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Procedia Structural Integrity 3 (2017) 176–190

Structural Integrity Procedia

www.elsevier.com/locate/procedia

XXIV Italian Group of Fracture Conference, 1-3 March 2017, Urbino, Italy

On field durability tests of mechanical systems. The use of the Fatigue Damage Spectrum

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Abstract

In the present paper the authors, starting from a previously proposed method for the combination and the synthesis of equivalent load conditions (by only managing PSD representations of the load conditions), developed a new approach based on the concept of Fatigue Damage Spectrum and on the system dynamics.
The proposed approach was then validated by a durability test case, in which two different acceleration motion based load conditions, a norm load condition (by using laboratory test) and an operative one (by using acceleration measurements acquired during an experimental activity conducted on a transport vehicle) were compared.

Keywords: Random loads, Accelerated tests, Fatigue Damage Spectrum, Vibration Fatigue

1. Introduction

In previous activities, conducted by the authors, the evaluation of damage generated by random loads (*i.e.* wind loads [Gong et al. (2914)], rail contact [Wolfsteiner (2013)], spindle forces [Decker et a. (2002)], proving grounds [Karlsson (2007)]) in generic mechanical system was analyzed by using and modifying consolidated approaches [Braccesi et al. (2010)] like the *Ashmore* one [Ashmore et al. (1992)].

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Peer-review under responsibility of the Scientific Committee of IGF Ex-Co.
10.1016/j.prostr.2017.04.034

Nomenclature

N	Number of cycle to failure
K	S-N Curve intercept
m	S-N Curve slope
$\Delta\sigma$	Stress amplitude
D	Fatigue damage
n	Applied loading cycle number
$f_\sigma(\Delta\sigma)$	Probability density function of stress amplitude
m_n	Central moment
$\Gamma(x)$	Gamma function
PSD_z	Output displacement power spectral density in z direction
$PSD_{\ddot{y}}$	Input acceleration power spectral density
$H(r_i)$	Frequency response function of elementary oscillator
f_n	Natural frequency of elementary oscillator
k	Stiffness of elementary oscillator
M	Mass of elementary oscillator
ξ	Damping ratio
Θ	Proportionality factor
PSD_σ	Power spectral density of stress
$D(f_n)$	Fatigue damage spectrum
T	Exposure duration
Q_ξ	Percentage displacement
$D_p(f_n)$	Potential damage
${PSD_{\ddot{y}}}^{eq}$	Equivalent accelerating and damaging power spectral density
T_{eq}	Duration of the accelerated test

The comparative evaluation of multiple load conditions and of their combination was managed by an evaluation of the potential damage of load cumulatives, evaluated either in frequency or in time domain, starting respectively from power spectral density (PSD) functions [Bendat et al. (1971)] and from time histories (TH). Actually, the previously proposed methodology for such evaluation, or for the combination and synthesis of equivalent load conditions, starts from a PSD representation of the load conditions [Braccesi et al. (2010)] and it is absolutely independent from the dynamic behavior of the excited system.

The reason, addressed in this work, that induced the authors to adopt and then compare the proposed method with the FDS (Fatigue Damage Spectrum [Svensson et al. (1993)]) approach starts from the assumption that there is a link between the cumulated damage and the natural frequency of the whole system or of structural components and consequently between cumulated damage and load conditions.

In this paper this request was satisfied for a particular class of systems and load conditions, that of mechanical systems subjected to motion based conditions. This class of load condition is very important for all mechanical systems (i.e. wind turbines [Fitzwater et al. (2001)], ball screws [Vicente et al. (1912)], landing gears [McGehee et al. (1979)]) and in particular for those that need experimental assessment tests related to transport load conditions (i.e. military and aeronautical applications). In such condition, the norms impose experimental assessment by test bench motion based acceleration inputs described by PSD functions related to specific exposure times, *i.e.* [MIL-STD-810F (1983)].

In the presented activity, firstly, a description of the stress-life fatigue (S-N) [Collins (1992)] is shown. Secondly, the concept of fatigue damage spectrum is introduced, then declined to the aim of this work.

A further aim achieved in this activity is the verification and the demonstration that, in case of a linear and single degree of freedom (*sdof*) type dynamic response of the structure or of a generic component and in case of two different but homogenous load conditions, the FDS proposed method obtains a global evaluation of the potential damage coincident with that obtainable by the previous approach [Braccesi et al. (2010)], but, in this case, under the hypothesis of dynamic amplification of the system response. So, a different formulation from that proposed in the past was developed with the idea to synthetize equivalent load conditions expressed in terms of PSD function, starting from arbitrary number of PSD functions and various exposure times.

