HERSCHEL SVM STM VIBRO-ACOUSTIC TEST / PREDICTION COMPARISON

L. Trittoni, P.C. Marucchi-Chierro
Thales Alenia Space Italia
Strada Antica di Collegno 253, 10146 Turin
Italy

ABSTRACT

At early 2006, Herschel STM vibro-acoustic qualification test was successfully executed in ESA-ESTEC test centre, with Ariane 5 launch vibro-acoustic environment application.
One test objective was the validation of subsystem / unit specified random qualification spectra, which for HERSCHEL SVM zones were defined by FEM random vibro-acoustic response analysis.
An original simplified model of the launch acoustic field was built up and applied to the structural SVM FEM, based on a modal approach which takes into account the spatial correlation between impinging pressure and structural modes in the relevant frequency domain.
A very good agreement between test and analysis results can be highlighted.

1. INTRODUCTION

During the launch phase, spacecraft structures are submitted to heavy acoustic excitation mainly generated by the engines and the aerodynamic forces over the vehicle. The acoustic excitation induces a vibration environment that shall be characterized to correctly design the secondary structure and equipment.
Two main analytical approaches are currently available for the analysis of items with a large area-to-mass ratio (plates and shell structures) submitted to the excitation of random sound waves:

- a deterministic approach that uses a mode-by-mode basis (R[1])
- a statistical approach (Statistical Energy Analysis) that considers many, closely spaced modes to be active (R[2])

The deterministic approach is suitable to the low frequency range, where the low modal density permits to identify the structural modes with sufficient accuracy, while the Statistical Energy Analysis (SEA) is most applicable in the high frequency range.
The purpose of the method reported in R[1] is to ***extend*** the ***Miles formula*** (which computes the random response of rectangular flat simply supported homogeneous panels subjected to broadband acoustic load with infinite acoustic wavelength, considering only the fundamental resonance) to ***several resonance frequencies***, and ***finite acoustic wavelengths***.
In other words, while the Miles formula is applicable to single-degree-of-freedom systems (1 mass, 1 spring, 1 resonance frequency), the modal approach is applicable to multi-degree-of-freedom systems since it takes into account the ***analytical mode shapes*** relevant to modes higher than the first.

A further generalisation of the modal approach to face ***complex structures with complex constraint*** has been achieved by TAS-I, taking into account the ***mode shapes computed by FEM models*** and relevant to the *n* resonance frequencies included in a defined frequency range of interest.
The activity described in this paper concerns the application of the deterministic approach to the ***vibro-acoustic analysis of spacecraft structures*** in the low frequency range, performed ***with the FEM technique.***

The theoretical aspects involved in launch acoustic field simulation have been deeply investigated.
The acoustic pressure field has been simulated as a set of travelling plane waves (as suggested in R[1]) and the joint acceptance function, which is introduced as scaling function to take into account the coupling between acoustic waves and structural waves, has been evaluated for rectangular flat homogeneous panels with different boundary conditions, using the mode shapes computed by FE normal mode analysis. A very good agreement with theoretical results reported in R[1] and R[3] has been found.
Then, the approach has been conservatively simplified and applied for the first time on a satellite Service Module (SVM) in the frame of Herschel/Planck project, with the aim to define a detailed random vibration environment for various identified zones. This activity has constituted the basis for the derivation of the qualification / acceptance random vibration levels and design loads to be applied to HERSCHEL / PLANCK SVM equipment units.
A comparison of HERSCHEL STM vibro-acoustic qualification test results and vibro-acoustic analysis results has been performed for each SVM zone and a good agreement can be highlighted.

2. VIBRO-ACOUSTIC ENVIRONMENT

The following acoustic levels have been used for the analysis, derived from specified Ariane 5 flight levels +4dB (R[4]).

Octave Band Central Frequency [Hz]	SPL Ref. 2xE-5 Pa [dB]	SPL Ref. 2xE-5 Pa [dB]
31.5	128	132
63	130	134
125	135	139
250	139	143
500	134	138
1000	128	132
2000	124	128
OASPL	**142**	**146**

FIG 2.1 HERSCHEL specified acoustic levels (R[4])

3. HERSCHEL CONFIGURATION DESCRIPTION

A sketch of HERSCHEL SVM FEM with PLM represented by a lumped mass of 2400 kg is reported in FIG 3.0-1, while the satellite internal configuration is visible in FIG 3.0-2.

