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NOTCHING GUIDELINES FOR MECHANICAL TEST

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OUTLINE

1. SCOPE

This note defines the Tests Notching philosophy for both sinus and random vibrations environment

It addresses in particular the way to determine notchings in the sine and random input spectra, first during the dimensionning and design phase, secondly during the testing of units.

Quasi-static and dynamic loads are specified in order to meet the requirements which are applicable at instrument level. However, in order not to overdesign and overtest assemblies or units, notchings in the specified input spectrum are allowed to some extent, based on experience and/or instrument coupled loads analysis results. Fulfillment of structural stiffness requirements is mandatory since load amplification is much dependent upon dynamic decoupling. It is warned therefore that the notching rules are applicable only if the equipment complies with these requirements.

2. MECHANICAL TESTS OBJECTIVES

The assemblies and units are designed to withstand the mechanical environments they will encounter during their lifetime without degradation of their performances.

The mechanical tests objectives are to verify that the assemblies and the units are capable to withstand the mechanical environments to which they have been designed.

The basic stiffness and load verification process includes the following mechanical tests at assembly/units level.

❶ Quasi-static loading tests :

These tests must demonstrate the ability of the unit to withstand quasi-static loads induced by launcher or by sine testing at instrument level.

At primary structure level, quasi-static loads provide the dimensionning loads in particular at attachment interfaces. They act at low frequency, typically below 10 Hz. They can be verified by two different ways :

- by means of a sine sweep in the range 20/30 Hz. The frequency range and sweep speed are adjusted to produce a test duration of about 1 minute (TBC).
- by means of a quasi-static test at a determined frequency, typically 20 Hz, during 1 minute (TBC).

❷ Sinus vibrations tests :

Sinus vibration testing is used to demonstrate the ability of the unit to withstand low frequency excitations of the launcher, typically in the range 5-200 Hz, with possible dynamic amplification of the unit.

Detailed objectives are :

At qualification level :

- to verify the eigenfrequencies content, thus to validate the mechanical finite element model and dynamic decoupling requirements.
- to demonstrate the ability of the unit to withstand the loads and resulting stresses induced by sinusoidal vibrations.

At acceptance level :

- to verify the absence of material and workmanship defects.

❸ **Random vibrations tests** :

Random vibration testing is used to demonstrate the ability of the unit to withstand the random excitations which may come from :

- a pure random environment : applicable to compact units located inside the main structure. The effects of direct acoustic environment are negligible, random excitations come from effects of acoustic environment induced on satellite, from aerodynamic excitations and transmitted structure vibrations.
- a direct acoustic environment : applicable to appendages as covers, baffle. The effects of direct acoustic environment is dimensionning, the effects of random excitations induced at interface levels by acoustic on satellite are neglected.

Both direct acoustic environment and induced random vibrations are applicable to other units. The effects induced by the two environments are combined to provide the random specifications applicable to units.

The random excitations relates to a high frequency range 100-2000 Hz. thus are dimensionning for local modes and secondary structures.

3. NOTCHING PHILOSOPHY

Primary notching

The first objective of notching is to reduce the excitation level during sinus or random vibration testing in order not to exceed the sizing loads at the equipment interface. (⇒ I/F loads < QSL I/F loads,

Secondary notching

If necessary, to avoid unrealistic overtesting of items within internal structures of unit's and in the case it is felt that the test levels will damage the unit, the sub-contractor may request in addition a secondary notching to the prime. This applies only if the primary notching is not sufficient and if the subcontractor can show strong evidence that the equipment will suffer damages.

The following sections give guidelines for raising a notching request and gives methodology for calculating the notching.

The guidelines apply not only during testing, but also in the early stage of development when the subcontractor has to identify notching needs in the sizing process of the unit.

4. ESSENTIAL RULES

The following rules **shall be followed**, otherwise the notching request **shall be rejected**.

a) Any notching must be formally requested as a waiver/deviation to a specification, submitted to the prime agreement.

b) A notching assessment shall be included in the frame of the PDR/CDR.

c) Formal notching request shall be raised before the equipment MRR, so that a possible rejection of the notching by the Prime will not impact the development of the equipment.

d) Any notching request shall be based on a technical report including at least :

. preliminary quasistatic analysis results (critical areas, margins, interface loads),

. preliminary harmonic response analysis results (critical areas, margins, interface loads).

e) The notching assessment shall be based on a detailed structural analysis. It is strongly advised to use a finite element model representative of the physical structure.