This activity was then tested by a qualification test case for transport conditions, in which two different acceleration motion based load conditions, a norm load condition (by using test bench) and an operative one (by using acceleration measurements acquired during an experimental activity conducted on a transport vehicle) were compared.

2. Fatigue failure and damage evaluation in stress domain (S-N)

The fatigue strength of a mechanical component can be represented in the S-N domain by the following expressions: $N = K/\Delta\sigma^m$ or $\Delta\sigma^m = K/N$ where N is the number of cycle to failure, $\Delta\sigma$ is the applied stress amplitude, m the S-N curve slope and K a further constant which characterize the S-N curve.

It is possible to express the fatigue damage with the cumulative damage law of Palmgren-Miner [Collins (1992)] defined as follows:

$$D = \frac{n}{K}\Delta\sigma^m \tag{1}$$

where n is the applied loading cycles number related to a given applied stress amplitude.

For a random stress process, the fatigue damage can be expressed as [Premount (1994)]:

$$D = \frac{n}{K}E\{\Delta\sigma^m\} = \frac{n}{K}\int_0^\infty \Delta\sigma^m f_\sigma(\Delta\sigma)\, d\Delta\sigma \tag{2}$$

in which $f_\sigma(\Delta\sigma)$ represents the probability density function (PDF) of stress amplitude obtained with a counting method in time domain (*i.e.* Rain Flow Counting, RFC [Collins (1992)]) or through an analysis of the power spectral density function of the signal.

For the case of spectral analysis, the distributions (3) and (4) are referred to the formulation proposed by *Rayleigh* (3) and *Dirlik* (4).

$$f_\sigma(\Delta\sigma) = \frac{\Delta\sigma}{4m_0}e^{-\frac{\Delta\sigma^2}{8m_0}} \tag{3}$$

$$f_\sigma(\Delta\sigma) = \frac{\frac{D_1}{Q}e^{(-Z/Q)} + \frac{D_2 Q}{R^2}e^{-(Q^2/(2R^2))} + D_3 Z e^{(-Q^2/2)}}{2\sqrt{m_0}} \tag{4}$$

where:

$\gamma = m_2/\sqrt{m_0 m_4}$,
$Z = \Delta\sigma/\left(2\sqrt{m_0}\right)$,
$x_m = (m_1/m_0)\sqrt{m_2/m_4}$,
$D_1 = [2(x_m - \gamma^2)]/(1 + \gamma^2)$,

$R = \left(\gamma - x_m - D_1{}^2\right)/\left(1 - \gamma - D_1 + D_1{}^2\right)$,
$D_2 = \left(1 - \gamma - D_1 - D_1{}^2\right)/(1 - R)$,
$D_3 = (1 - D_1 - D_2)$,
$Q = 1.25 \cdot (\gamma - D_3 - D_2 R)/D_1$

and where the following equation (5) introduce the *n*-th spectral moment of the signal.

$$m_n = \int_0^\infty f^n G(f)\, df \tag{5}$$

Assuming that the stress cycle PDF, in the hypothesis of Gaussian process, is feasible with a *Rayleigh* (3) (*narrow-band* process, typical response of a *sdof*), then the expected value is given by (6):

$$E\,\{\Delta\sigma^m\} = b^m \Gamma\left(1 + \frac{m}{2}\right) \tag{6}$$

where b is defined as follow:

$$b = 2\sqrt{2\, m_0} \tag{7}$$

and m_0 is the zero-order spectral moment (*variance*) (5).
The operator Γ is the gamma function of Euler, defined as:

$$\Gamma(x) = \int_0^\infty t^{(x-1)}\, e^{-t} dt \tag{8}$$

Indeed, the fatigue damage can computed by the following equation:

$$D = \frac{n}{K} b^m \Gamma\left(1 + \frac{m}{2}\right) \tag{9}$$

3. Fatigue Damage Spectrum

It is important now to interpret the previous result in a general sense. In such way it is possible to evaluate the response of system subjected to an excitation condition (motion based) expressed in terms of acceleration PSD.

By reducing the considered system to an elementary oscillator (sdof), it is possible to compute the response of the system subjected to an acceleration input by the following formulation between the displacement z PSD of the sdof and the imposed acceleration $\ddot{y}$ PSD:

$$PSD_z(f_i, \xi, f_n) = PSD_{\ddot{y}}(f_i)\ |H(r_i)|^2 \tag{10}$$

$H(r_i)$ represents the frequency response function (FRF) of the elementary oscillator defined as follows:

$$H(r_i) = \frac{1}{(2\pi f_n)^2\sqrt{(1 - r_i^2)^2 + (2\xi r_i)^2}} \tag{11}$$

In Eq. (11), f_n is the natural frequency equal to $1/2\pi\sqrt{k/M}$ (with respectively k and M the stiffness and the mass of the *sdof*), ξ represents the damping ratio and r_i the ratio f_i/f_n .

