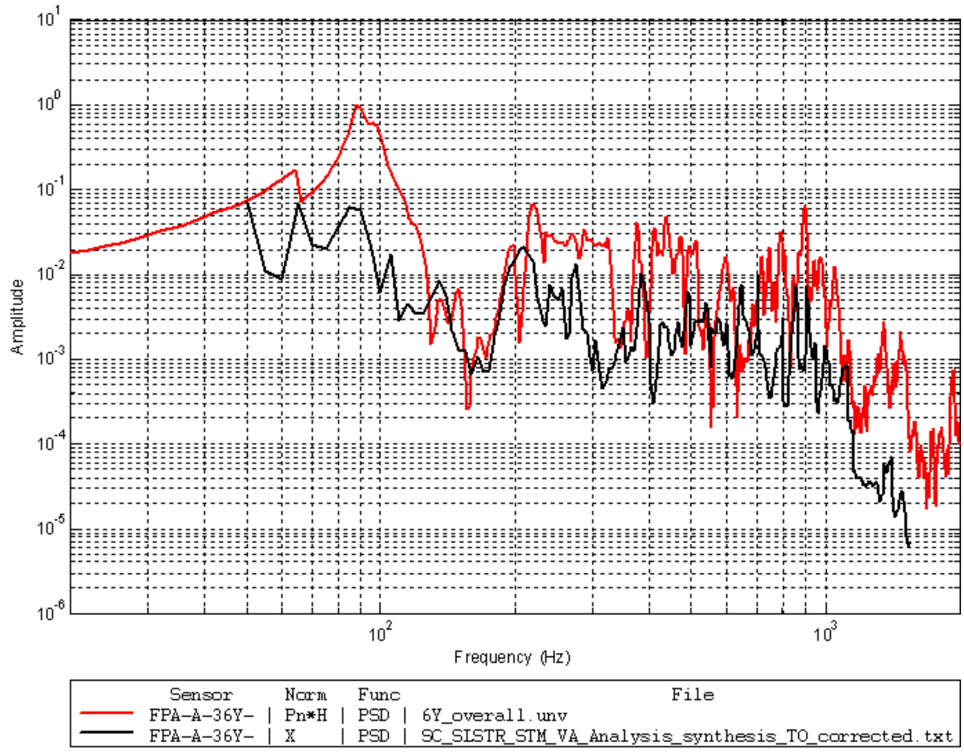

Figure 7: Location FPA-A-36Y: expected response at qualification level run in Y direction - red curve - vs. computed response by vibroacoustic analysis - black curve (source TAS-F)

5.2 Additional remarks

It should be pointed out that the calculated random response coming from the vibro-acoustic analysis (acoustic excitation of the spacecraft mathematical model which includes the one of the instrument), has been compared to the response from a base-drive random vibration test on a "rigid" interface (assuming an infinite mechanical impedance of the shaker).

The response at the instrument interface recovered by the vibro-acoustic analysis can include the contribution of modes of the interface not necessarily related to a significant response inside the instrument. So, trying to reach the same interface levels with the test item mounted on a rigid interface (i.e. the shaker), with a different transmissibility, can increase the risk of over-testing the subsystems.

In principle a conservative approach should be to apply test levels covering the ones predicted at the instrument/spacecraft interface recovered from the vibro-acoustic analysis. This is often the approach used for non-flying hardware (e.g. STM), where often the levels to apply are "as high as possible". The reason is to keep some margins which can cover the uncertainties related to the (non-validated) mathematical models, in particular of the test item. In fact that uncertainties can also affect the predictions obtained by the vibro-acoustic analysis.

In some cases this conservative approach could be not feasible without the risk to damage something and it is perhaps also not necessary. In fact even if the test levels applied at the interface of the item are apparently somewhat low in some frequency bands, the response (PSD) obtained on a critical location could be globally much higher than the one coming from the vibro-acoustic analysis.

6 CONCLUSIONS

The purpose of random vibration testing is generally to verify strength and structural life by introducing random vibration through the mechanical interface. The base-drive configuration is commonly employed to achieve test levels comparable to the launch environment. This conventional approach to testing has been known for decades to potentially overtest the test article at its own resonance frequencies. For this reason "notching" of the input spectrum is often necessary.

In this paper a critical investigation of the overall approach from the definition of the test specification to the criteria to justify the notching has been presented. Special attention has been paid to the "phenomenology" of secondary notching, which is related to level reduction on critical areas inside the test item. In this case, the frequency response function involved in the notching quantification is the "test item" dynamic transmissibility between the considered area and the test item interface. Another important aspect of the secondary notching is that it should be quantified based on the predicted equivalent CoG acceleration of the specific "internal item" which needs protection. However the mentioned CoG acceleration is difficult to evaluate, mainly because the relevant interface force is normally not available (i.e. not measured) during the test.

On the other hand, recent developments in the area of computational mechanics allow performing vibro-acoustic response analysis of a complete spacecraft. The analysis allows the random levels on units and instruments to be compared to technical specifications or qualification levels. In particular an important advantage of performing the vibro-acoustic analysis is that the results can be compared to the responses from base drive random vibration test predictions (calculated by analysis or by extrapolation of measured levels) to assess if some adjustments of the levels (e.g. secondary notching) are possible.

An example which illustrates the importance and the benefits of performing a vibro-acoustic response analysis at spacecraft level has been reported. Of course a crucial aspect of the approach is that, since the assessments are largely based on computational analyses, it is fundamental to validate the relevant mathematical models.

The presented guidelines should be used in the frame of the Euclid spacecraft hardware verification when base-drive random vibration testing will be performed to show compliance to the random vibration and vibro-acoustic environment requirements.

Acknowledgements

The author would like to thank the colleagues from Industry and the companies that have been involved in the design and verification of SLSTR STM. The companies are: Thales-Alenia Space France (F), Selex Galileo (I), Jena-Optronik (D), CarboFibretec (D) and Studiel (F). The test was performed at IABG (D) premises in November 2011.

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