5. NOTCHED SINE SPECTRUM

The following guidelines are provided (see flow diagram).

Notching predictions

a) Forbidden notching range :

Notching of the input spectrum may be forbidden by the Prime in part of the frequency range.

b) Notching level determination :

For each mode, under sine/vibration, the algebric sum of loads at the unit/assembly interface (for all specified axis) is divided by the total mass of the unit/assembly, thus providing the equivalent quasistatic acceleration on the unit/assembly. The sine level reduction is authorized if this equivalent quasistatic acceleration if higher than the specified quasistatic acceleration at qualification level. (QSL x 1.25). This must be fully justified by a technical report.

The damping factor used for the notching analysis shall be justified by the contractor experience.

Testing

The unit/assembly supplier shall identify the location of the accelerometers at the adequate areas (interfaces) in order to obtain a good correlation between levels on accelerometers and interface loads. The test instrumentation is under the supplier responsibility. When used for notching purposes. strain jauge instrumentation shall be delivered with the units.

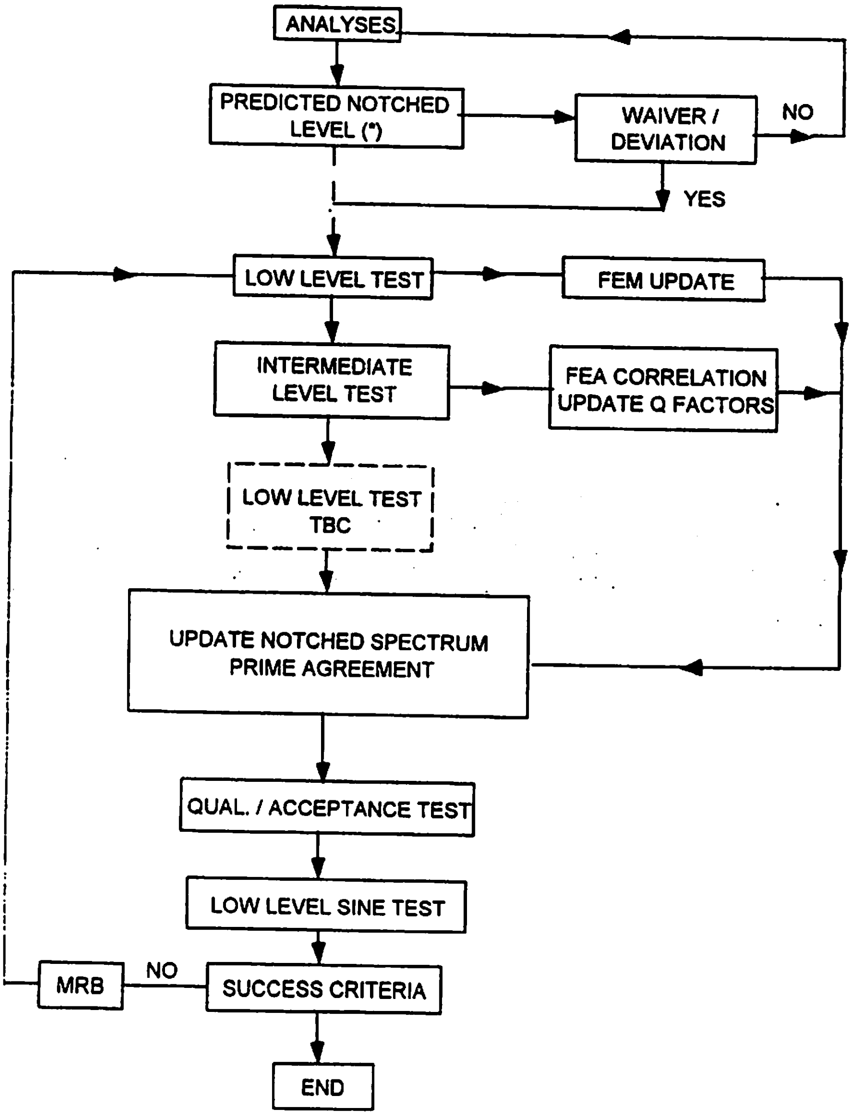
Ways to identify eigenmodes, to calculate damping factors and to determine the interface loads is common to sinus and random testing. They are developped in the next section.