Assuming a direct proportionality between the relative displacement z and the stress σ, in a generic point of the system, it is possible to write:

$$\Delta\sigma = \Theta \cdot z \tag{12}$$

where Θ is the proportionality factor between displacement and stress. Under this hypothesis, the following relation can be used:

$$PSD_{\sigma}\left(f_i,\xi,f_n\right)=\Theta^2 PSD_z\left(f_i,\xi,f_n\right) \tag{13}$$

Once the exposure time and the stress cycle probability density function (PDF) associated to the PSD_{σ} (*i.e.* to the elementary oscillator with natural frequency f_n) are known, the cumulable damage at each frequency can be computed as follow:

$$D\left(f_n\right)=\frac{n}{K}E\ \{^m\}=\frac{n}{K}\int_0^{\infty} {}^m f\ ()d \tag{14}$$

Once the $f\ ()$ is known by using *Rayleigh* (3) or *Dirlik* (4) formulation, it is possible to directly compute the fatigue damage spectrum by the previous equation.

In the hypothesis of *Rayleigh* cycles distribution, the cumulated damage $D\left(f_n\right)$ (15) can be obtained by equation (8):

$$D\left(f_n\right)=\frac{T\,f_n\Theta^m}{K}\left(\sqrt{PSD_z\left(f_n\right)}\right)^m\left(1+\frac{m}{2}\right) \tag{15}$$

where T is the spectrum application duration (exposure).

$D\left(f_n\right)$ represents the Fatigue Damage Spectrum (FDS). By the response spectrum of *Miles* [Miles (1954)] it is possible to obtain the following more common formulation for the cumulated damage:

$$D(f_n)=\frac{T\,f_n\Theta^m}{K}\left(\sqrt{\frac{Q_{\xi}\,PSD_{\ddot{y}}\left(f_n\right)}{2\left(2\pi f_n\right)^3}}\right)^m\left(1+\frac{m}{2}\right) \tag{16}$$

where Q_{ξ} represents the dynamic amplification factor, a different representation of the percentage damping:

$$Q_{\xi}=\frac{1}{2\xi}=\frac{\sqrt{k\,M}}{\xi} \tag{17}$$

A qualitative evaluation of damage, but always related to its absolute evaluation, can be obtained. This is represented by the potential damage D_p .

The term potential states that, in the evaluation process, the evaluable signals (i.e. accelerations) are not always directly linkable to the stress state, obtaining an unreal value (potential) of the damage. This, under the hypothesis of proportionality between stress and those signals (displacement/acceleration) (12), can be used to compare different behaviors in terms of their damaging potential.

Assuming an unitary value for the constant K of the S-N curve, for the Θ factor and for the exposure time T , it is possible to obtain (18).

$$D_p(f_n) = f_n \left(\sqrt{\frac{Q_\xi \, PSD_{\ddot{y}}(f_n)}{2(2\pi f_n)^3}} \right)^m \left(1+\frac{m}{2}\right) \tag{18}$$

This value is potentially storable in the time unit, meaning that it is independent to the time exposure.
The real FDS will be given from the following equation:

$$D(f_n) = \frac{T\,\Theta^m}{K} D_p(f_n) \tag{19}$$

By this formulation, the fatigue damage spectrum can be easily obtained by a simple product between the potential spectrum (valid for all systems with the same damping ratio and the same S-N curve slope) and the characteristics of mechanical component material and time exposure.

If the system response is evaluated in terms of relative displacement *z,* once the PDF of cycles of relative displacement $f_z(z)$ is known (by Rayleigh (3) or Dirlik (4) formulation), the following relation gives the potential fatigue damage spectrum in time unit:

$$D_p(f_n) = \frac{n}{T} E\,\{z^m\} = \frac{n}{T}\int_0^\infty z^m f_z(z)\,dz \tag{20}$$

In addition, it is possible to compute the real fatigue damage spectrum by (19).

4. Accelerated test

Once the real or potential frequency damage spectrums $D(f_n)$, $D_p(f_n)$ are known, by expliciting the input acceleration PSD function, from (16) and (18) it is possible to obtain the following relations

$$PSD_{\ddot{y}}^{eq}(f_n) = \frac{2(2\pi f_n)^3}{Q_\xi} \left(\frac{D(f_n)K}{T_{eq}\, f_n\, \Theta^m \left(1+\frac{m}{2}\right)} \right)^{2/m} \tag{21}$$

$$PSD_{\ddot{y}}^{eq}(f_n) = \frac{2(2\pi f_n)^3}{Q_\xi} \left(\frac{T}{T_{eq}} \frac{D_p(f_n)}{f_n \left(1+\frac{m}{2}\right)} \right)^{2/m} \tag{22}$$

These equations allow to directly obtain an acceleration input PSD function $PSD_{\ddot{y}}^{eq}$ related to an arbitrary exposure time T_{eq} and equivalent in terms of damage to the known FDS $D(f_n)$ or potential FDS $D_p(f_n)$ obtained for a known exposure time T. Usually this approach and these equations are used to accelerate the test by adopting an equivalent exposure time T_{eq} shorter than T.