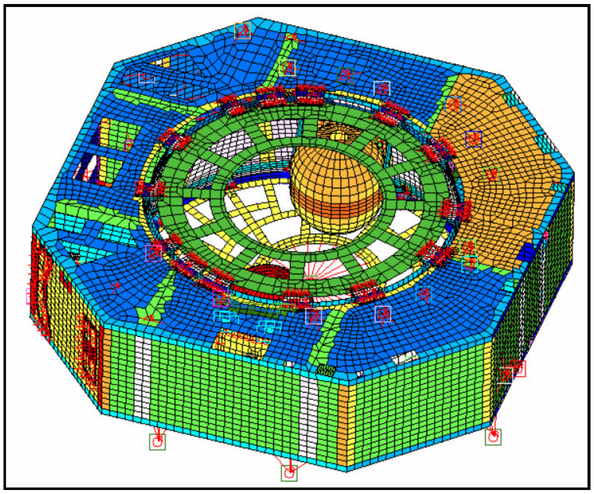


FIG 3.0-1 HERSCHEL FEM

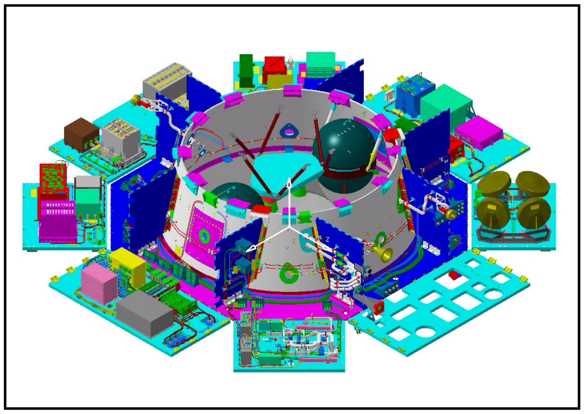


FIG 3.0-2 HERSCHEL internal configuration

The mass distribution relevant to SVM is reported in TAB 3.0-1.

Mass summary	Max Mass [kg]
ACMS	70.8
CDMS	16.0
CFE	3.8
CRYO	10.9
HIFI	120.1
HRN	101.8
PACS	61.7
PCS	37.2
RCS	57.6
SPIRE	43.3
STRUCTURE	275.4
STR ASSY	22.6
TCS	25.4
TT&C	27.7
Total	**874.2**

TAB 3.0-1 SVM mass budget

The total mass of HERSCHEL FEM was therefore 3274.2 Kg.

4. VIBRO-ACOUSTIC ANALYSIS METHODOLOGY AND COMPUTATIONAL APPROACH

The starting point was the method described in R[1], which was developed for the analysis of flat and moderately curved rectangular panels under broadband acoustic excitation and is based on an analytical formulation.
This method requires the simulation of the acoustic pressure field as a set of travelling plane waves and the evaluation of the joint acceptance function, which is introduced as scaling function to take into account the coupling between acoustic waves and structural waves.
The modal approach is applied to both the structure and the acoustic load. In fact, under some general conditions, the solution of the equation of motion of a structure can be expressed as a sum over the modes, with time and spatial dependencies separated mode by mode:

(1) $$D(x,y,z,t)=\sum_i \tilde{w}_i(x,y,z)w(t)$$

where D(x,y,z,t) is the physical displacement,
$w_i(t)$ are only functions of time and
$\tilde{w}(x,y,z)$ are the mode shapes functions only of space.

It is assumed that also the pressure field can be decomposed into a temporal and a spatial part in a manner similar to the structural modal decomposition:

(2) $$P(x,y,z,t)=\sum_i \tilde{p}_i(x,y,z)p_i(t)$$

where $p_i(t)$ represents the range of frequencies in the neighbourhood of the i_{th} mode frequency.
With the previous assumptions, structural responses are computed in the proximity of each modal resonance, as solutions of the following equation:

(3) $$\frac{1}{\omega_i^2}\ddot{w}_i(t)+\frac{2\zeta_i}{\omega_i}\dot{w}_i(t)+w_i(t)=J_i p_i(t)$$

where ω is the circular frequency, ζ is the damping ratio and ***J*** is the **joint acceptance**, constant for each mode and given by:

(4) $$J_i=\frac{\int_D \tilde{p}_i(x,y,z)\tilde{w}_i(x,y,z)dS}{\omega_i^2\int_D m\tilde{w}_i^2(x,y,z)dS}$$

where ***m*** is the ***mass per unit area***, D is the domain of the structure and dS is an element of this domain.
The method proposed by Blevins consists in two steps:

- first, in equation (3) J_i is assumed equal to 1 and the classic single d.o.f. forced oscillator equation is obtained
- then, the joint acceptance is evaluated and applied to the results as a scale factor

For stationary random pressure loading, with pressure spectral density $S_p(f)$, the acceleration response spectrum $S_A(f)$ of a ***single d.o.f.*** oscillator is:

(5) $$S_A(f) = |H(f)| S_p(f)/m^2$$

where the transfer function $|H(f)|$ is given by:

(6) $$|H(f)| = \frac{1}{\left[1-\left(\frac{f}{f_i}\right)^2\right]^2 + \left(2\zeta_i \frac{f}{f_i}\right)^2}$$

Supposing that $S_p(f)$ be constant over frequency and integrating (5) from f=0 to f= ∞, we obtain ***Miles' equation for a single d.o.f. system***:

(7) $$A_{i,RMS} = \frac{1}{m}\sqrt{\frac{\pi f_i S_p(f_i)}{4\zeta_i}}$$

Considering now the spatial contribution, the assumption that the joint acceptance is equal to 1 corresponds to the hypothesis that the acoustic wavelength is equal to the structural wavelength, that is:

(8) $$\tilde{p}_i = m\omega_i^2 \tilde{w}_i$$

This is not in general verified for a certain frequency; the correct value of the joint acceptance takes into account the real acoustic wavelength and can be inserted in (5) in order to obtain the final expression of the physical acceleration power spectral density (ASD):

(9) $$S_A(x,y,z,f) = |H(f)| S_p(f) \frac{J^2}{m^2} \tilde{w}_i^2(x,y,z)$$

which for f = f_i becomes:

(10) $$S_A(x,y,z,f_i) = \frac{J^2}{4\zeta_i^2 m^2} S_p(f) \tilde{w}_i^2(x,y,z)$$

Supposing that $S_p(f)$ be constant over frequency, the RMS acceleration in the ith mode is:

(11) $$A_i(x,y,z,t)_{RMS} = \frac{J}{m}\sqrt{\frac{\pi f_i S_p}{4\zeta_i}} \tilde{w}_i(x,y,z)$$

From this expression clearly results that, for each structural location, $A_{i,\ RMS}$ is directly proportional to J and to the modal displacement.