In any case. the authorization to reduce the sine level during the test shall be confirmed by the Prime after review of test results at sine low level. A correlation between analysis and test shall be provided by the supplier.

Each notching request shall clearly identify the underlying assumptions :

- frequency range,
- level (g or g^2/Hz).
- slope.
- amplification factor,
- effective mass and total mass.
- notching level constant C (see section 6.2.2).

QUALIFICATION / ACCEPTANCE SEQUENCE



* Based on quasi-static loads

6. RANDOM TEST INPUT SPECTRUM

Notching of the specified input spectrum may be allowed in some specific cases. The general rules are described hereafter.

Predictive analysis of the test is strongly advised since it will allow to identify in advance the resonances which may be critical for the equipment. prepare the measurement plan. determine the notched input spectrum according to the specified rules and request deviations if need be.

6.1. GENERAL NOTCHING CRITERIA

Applicable rules are slightly different for large and small equipments. For large equipents (> 10 Kg). it is expected that quasi-static and/or sine loads are more critical than acoustically induced stresses except possibly for very large baffles or covers.

For small equipments (< 3 Kg FPO for instance), high frequency dynamic excitation may be the sizing load case.

6.1.1. Notching criteria for large equipments

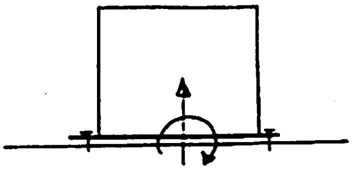
The input spectrum may be notched so that the total interface load at 3σ is not higher than the quasi-static loads at qualification level. The input spectrum shall be first notched at the main resonances. and if needed outside the main resonances in the high frequency range.

Notching outside resonances however shall be approved by the main contractor.

6.1.2. Notching criteria for small equipments

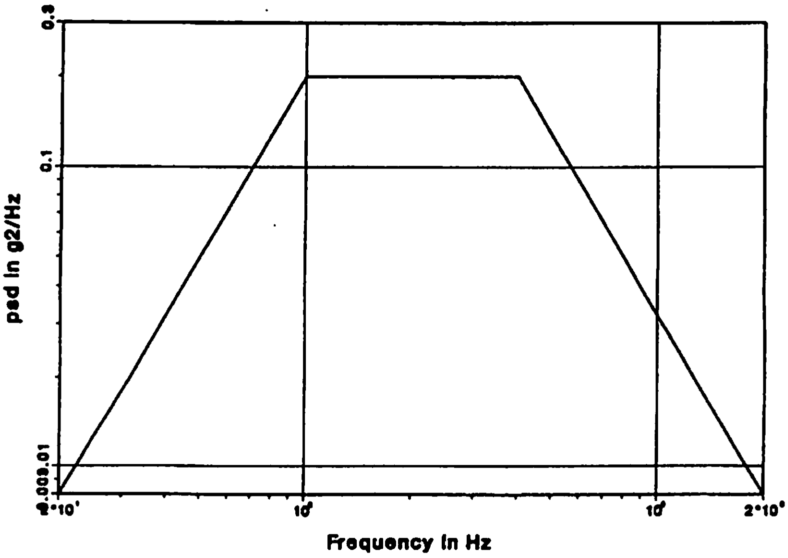
Notching at main resonances on the basis of quasi-static loads is also applicable, but minimum input levels are required on the whole frequency range (see fig. 6.1.2). This means that the random test may be a sizing load case for some equipments.

GENERAL NOTCHING CRITERIA



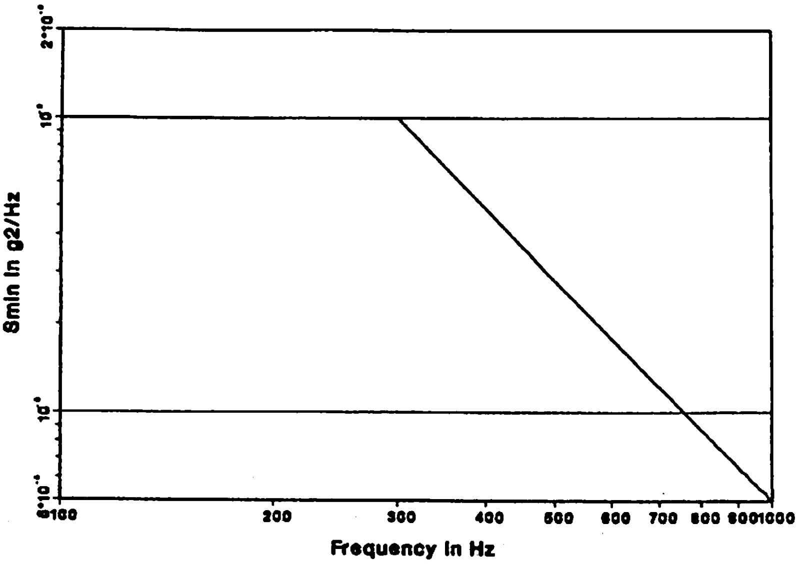
F(ω) = Load resultant at interface for a unit input (N^2/Hz)