Equation (22) is very interesting because it allows to compute the equivalent $PSD_{\ddot{y}}^{eq}$ just by the assessment of the S-N curve slope without the assessment of the proportionality factor Θ between stress and strain and of the second constant of the S-N curve.

By equations (21) and (22) and according to authors previous research activities [Braccesi et al. (2010)], it is possible to synthetize r random excitation conditions (i.e. highway, *Belgian* blocks) into only one input PSD function. Each i-th excitation is characterized by a power spectral density function $PSD_{\ddot{y}_i}\left(f_n\right)$ and by a given exposure time T_i.

By some substitution, the following equations allow to compute the equivalent input spectral density function, $PSD_{\ddot{y}}^{eq}$, associated to a desired exposure time T_{eq} :

$$PSD_{\ddot{y}}^{eq}\left(f_n\right)=\frac{2\left(2\pi f_n\right)^3}{Q_\xi}\left(\sum_{i=1}^{r}T_i D_{pi}(f_n)\right)^{2/m} \tag{23}$$

$$PSD_{\ddot{y}}^{eq}\left(f_n\right)=\frac{2\left(2\pi f_n\right)^3}{Q_\xi}\left(\frac{1}{T_{eq}\,f_n\left(1+\frac{m}{2}\right)}\sum_{i=1}^{r}T_i\left[f_n\left(\sqrt{\frac{Q_\xi\,PSD_{\ddot{y}_i}\left(f_n\right)}{2\left(2\pi f_n\right)^3}}\right)^m\left(1+\frac{m}{2}\right)\right]\right)^{2/m} \tag{24}$$

$$PSD_{\ddot{y}}^{eq}\left(f_n\right)=\left(\sum_{i=1}^{r}\frac{T_i}{T_{eq}}\left(PSD_{\ddot{y}_i}\left(f_n\right)\right)^m\right)^{2/m} \tag{25}$$

Equation (25) is similar to that obtained in [Braccesi et al. (2010)] (obtained without considering the dynamic amplification of the response of the system) but obtained by considering the system dynamics (*sdof* system with natural frequency f_n).

Such result is obvious under the hypothesis of linear behavior of the system.

5. Test Case

The considered test case concerns the durability test of a climatic chamber according to EN378–2:2008 norm [UNI, EN378–2 (2012)]. The experimental activity could not be carried out by the use of a shaking table due to dimensional limits. The available shaking systems (2.5 × 2.5 m) of SERMS, laboratory of University of Perugia (Italy), allow to perform vibrational tests, controlled in acceleration, in a frequency range of 5-3000 Hz up to a maximum force of 80 KN. Due to these features, the shaking table was not suitable with regards to the dimensions and inertia of the payload and with the acceleration regimes imposed by the above norm.

The acceleration input PSD function of the norm is shown in fig. 1.

In order to evaluate the component durability, a field tests activity (*i.e.* acceleration inputs induced by wheeled vehicle transport), has been accurately designed. Three tracks were chosen (fig.2), each one characterized by different roughness. For each one a specific vehicle speed was chosen. The most important characteristics of the three test tracks were reported in table 1.

The designed proving ground test wants to generate the same acceleration PSD input and the same damage of those that the norm laboratory test can induce.

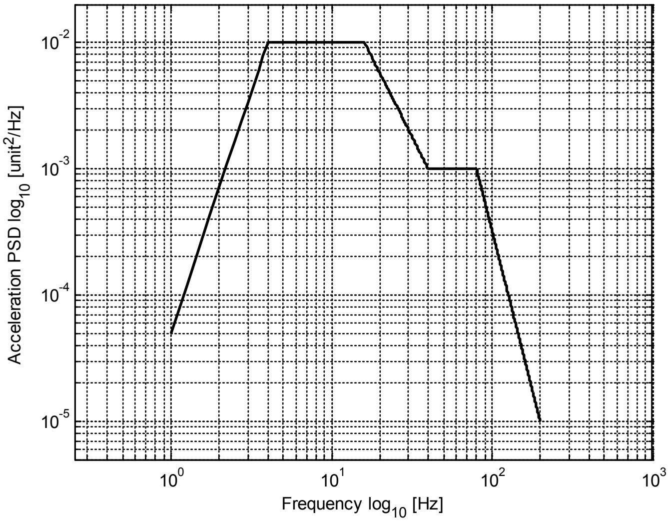

Fig.1. Input PDS function for the norm test

Tab 1. Track tests characteristics.