The procedure proposed by Blevins for the evaluation of the joint acceptance assumes that the acoustic waves propagate along both coordinates and the modes are separable along these coordinates; in addition, the method is based on the consideration that both the structural and acoustic waves can be approximated as sinusoids.

This formulation permits to decompose the problem of the two-dimensional joint acceptance evaluation into the calculation of the one-dimensional joint acceptances referred to the orthogonal surface directions of the plate.
A one-dimensional structural shape can be expressed as:

(12) $$\tilde{w}(x) = \sin\left(\frac{i\pi x}{L}\right)$$

where i represents the mode number, which is approximately equal to the number of structural half waves in the length L.
Expression (12) is rigorously exact only for completely simply supported rectangular plates.

A one–dimensional travelling wave can be expressed as the sum of two components 90 degrees out of phase, each component constituted by a spatial and a temporal term.
For acoustic waves in air, the acoustic wavelength is given by:

(13) $$\lambda_a = \frac{c}{f}$$

where c is the speed of sound

Now, the one-dimensional joint acceptance for a structural mode and an acoustic travelling wave can be evaluated as a vector sum of the two components:

(14) $$J_{1-Dwave} = \sqrt{J_{1D}^2(\phi = 0°) + J_{1D}(\phi = 90°)}$$

Then, the two-dimensional joint acceptance can be approximated by:

(15) $$J_{1Dwave} = J_{1Dwavex} \times J_{1Dwavey}$$

The one-dimensional joint acceptance for a sinusoidal structural mode and an acoustic travelling wave can be computed as a function of:

- the ratio of the structural wavelength to the acoustic wavelength $\frac{\lambda_f}{\lambda_a}$
- the structural mode number i

In Figure 4.0-1 the one-dimensional joint acceptance of a simply supported beam is represented as a function of $\frac{\lambda_f}{\lambda_a}$, for different structural mode numbers.

The two-dimensional joint acceptance relevant to a square plate can be computed by simply squaring the one-dimensional joint acceptance ***for those modes that have the same number of structural half waves in the two side directions***.

In Figure 4.0-2 the two-dimensional joint acceptance of a square plate is represented as a function of $\frac{\lambda_f}{\lambda_a}$, for different structural mode numbers.

The following considerations can be performed:

- if the acoustic wavelength greatly exceeds the structural wavelength in a fundamental mode, the maximum value of the two-dimensional joint acceptance is $16/\pi^2$ = 1.621, that represents the absolute maximum for all modes and all structural to acoustic wavelength ratios (i.e. all frequencies)
  In fact, by introducing in equation (4) the following hypothesis:

(16) $$\frac{\tilde{p}_1(x,y,z)}{m\omega_1^2}=1$$

(17) $$\tilde{w}_1(x)=\sin\left(\frac{\pi x}{L}\right)$$

where L is the panel length and

(18) $$\tilde{w}_1(y)=\sin\left(\frac{\pi y}{W}\right)$$

where W is the panel width.

We obtain:

(19) $$J_1=\frac{\int_D \tilde{w}_1(x)\tilde{w}_1(y)dS}{\int_D \tilde{w}_1^2(x)w_1^2(y)dS}$$

that is:

(20) $$J_1=\frac{\int_L \sin\left(\frac{\pi x}{L}\right)dx\int_W \sin\left(\frac{\pi y}{W}\right)dy}{\int_L \sin^2\left(\frac{\pi x}{L}\right)dx\int_W \sin^2\left(\frac{\pi y}{W}\right)dy}$$

Solving the integral we have:

$$J_1=\frac{\frac{2L}{\pi}\times\frac{2W}{\pi}}{\frac{L}{2}\times\frac{W}{2}}=\frac{16}{\pi^2}=1.621$$

- the joint acceptance value is always 1 in the coincidence condition (λ_f = λ_a) and represents the maximum value for all the modes higher than the fundamental one

- if there is a mismatch between acoustic and structural wavelength, the joint acceptance decays rapidly in the higher modes, that is the structural response decrease dramatically in the higher modes

- in synthesis, ***the joint acceptance function*** is an index of the efficiency of the acoustic field in exciting the structural modes and ***has a strong filtering effect on frequency responses to acoustic load***