$$3\sqrt{\int_{\omega min}^{\omega max} S(\omega)\,F(\omega)\,d\omega} \le \text{Quasi-static loads}$$

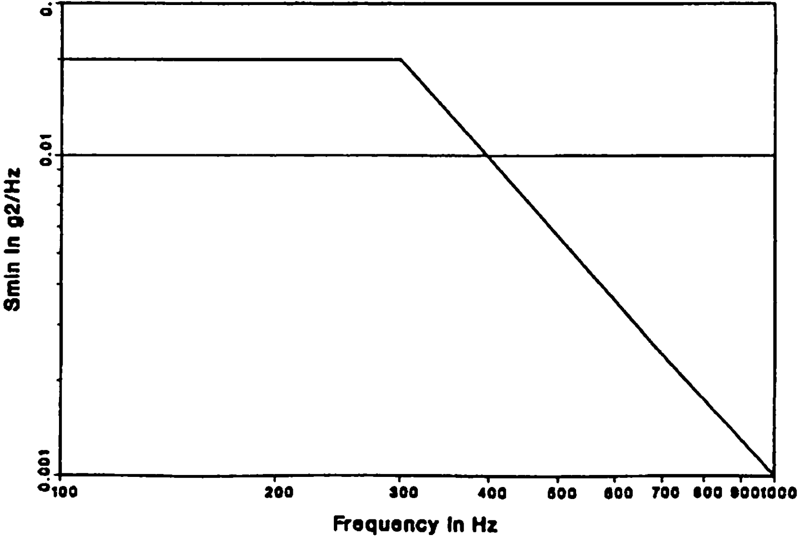


Input spectrum S(ω)

MINIMUM INPUT LEVELS FOR SMALL EQUIPMENTS



In plane excitation axis



Out of plane excitation axis

6.2. DETAILED NOTCHING CRITERIA

6.2.1. Notching of a single resonance

When the equipment may be considered as a single DOF structure, in the frequency test range its behaviour is described by the following equations :

M : equipment mass

k : stiffness

ξ : 1/2Q : damping coefficient

$W_0 = \sqrt{\frac{K}{m}} = 2\pi f_0$: eigenvalue

m : modal effective mass

The PDS of the acceleration γ (w) at the center of gravity of the equipment is :

$$\gamma(w) = \overset{2}{H}(w)\, S(w)$$

where the transfer function H(w) can be written

$$H(w) = \frac{1 + 2\,i\xi \left(\frac{w}{w_0}\right)}{1 - \left(\frac{w^2}{w_0^2}\right) + 2i\xi \left(\frac{w}{w_0}\right)}$$

The average mean squared acceleration is computed from :

$$\gamma^2 = \int_{w_{min}}^{w_{max}} \left|H(w)\right|^2 S(w)\, dw$$

After integration, the RMS acceleration is approximatively equal to :

$$\gamma RMS = \sqrt{\frac{\pi}{2}\ f_0\ QS\ (f_0)} \text{ (Mile's formula)}$$

It has been assumed that the eigenfrequency fo is within the excitation range.