Track ID	Vehicle Speed [km/h]	Exposure [h:min]
1	30	0:44
2	40	0:31
3	50	1:18
Eqv	-	2:33

To evaluate if the test tracks can reproduce an equivalent condition to the norm spectra the vehicle was instrumented by uni and tri axial accelerometers (fig.3) mounted on the vehicle floor frame.

A lot of data were acquired in terms of acceleration time histories to the aim to compute the acceleration PSD functions that these conditions can apply to a generic payload (fig.4).

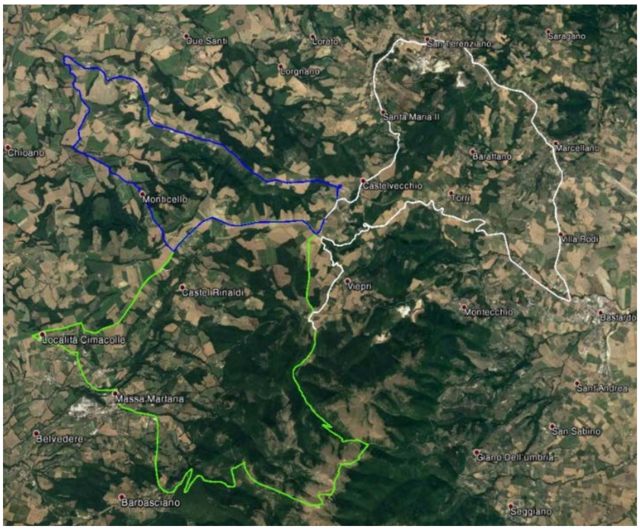

Fig.2: Geographical representation of the three considered tracks

Fig.3. Picture of one of the mounted accelerometers (triaxial no. MP01).

In figure 4, from a set of recorded acceleration time histories three windows of 50 seconds were extracted and displayed to show differences among test tracks in terms of induced acceleration.

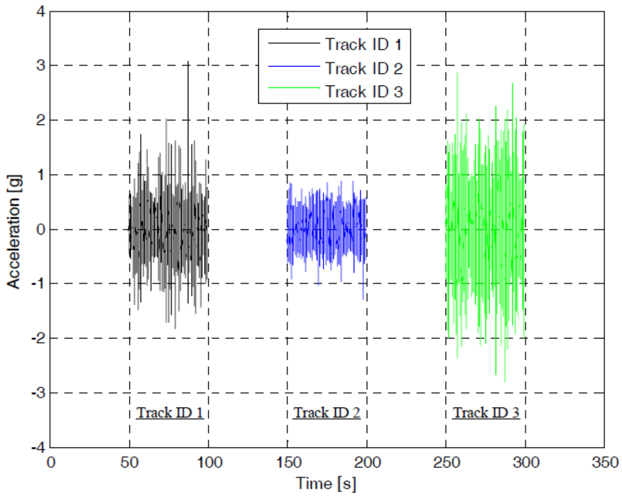

Fig.4. Time histories comparison among three road tracks (arbitrary windows of 50 s are considered for each track).

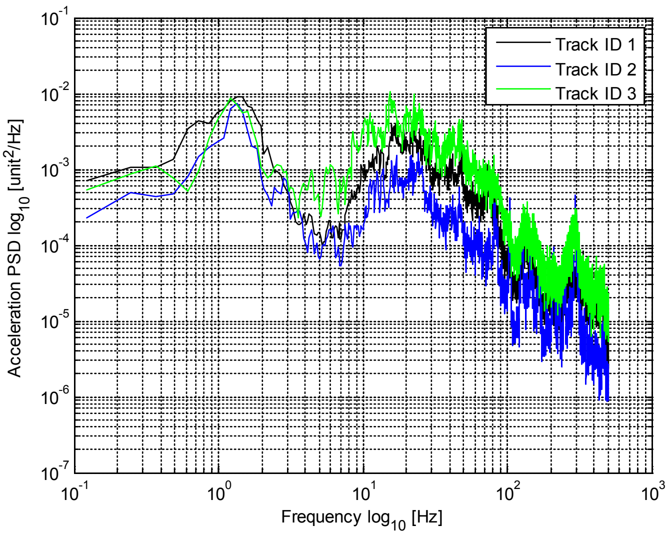

Fig.5. PSD functions comparison among three considered tracks.

In figure 5, for the same time histories, the PSD functions of the accelerations of figure 4 are shown.

The problem to solve was: "*how many laps of the whole track have to be performed to apply an equivalent acceleration input and an equivalent damage respect to those the norm induce ?*".