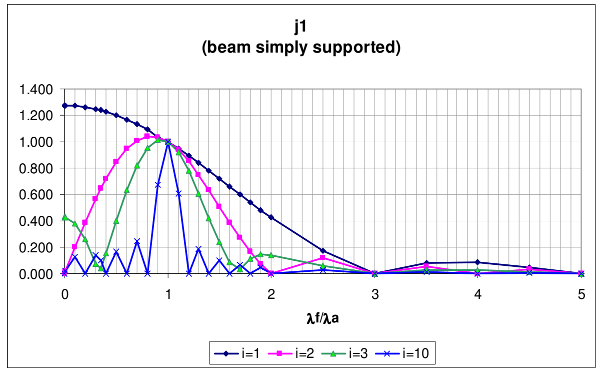


FIG 4.0-1 One-dimensional joint acceptance for simply supported beams

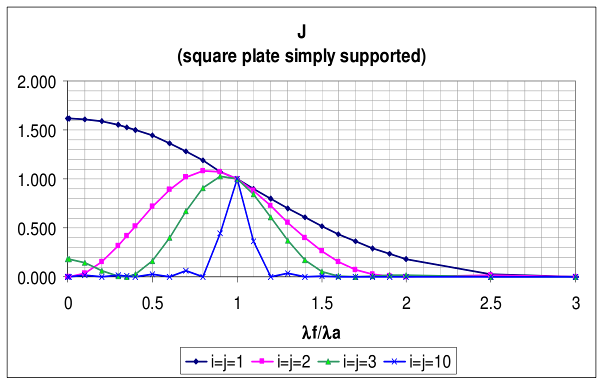


FIG 2.1-2 Two-dimensional joint acceptance for simply supported square plates

In order to extend the calculation of the joint acceptance to different boundary conditions and more complex panels, a FORTRAN program has been developed.

The program does not use analytical formulations for the structural shapes, but reads the natural frequencies and eigenvectors directly in the MSC.NASTRAN output file and calculates the joint acceptance by approximating the integrals in (4) with summations extended to the structural nodes (21):

(21) $$J_i=\frac{\sum_{j=1}^{N}\tilde{p}_j\tilde{w}_{ij}A_j}{\omega_i^2\sum_{j=1}^{N}m_j\tilde{w}_{ij}^2A_j}$$

Moreover, in case of pressure uniform on the structure, i.e. infinite acoustic wavelength, the joint acceptance may be computed, for each direction, as the ratio of the

participation factor and the generalized mass of the mode.
In R[1] ***one–dimensional joint acceptance*** values relevant to ***simply supported*** conditions are reported as function of mode order and structural to acoustic wavelength ratio.
The ***two-dimensional joint acceptance*** relevant to a square plate can be computed as the ***product of one-dimensional joint acceptances*** related to x and y directions.

The values in this way obtained have been compared with those numerically computed for a square simply supported plate in case of infinite and finite acoustic wavelength.
The joint acceptance values analytically and numerically computed for the main modes in case of infinite acoustic wavelength are reported in TAB 4.0-1.

Mode nr.	Frequency [Hz]	$L_x/(\lambda_{fx}/2)$	$J_{x, Blevins}$	$L_y/(\lambda_{fy}/2)$	$J_{y, Blevins}$	$J_{xy, Blevins}$	$J_{numerical}$	$J_{Nastran}$
1	293	1	1.273	1	1.273	1.621	1.619	1.619
2	731	2	0.000	1	1.273	0.000	0.000	0.000
3	731	1	1.273	2	0.000	0.000	0.000	0.000
4	1166	2	0.000	2	0.000	0.000	0.000	0.000
11	2610	3	0.424	3	0.424	0.180	0.178	0.178

$J_{xy, Blevins}$	Two-dimensional joint acceptance, analytically computed with the method reported in R[1]
$J_{numerical}$	Two-dimensional joint acceptance, numerically computed with the FORTRAN program
$J_{Nastran}$	Two-dimensional joint acceptance, computed from the modal parameters

TAB 4.0-1 Joint acceptance analytically and numerically computed for a square simply supported plate in case of infinite acoustic wavelength

Maximum values of joint acceptance squared for the first mode of uniform plates, with various combinations of elementary boundary conditions, are reported in TAB 4.0-2. These values are referred to the condition of infinite acoustic wavelength.

BOUNDARY CONDITIONS AT x = 0, a	BOUNDARY CONDITIONS AT z = 0, b			
	SS	SC	CC	CF
FF	1.409	1.448	1.487	1.577
SS	1.614	1.638	1.669	1.947
SC		1.659	1.692	1.988
CC			1.721	2.018
CF				2.222

TAB 4.0-2 Joint acceptance squared max. values vs. boundary conditions of squared plates

These values have been validated by comparison with those reported in R[3], taking into account the different formulation reported in R[3] w.r.t. R[1].

In addition, it has been verified that the same responses are obtained by performing a Random Response Analysis on the square plate FE model with the following simulation hypothesis:

a) $\tilde{p}_i = m\omega_i^2 \tilde{w}_i$

$J = J(\lambda_a = \infty)$

b) $\tilde{p}_i = 1$

J=1

This is an important results, because the Hypothesis b) allows to strongly simplify the FEM analysis.
This simplification, as explained in paragraph 4.3, has been adopted in the frame of HERSCHEL Random Response Analysis.

4.1. Acoustic Field Simulation

4.2. Frequency dependent component of the acoustic load

The third octave band average Sound Pressure Level (SPL) defined in Table 2.1 has to be converted in Pressure Spectral Density (PSD) and given as input data to NASTRAN random analysis module.

The RMS of the acoustic pressure can be derived using the following formula:

$$p_{RMS} = p_{REF} \times 10^{SPL/20} \quad (22)$$

where SPL is the sound pressure level and
$p_{REF} = 2 \times 10^{-5}$ Pa.

The corresponding PSD can be computed as:

$$S_p(f) = \frac{p_{RMS}^2}{f_2 - f_1} \quad (23)$$

where f1 and f2 are the limits of the frequency range over which p_{RMS} has been calculated.

4.3. Spatial dependent component of the acoustic load and excitation surfaces

In R[1] the formula to compute the wavelength of acoustic waves in air and flexure waves in plates versus frequency are reported. The coincident frequency is the frequency at which acoustic and flexure wavelenths are equal.