The notched input spectrum S* (w) will now be determined so that, unless some minimum rules apply, the total interface load at 3σ does not exceed the quasi-static load :

$$3m \sqrt{\int_{w_{min}}^{w_{max}} \left|H(w)\right|^2 S^*(w)\, dw} \le M\gamma_{QS} \qquad (1)$$

Or approximately $3m \sqrt{\frac{\pi}{2}\ f_0\ QS\ (f_0)} \le M\gamma_{QS}$

This condition will be met in general by notching exclusively around the resonance. The notched input spectrum is then characterized by :

- the width of the notch
- the minimum input level at the resonance
- the slope of the PSD

The width of the notch around the resonance is dependent upon the amplification factor and is set to :

$$\Delta w = \frac{3\ w_0}{Q} \qquad \Delta f = \frac{3f}{Q}$$

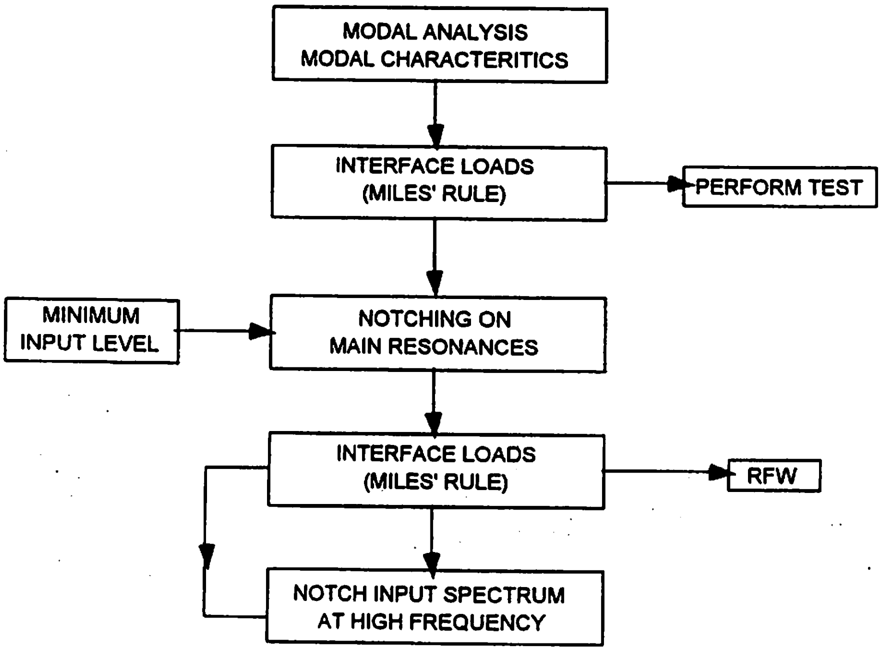
The slope of the PSD will be approximately : p = Q dB/oct .

Once these parameters are defined, the minimum input level at the resonance $S(w_o)$ will be computed in order to satisfy criterion (1).

NB : If the residual rigid mass is not negligible, (> 20 %) criterion (1) should be written.

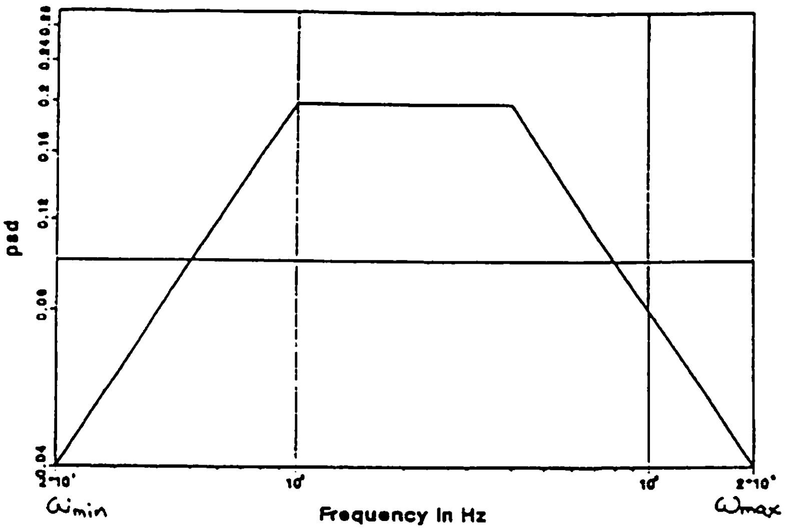
$$3\sqrt{\int_{w_{min}}^{w_{max}} \left[m^2\ |H(w)|^2 + (M-m)^2\right]\ S^*(w)\ dw} \leq M\gamma_{QS}$$