By adopting equation (25), assigning to the equivalent time length T_{eq} of table 1 (2 h and 33 min), for the three tracks (T_i) the corresponding test values (table 1) and considering $m = 4$, it is possible to obtain the equivalent PSD function, represented in figure 6 together with norm PSD one. Starting from classical techniques to generate time histories from given PSD functions [Braccesi et al. (2014)] it is possible to generate time histories of predetermined lengths both from norm PSD function and from equivalent one. To answer to the previous question were generated time histories of various length, representing just as various number of laps. In table 2 the time lengths of the hypothesized test runs are shown. For the norm test the standard indication of 2 h was adopted. For the equivalent road test three exposures were considered, 14 ($m = 4$), 35 ($m = 6$) and 100 ($m = 8$) h corresponding to 7, 15, 40 laps. If the first exposure (tab.2) is considered (14 h), the load spectra obtainable by the counting techniques previously introduced for norm input and from equivalent one are compared in figure 7.

Tab. 2. Results comparison in terms of normalized damage

	Norm Test	Eqv Road Test
$m = 4$		
Exposure	Norm 2 h	Road 14 h
Damage	1	1.23
$m = 6$		
Exposure	Norm 2 h	Road 35 h
Damage	1	1.08
$m = 8$		
Exposure	Norm 2 h	Road 100 h
Damage	1	1.08

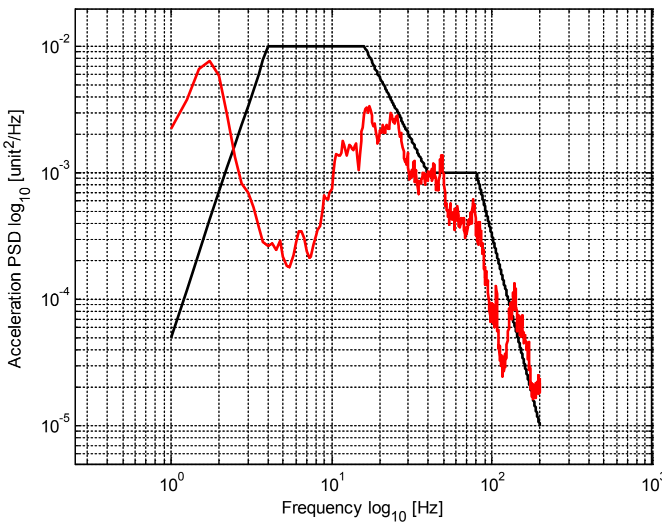

Fig.6. Input PDSs comparison between equivalent track PSD (red line) (25) and norm PSD (black line).

Figure 6 show that in frequency domain the two inputs are comparable from 20 Hz to the upper frequency limit of PSDs. At the low frequencies in the range from 1 to 3 Hz the equivalent input shows an higher power density than norm one, instead from 3 to 20 Hz the opposite condition is observable. These characteristics do not limit the usability of the equivalent on field test if a payload with a *rigid* behavior is considered or if its dynamics starts from 20 Hz. In these hypotheses the differences between the potential damage induced by the two input conditions of figures 6 and 7 could be eliminated by adopting different exposures. As it is observable in figure 7, to consider 14 h of exposure (equivalent track) generates a lot of acceleration cycles (10^7), more than those of norm test (10^6). This difference in terms of counted cycles regains that observable in terms of cycles amplitude that shows greater amplitudes for the norm test.

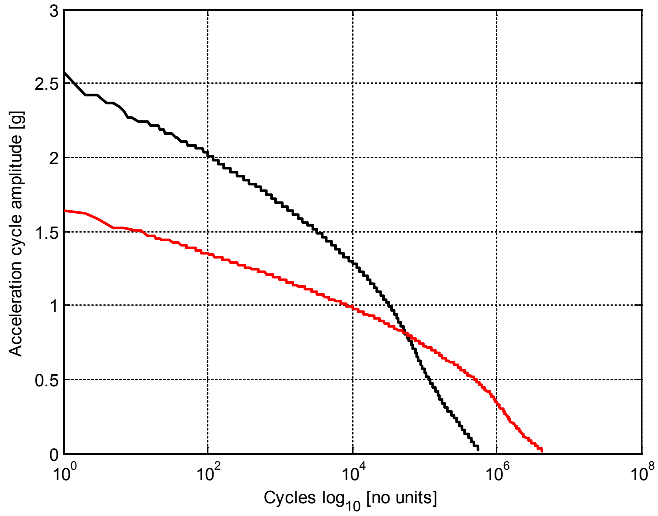

Fig.7. Cumulatives comparison (equivalent track, red line, norm test, black line), exposure times: 2 h for the norm test and 14 h for the equivalent track.