In figure 4.3-1 the wavelength of acoustic wave in air, 1.25 mm. thick Aluminium plate and HERSCHEL lateral equipment panels are plotted.

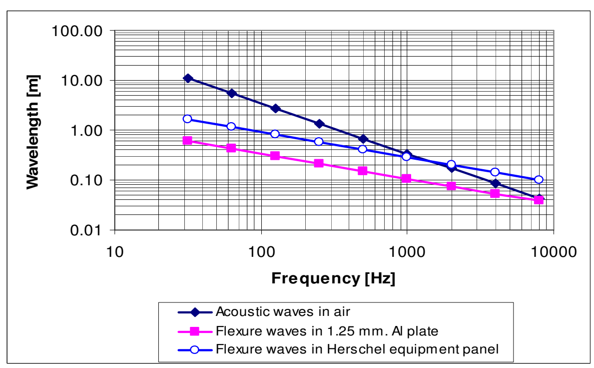


FIG 4.3-1 Acoustical and structural flexure wavelengths versus frequency

For the Al plate the coincident frequency is at 9770 Hz, while for HERSCHEL equipment panel it is at 1400 Hz.

From FIG 4.3-1, it can seen that the ratio $\frac{\lambda_f}{\lambda_a} = 0.4$ at 250 Hz and from FIG 2.1-2 it can be seen that J1 = 1.5 for the first mode and J2 = 0.5 for the second mode.

Since 250 Hz is the octave band central frequency of the band with the highest acoustic load, it has been fixed as the upper bound of the analysis.
At 250 Hz the acoustic wavelength is equal to 1.36 m, while the flexure wavelength is equal to 0.57 m.

In order to simplify the acoustic field model, an infinite acoustic wavelength has been assumed and the contribution to the responses due to the panels second mode has been embedded in that of the first mode, which was conservatively overestimated, assuming J = 1.621 in the whole frequency range.

In fact, with this simplified assumption, the spatial component of acoustic pressure becomes constant with respect to both position and frequency.
Therefore, the acoustic pressure distribution becomes independent from the mode shapes and the responses have not to be scaled with joint acceptance parameter.
In addition, with the adopted approach, the FEM derived mode shape is taken into account for the whole structure.

4.4. Random Analysis Performed with the MSC.NASTRAN Program

The implementation of the developed method in the MSC.NASTRAN program is articulated in a series of steps:

- A Modal Analysis is performed in order to obtain the eigenvalues and eigenvectors of the structure
- A unitary pressure load uniformly distributed is created, in the whole frequency range, and applied to all the surface directly exposed to the acoustic field.
- The damping ratio is estimated from analytical and/or experimental data
- A modal Frequency Response Analysis is performed to compute the transfer functions in the frequency range of interest
- A Random Response Analysis is performed applying the PSD of the acoustic pressure and computing the response levels in terms of spectral density and RMS in the selected output locations

From the Random Vibration Analysis any kind of output can be obtained, besides the acceleration levels, both in terms of PSD and RMS. For instance interface forces, forces in the spring elements and stresses in shell and beam elements can be required.
A modal basis up to 500 Hz has been extracted in order to compute the results of the vibro-acoustic analysis up to 250 Hz. A value of 2% has been adopted for ξ, which means a dynamic amplification factor Q=25.

The test spectra have been derived by enveloping the Random Vibration Environment (R.V.E.) computed in the frequency range 20 – 250 Hz and then extrapolated up to 2000 Hz by adopting commonly used profiles.

5. TEST DESCRIPTION

HERSCHEL vibro-acoustic qualification test has been performed with the qualification acoustic levels reported in FIG 5.0-1(R[5]), less severe than those used for the analysis in the medium / high frequency range.

Qualification level test		
OCTAVE BAND CENTRE FREQUENCY (Hz)	QUALIFICATION LEVEL (dB) Ref.: 0 dB = 2.10^{-5} Pa	TEST TOLERANCE (dB)
31.5	131dB	-2, +4
63	134dB	-1, +3
125	139dB	-1, +3
250	138dB	-1, +3
500	135dB	-1, +3
1000	129dB	-1, +3
2000	123dB	-1, +3
Integrated level	143.5dB	-1, +3
Test duration :	2 min	

TAB 5.0-1 HERSCHEL vibro-acoustic qualification test acoustic levels (R[5])

The vibro-acoustic test has been performed in the Large European Acoustic Facility (LEAF).
The satellite has been mounted on a test adapter previously used for XMM provided by ESA. The clamp band interface was mechanically identical to a flight one.
The measured satellite mass during the whole HERSCHEL qualification test campaign was varying from 3201 to 3268 (depending from the Helium filling).

Measurement of sound pressure levels has been done by microphones. Power spectral density response has been measured by accelerometers.
1/3-octave control has been used.
9 omnidirectional microphones have been laid around the flight model in order to check and pilot the noise levels of the acoustic environment.
The average level of these 9 microphones was the basis for the tolerance check in each octave band.
The test measurements has been performed at a minimum distance of 1 m from the spacecraft.
Two supplementary microphones have been added for information only (not for piloting purpose):

- M10 : Between HSS and CVV (X=2200 , Y=0 , Z = 1400 in S/C coordinate frame)
- M11 : Inside SVM box near the reaction wheels (near lateral panel –Y+Z)

A sketch of HERSCHEL Vibro-acoustic Test configuration is reported in FIG 5.0-1a and b.