PREDICTED NOTCHED LEVEL FOR RANDOM QUALIFICATION

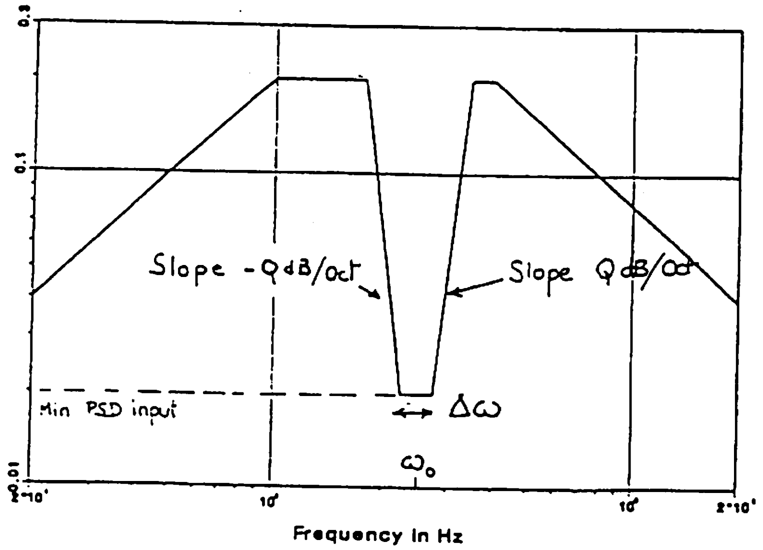


$dB = 10 \log_{10}\left(\frac{R_1}{R_2}\right)$

So $25 \times 10 \log_{10}\left(\frac{R_1}{R_2}\right) \Big/ \text{oct.}$



Specified input spectrum



Notched input spectrum

6.2.2. Notching of a multi DOF equipment (see flow diagram)

For a multi resonance structure, the same basic philosophy is applicable. The notching criterion however must take into account all the resonances :

$$3\sqrt{\int_{w_{min}}^{w_{max}} \left| \sum_{i=1}^{n} mi^{2} \{Hi(w)\}^{2} + M Res^{2} \right|^{2} S^*(w)dw} \le M\gamma_{\underset{\sim}{QS}} \qquad (2)$$

where mi are modal effective masses and M_{res} is the residual rigid mass.

Usually inequality may be replaced by :

$$3\sqrt{\sum_{i=1}^{n}\left\{mi\sqrt{\frac{\pi}{2}\ f_i\ Q_i\ S^*\ (f_i)}\right\}^2 + M^2 Res \int_{w_{min}}^{w_{max}} S^*(w)dw} \le M\gamma_{QS} \qquad (3)$$

If we assume now that the input PSD S(w) is representative of the mechanical power delivered to the equipment, we can write for each mode :

$$P_i(w_i) = \underline{F_i} V_i = C S(w_i) \qquad \frac{Nm}{s}$$

where F_i is the interface load and V_i the ~~mobility~~. velocity

$$\text{Hence.}\ \frac{m_i\ \gamma_i^2}{w_i} = CS(w_i) \qquad m\,a\,V = \frac{m a^2}{\omega} \qquad a = \frac{\pi}{2} f\, Q\, W(I)$$

we may use Mile's approximation for the acceleration γi, which yields :

$$= \frac{f_i}{4}\, Q\, W(I)$$

$$\frac{m_i}{4}\ Q_i\ S^*\ (f_i) = CS\ (f_i)$$

The minimum input level at the resonances will therefore be such that :

$$\frac{m_i\ Q_i\ S^*\ (fi)}{S\ (fi)} = \text{constant} \quad (4) \qquad \Rightarrow \quad PSD_{notched} = \frac{PSD_{initial} \times K}{m_i \times Q_i}$$

K [illegible]

Numerical applications are shown in annex 1.

When (3) = M γ_{QS} [illegible]

Equation (4) means that resonances with large effective mass and amplification shall be more deeply notched than local resonances which induce small interface loads.

6.2.3. Combined interface loads

When complex structures are tested, combination of loads in different directions have to be considered. In that case, the equipment supplier shall extend the verification so that the set of RMS interface loads remains inside the domain of definition of the quasi-static loads.

6.3. NOTCHING DURING THE TESTS

For each axis and each type of vibration test, the test campaign shall begin and end with a low level sine test. Identical signatures for these tests is necessary to infer that the equipement has not suffered damages during the qualification/acceptance tests (see flow diagram).