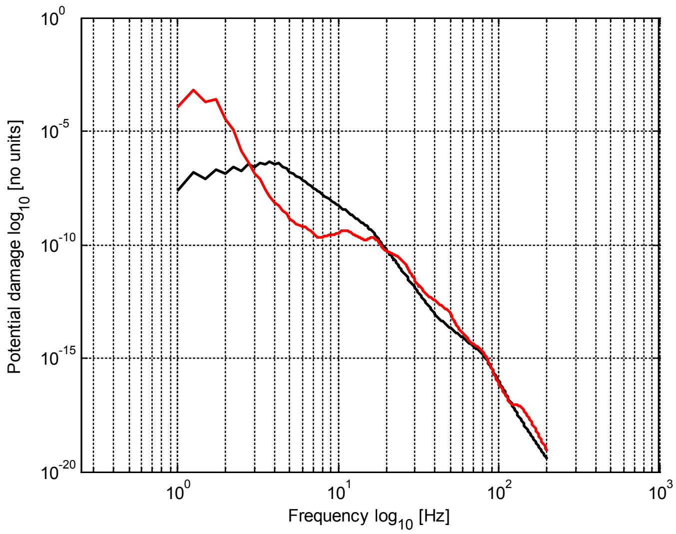

Fig.8. FDSs comparison (track test, red line, norm test, black line) with m=4, exposure times: 2 h for the norm test and 14 h for the equivalent track

In this case, considering $m = 4$, the two input conditions post processed as *stress* signals lead to the same potential damage D_p (see tab. 2). This affirmation is as more real as the dynamic behavior of the component is negligible during the test. The potential damage values shown in table 2 are normalized to that obtained by post processing norm test input signal and assigning an unitary value to S-N parameter K .

First of all it is important to highlight the importance of S-N slope in the damage evaluation. Values of 4, for metallic components in case of medium notch (the maximum value of notch are obtained from steel component with a value of 3), until values of 8, for components without notches and with low applied stress, are generally used [Braccesi et al. (2010), Ashmore (1992), MIL-STD-810F (1983)]. In table 2 is evident the influence of m in designing the equivalent track length if the aim is to obtain the same potential damage of norm test (Eqv Road/Norm Ratio equal to 1). Similar damages are obtainable by considering $m = 4$ and approximately 7 laps (exposure of 14 h), $m = 6$ and approximately 15 laps (exposure of 35 h) and $m = 8$ and approximately 40 laps (exposure of 14 h).

If system dynamics is considered, the potential fatigue damage spectrum can help to verify above affirmations. In figure 8 the potential fatigue damage spectra of equivalent road test (14 h) and of norm one are compared ($m = 4$). The representation of potential damage in frequency domain gives not a single value of potential damage of the component (table 2) but a potential damage value for each frequency f_i, that is, for each component that has that as its natural frequency $f_n = f_i$.

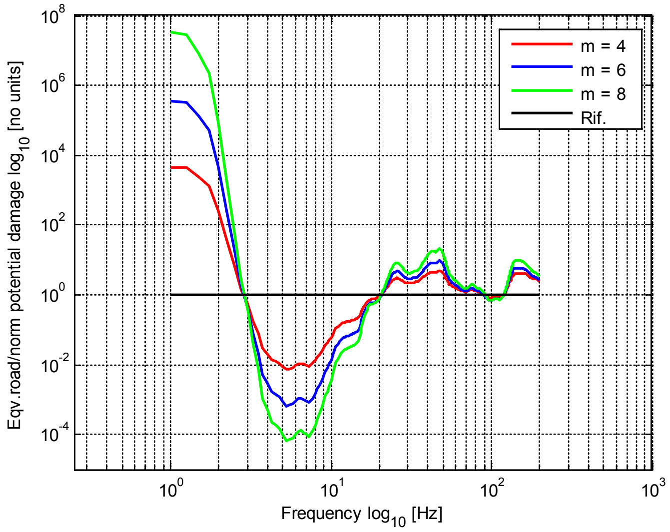

Fig.9. Comparison between the FDS variation with respect to the reference (Rif., Norm) with variation of m and of the exposures (see tab.2)