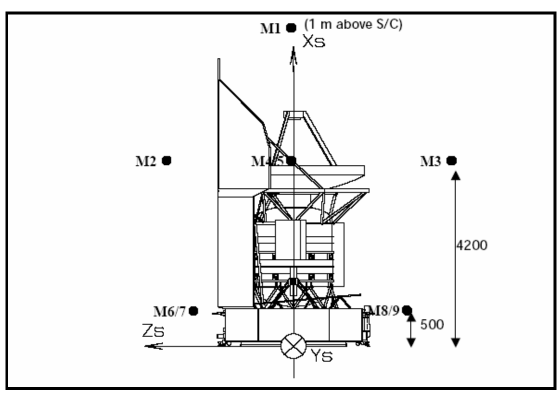


FIG 5.0-1a Acoustic test configuration

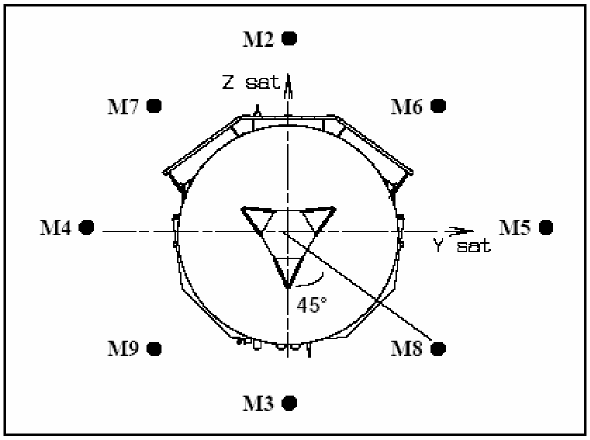


FIG 5.0-1b Acoustic test configuration

6. TEST RESULTS

The vibro-acoustic test has been successful, since all the immediate and delayed success criteria have been satisfied.

6.1. Quality of the Acoustic Environment

The quality of the acoustic environment is to be intended as homogeneity of the response of any couple of control microphones in the acoustic spectrum bands.
The specified requirement was:

- 1st and 2nd octave bands : +/- 2 dB per octave band
- All other octave bands : +/- 2 dB per 1/3 octave band

This requirement has not been satisfied, in particular for 31.5 Hz octave band, nevertheless the obtained homogeneity has been deemed acceptable, since it is in line with the acoustic environment quality reached in the frame of previous projects.

In particular microphone 8, located near the SVM –Z+Y lateral panel, has recorded the highest levels, and microphone 1, above the S/C, has recorded the lowest level.
The averaged acoustic field compared with upper and lower tolerances per octave band is sketched in FIG 6.1-1.

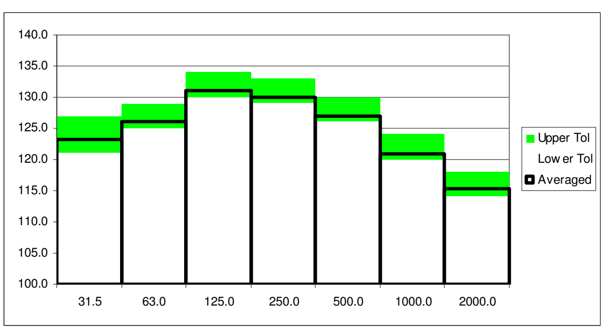


FIG 6.1-1 Averaged acoustic field

6.2. Comparison between LL1 & LL2

The comparison between pre- and post-low level runs for SVM locations confirms that the structural integrity requirement has been satisfied.
An example is reported in FIG 6.2-1.

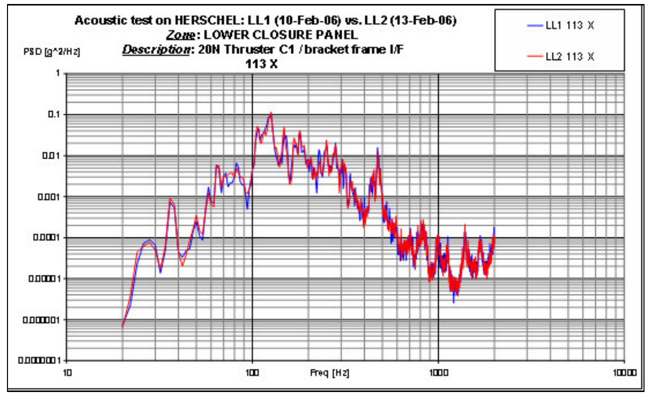


FIG 6.2-1 Pre- and post-low level runs comparison (20 N thruster)

6.3. Comparison with HERSCHEL Equipment Qualification Levels

The SVM equipment specified random environment has been generally validated, nevertheless for some units some exceedances w.r.t. the specified test levels at their I/F have been detected.

The list of critical units is reported in Table 6.3-1 and , for each of them, a dedicated assessment has been performed.