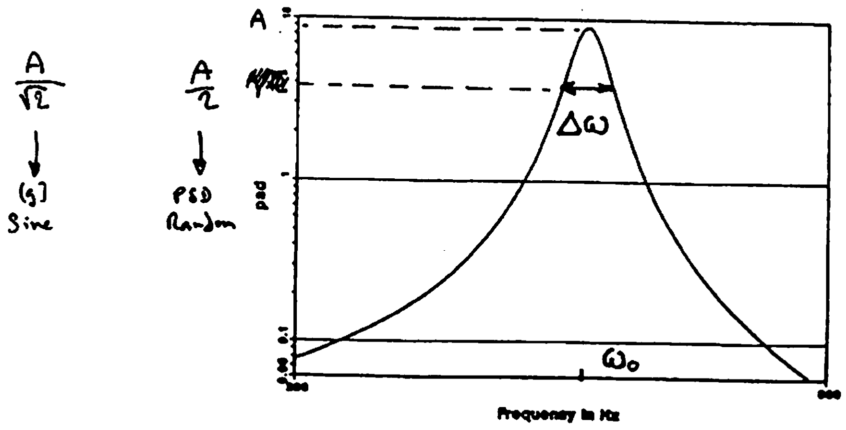
6.3.1. Identification of eigenmodes

The sine test results are used to identify the main equipment structural modes and determine the amplification factor. Special attention will be exercised in the 100-500 Hz range where the input levels are high and the equipments are usually sensitive. If the modal density is expected to be large, the instrumentation shall be devised accordingly.

Two basic methods may be used to determine the associated damping coefficients.

The first method requires the results of a predictive (FEM) analysis. If the correlation between the analytical results and the test measurements is correct, direct tuning of the damping coefficient is then possible for each mode based on the measured peak levels.

If no analytical result is available or the FEM analysis appears to be unreliable, the amplification factor shall be determined from the resonance width (see fig.).



$$Q = \frac{f}{\Delta f} = \frac{\omega_0}{\Delta\omega_0}$$

The amplification factor Q and structural damping coefficient ξ are related to the resonance width measured at the "half power point" through the following relationship :

$$\frac{\Delta w}{w_o} = 2\xi = \frac{1}{Q}$$

This method may be unreliable when the modal density is large or when the tests results are noisy.

When none of the above methods is applicable, the only way is to analyse closely the mode shapes from the test measurements, determine the CoG of the deformed structure and infer the amplification factor from a simple one dof result :

$$Q = \frac{A}{A_o}$$

where A_o is the base acceleration and A the acceleration of the moving mass.

The method is only a crude one and is really applicable for the first lateral mode and well identified local modes.

6.3.2. Determination of the interface loads

Estimation of the net resultant forces and moments at the equipment interface is necessary since they will be used to determine the final notched input spectrum. Four different methods which are briefly described hereafter may be implemented.

The easiest way is based on the FEA results. The method is applicable only if the mode shapes of the main eigenmodes are properly described by the model. When this is the case, simple tuning of the damping coefficients is sufficient to obtain reliable interface loads.

Another simple method relies upon the measurement of the electric intensity in the coil of the shaker. The current is proportional to the total load in the excitation axis. Correcting factors are necessary to account for the rigid moving mass (test rig and coil). Hence, the method may not be accurate for small equipments. Furthermore loads are not available in directions other than the excitation axis. In particular the interface moment induced by the lateral excitation cannot be directly measured.

The third method which requires strain gauges or other load measuring devices (piezoelectric for instance) may not be practical. It is probably however the most reliable way to determine the interface loads since in principle no correction factor is necessary.

The last method requires the knowledge of the acceleration levels in the structure and the associated mass distribution. The amount of instrumentation which is necessary depends upon the number of modes which are investigated, but in general a fairly large number of accelerometers will be needed to obtain accurate interface loads.

6.3.3. Determination of the notched spectrum

The rules which have defined in 6.1 remain applicable. The only difference lies in the origin of the basic data which are experimental rather than analytical.

In the special case of units equipped with blades (ISM), the output level psd measured at the upper blade interface may be limited to $1 g^2/Hz$ above 600 Hz.

6.3.4. Random tests at intermediate levels

It is strongly advised to run tests at intermediate levels when the extrapolation of the qualification results shows low margins of safety on the equipment, or when the damping coefficient is expected to change because of high excitation levels (for instance -10 dB).

6.3.5. Measurement plan

The number of accelerometers / strain jauges needed depends of course on the size of the equipment and the number of expected eigenmodes in the critical frequency range (100-500 Hz).