	ITEM	LOCATION	ACCELEROMETER
1	LGA	(+Z-Y shear)	412OOP
2	MGA	(+Z+Y shear)	423OOP
3	Gyro	(+Z+Y shear)	424Y
4	CRS	(+Y+Z shear)	432Z
5	20N Thruster	(bracket I/F)	114X, 117X
6	AAD	(+Z+Y shear)	423OOP
7	SAS	(+Z+Y shear)	424Y
8	ACC	(+Y lateral)	332Y, 333OOP
9	CDMU	(+Y lateral)	337OOP, 338OOP

TAB 6.3-1 Random spec. exceeded units on SVM

6.3.1. Analytical 1 D.O.F. Transmissibility Function Method

Due to the lack of accelerometers at unit CoG no experimental data are available about the response at CoG for most critical units, except LGA and MGA.
Therefore, from LGA experimental results a 1 d.o.f. transmissibility function has been tuned:

- LGA resonance frequency has been fixed to 185 Hz
- LGA amplification factor at resonance has been fixed to 10

A minimum threshold equal to 10/2 has been fixed for the transmissibility function to cover eventual secondary resonances (FIG 6.3-1).

The ASD measured at LGA to panel I/F has been multiplied for the transmissibility function and the g_{RMS} of the analytical function has been computed.

The 1 d.o.f. transmissibility function has been validated by verifying that the analytical assessed g_{RMS} is conservative w.r.t. the measured one (FIG 6.3-2).

Also from MGA experimental data an additional validation of the 1 d.o.f. transmissibility function approach has been obtained.

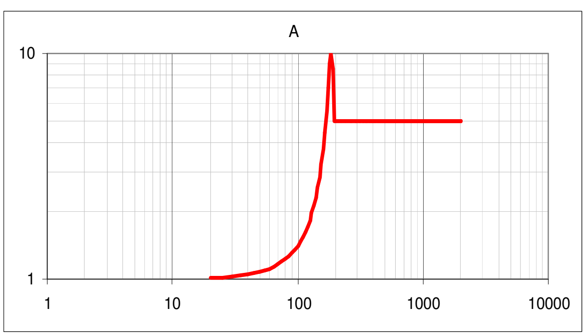
Then, an analytical 1 d.o.f. transmissibility function has been tuned for each critical unit, based on the experimental main resonance frequency in the out-of-plane direction and the related amplification factor.

The response at CoG has been evaluated by multiplying the experimental ASD at unit to panel I/F by the transmissibility function.

The g_{RMS} of the assessed response at system level has been compared with the g_{RMS} reached at unit level, to verify if a re-qualification of the unit is needed.

In fact, the g_{RMS} of the unit CoG response at system level takes into account the effect of the exceedances detected w.r.t. the specified levels at unit I/F.

The Miles approach instead does not take into account the effect of the same exceedances, since only the contribution of the response at the frequencies corresponding to unit modes with significant effective mass is considered for load computation.



EXPERIMENTAL DATA

- f_{RES} = 185 Hz
- Q_{RES} = 10

FIG 6.3-1 LGA tuned 1 d.o.f. transmissibility function

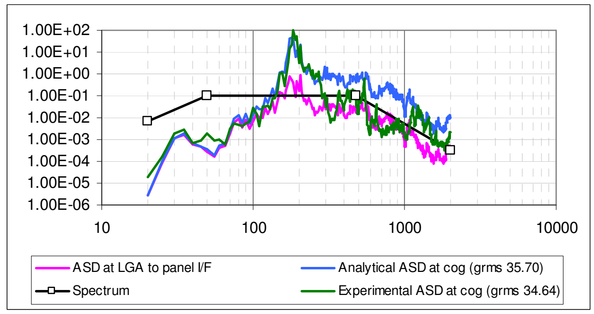


FIG 6.3-2 LGA tuned 1 d.o.f. transmissibility function validation

For most equipment, the identified exceedances are decoupled from units resonance frequencies and therefore no re-qualification is required.

For the remaining ones, the g_{RMS} reached during the STM acoustic test is always lower than the g_{RMS} reached during the unit qualification test and therefore no re-qualification is required.

The identified exceedances have been mainly detected on shear panels located in +Z area.

They seem to be related to the interactions between PLM and SVM (in particular high Solar Array panels responses to the acoustic field in the frequency range 100 to 200 Hz,) that has been disregarded by the analysis performed with a rigid PLM.

In FIG 6.3-3 the ASD levels recorded by accelerometers located on Solar Array shield, along its axis of symmetry (satellite X axis), are shown. The out of plane (Z) direction is considered.

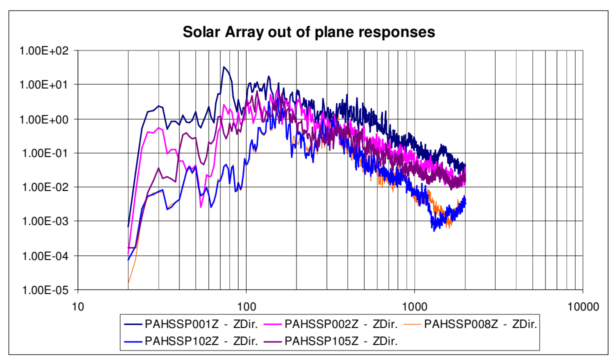


FIG 6.3-3 Solar Array out of plane responses

7. TEST RESULTS COMPARISON VS RANDOM VIBRATION ANALYSIS RESULTS

For each unit installed on each SVM area, the comparison between analytically computed environment, defined qualification spectrum and measured ASD levels has been performed.

An overview is given in FIG 7.0-1 a, b, c, d.

The results relevant to three lateral equipment and one shear panel are sketched.

A very good agreement, particularly in terms of broadband levels, can be highlighted. This allows to define reliable qualification spectra, especially in the flat zone.
The flat zone has been generally extended up to 300 Hz and standard slopes have been adopted in both the low and high frequency range, as conservatively suggested in R[3].

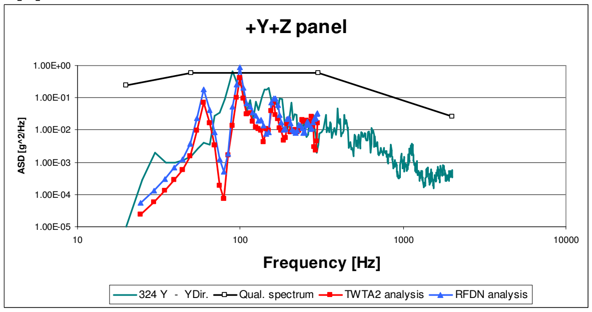


FIG 7.0-1a Test vs Random Vibration Analysis Results comparison (+Y+Z lateral panel)

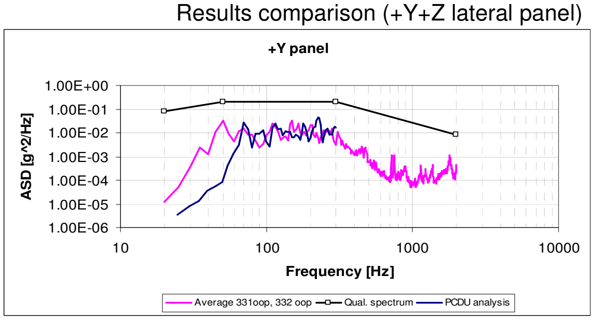


FIG 7.0-1b Test vs Random Vibration Analysis Results comparison (+Y lateral panel)

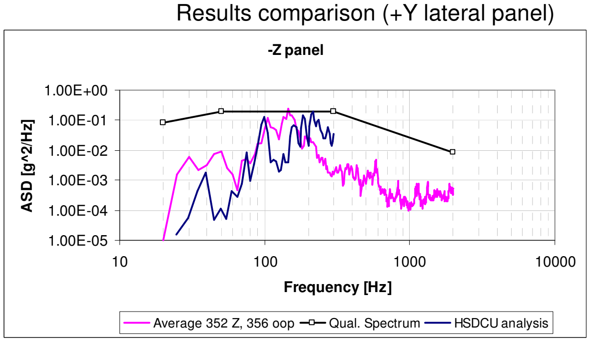


FIG 7.0-1c Test vs Random Vibration Analysis Results comparison (-Z lateral panel)

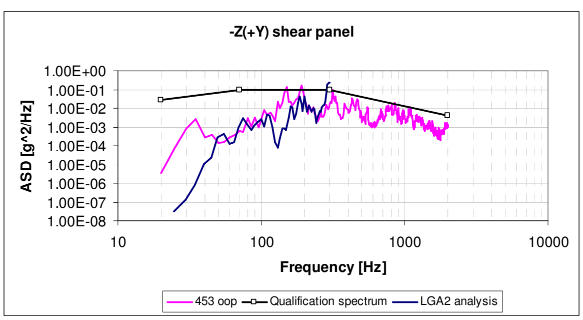


FIG 7.0-1d Test vs Random Vibration Analysis Results comparison (-Z+Y shear panel)

8. CONCLUSIONS

An original simplified model of the launch acoustic field was built up and applied to the structural SVM FEM of HERSCHEL satellite, as input to Random Vibro-acoustic Analysis performed with the Nastran program, in order to derive the random environment relevant to SVM zones and to specify Random Spectra for equipment units.

The method is based on a modal approach which takes into account the spatial correlation between impinging pressure and structural modes in the relevant frequency domain.

250 Hz has been fixed as the upper bound of the analysis, which was sufficient to define the levels relevant to the flat zone of the random spectra.
Consequently, the hypothesis of infinite acoustic wavelength could be adopted in the whole frequency range of the analysis, resulting in a strongly simplified and low time-consuming analytical activity.
In fact, a modal Frequency Response Analysis with uniform pressure input, followed by a Random Response Analysis with specified PSD input, was sufficient to derive the complete SVM environment.

The results of the Vibro-acoustic Test performed on HERSCHEL STM has shown a very good agreement w.r.t. prediction.

9. ACRONYMS & SYMBOLS

STM	Structural Thermal Model
FEM	Finite Element Model
SVM	Service Module
SEA	Statistical Energy Analysis
PLM	Payload Module
PSD	Pressure Spectral Density
ASD	Acceleration Spectral Density
RMS	Root Mean Square
$\tilde{w}$	Mode shape
J	Joint Acceptance
ω	circular frequency, radians per second
f	frequency, Hz
λ_a	Acoustic wavelength
λ_f	Structural wavelength
C	Speed of sound
CoG	Centre of Gravity
d.o.f.	degree of freedom
w.r.t.	with respect to
LGA	Low Gain Antenna
MGA	Medium Gain Antenna

10. REFERENCES

[1] Blevins R.D., "An Approximate Method for Sonic Fatigue Analysis of Plates and Shells", Journal of Sound and Vibration (1989) 129(1), 51-71
[2] Lyon R.H., Statistical Energy Analysis of Dynamical Systems: Theory and Applications, MIT Press,1975
[3] Structural Acoustic Design Manual, ESA PS-03-1201
[4] Ariane 5 User's Manual Issue 3
[5] Ariane 5 User's Manual Issue 4