For small equipments (mass below 1 Kg) which are isostatically mounted with only a few modes in the low frequency range, a single 3-axis accelerometers may be sufficient.

On the other hand, for large equipments with a number of complex modes, the instrumentation shall be much more important.

It is recalled that in all cases, the quality and quantity of experimental results must allow unambiguous determination of critical modes and associated modal effective masses before the final qualification notched spectrum is defined.

ANNEX 1

Equipment dynamic characteristics

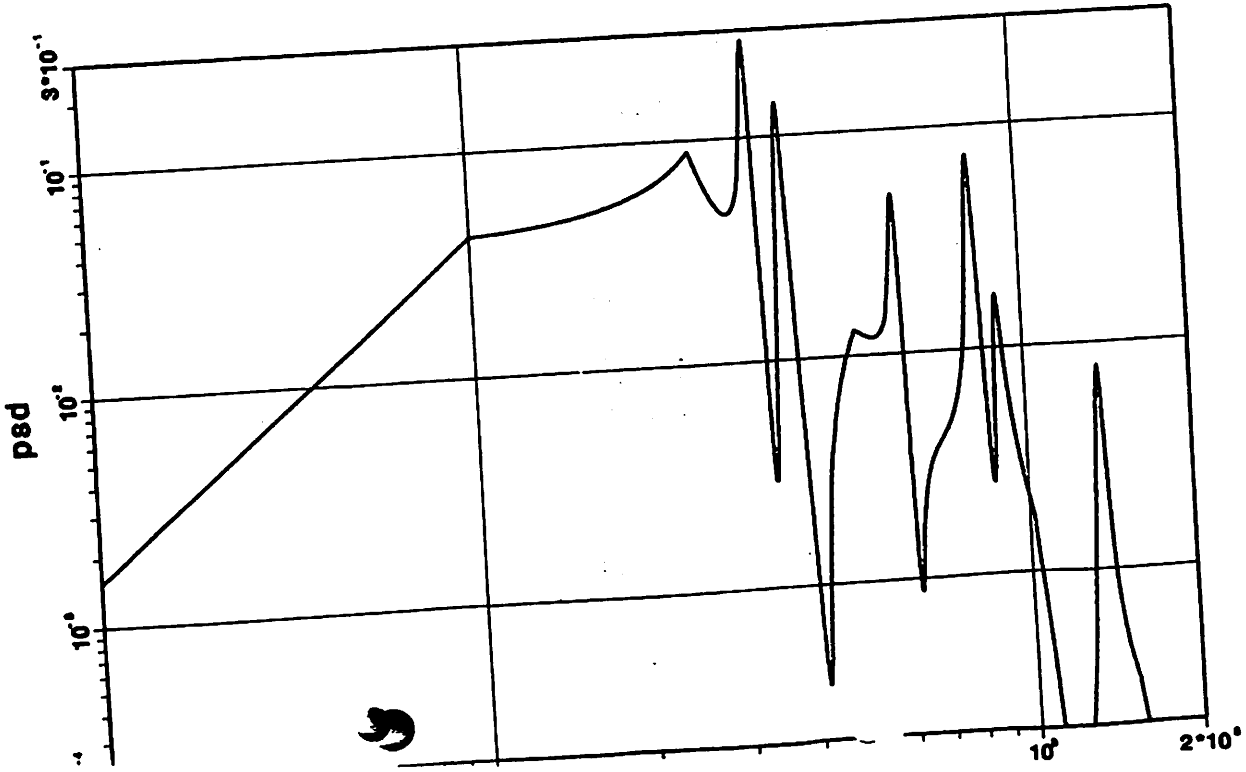
Total mass = 0.381

Quasi-static load = 35 g

Mode number	Eigenfrequency (Hz)	Effective mass	Amplification	PSD Input g2/Hz	Notch level g2/Hz
1	326.2	0.099	30	0.2	0.0280
2	371.2	0.053	30	0.2	0.0522
3	595.2	0.0368	30	0.0903	0.0340
4	817.6	0.118	30	0.0479	0.0~~0~~56
5	892.1	0.038	30	0.0402	Covered by mode 4 notch
6	1360.8	0.036	30	0.0173	0.0066
Residual		0.0592			

Notched PSD interface loads

Equipment random test



Notched input spectrum in g2/Hz - 9.1 g rms

Equipment random test