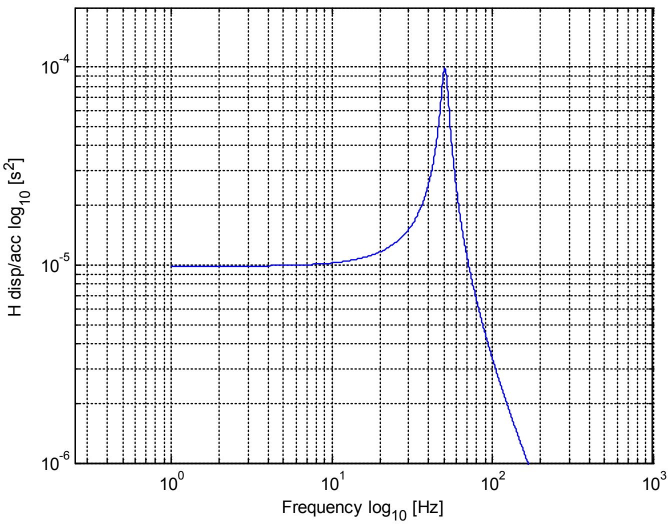

Fig.10. Frequency response function: relative displacement/base acceleration for a sdof with natural frequency equal to 50 Hz and ξ equal to 0.05

In figure 9, the potential FDSs obtained as ratio between FDS of the equivalent road test conditions (table 2) and that of the reference one are shown by considering a damping ration $\xi = 0.05$.

Consider a component represented by a *sdof* with natural frequency f_n equal to 50 Hz and a damping ratio ξ equal to 0.05. In figure 10 its frequency response function (FRF) in terms of displacement/acceleration $H(r)$ (11) is shown.

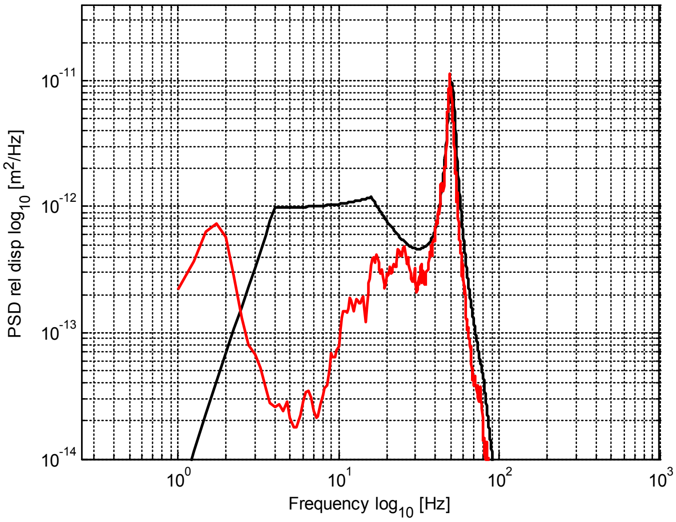

Fig.11. Comparison between the response in terms of PSD functions of relative displacement for a sdof with natural frequency equal to 50 Hz and ξ equal to 0.005. In black the response to norm excitation and in red the response to equivalent track excitation

The PSD functions of relative displacements obtained as outputs of equation (13) are shown in figure 11, obtained by considering as inputs the norm PSD function and that of equivalent road.

In fig. 12 the cumulatives associated to the above PSDs and in particular coming from the first proposal of equivalent tracks of table 2 (exposure of 14 h) and norm are shown.

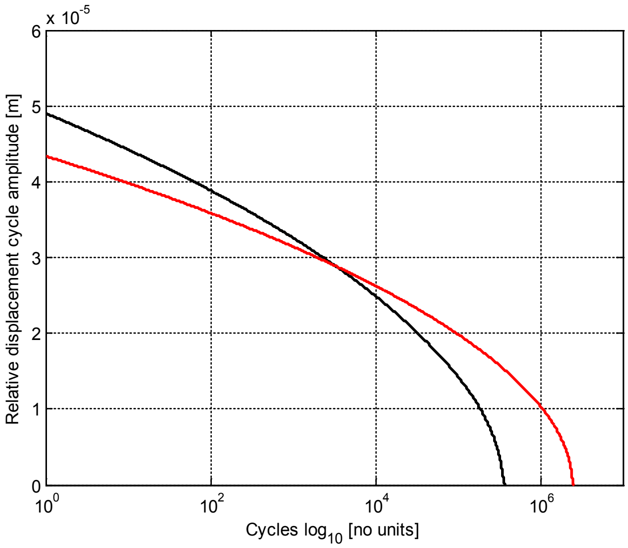

Fig.12. Comparison between relative displacement cumulatives for a sdof with natural frequency equal to 50 Hz and ξ equal to 0.005. In black the response to the norm excitation (2 h of exposure) and in red the response to equivalent track excitation (14 h of exposure)

Finally, in figure 13 the relative potential fatigue damage values are plotted by markers over the previously obtained FDS (fig.8) evaluated for a S-N curve with slope $m = 4$ and a damping ratio ξ equal to 0.05.

This test case demonstrates how the FDS approach could be useful to design test conditions, off laboratory standards, as well as to accelerate tests or to monitor loading conditions and damage evolution in situ (i.e. wind turbines). In particular, if the system or the components dynamics is known it is possible to foresight the error that could be done by adopting the designed test.

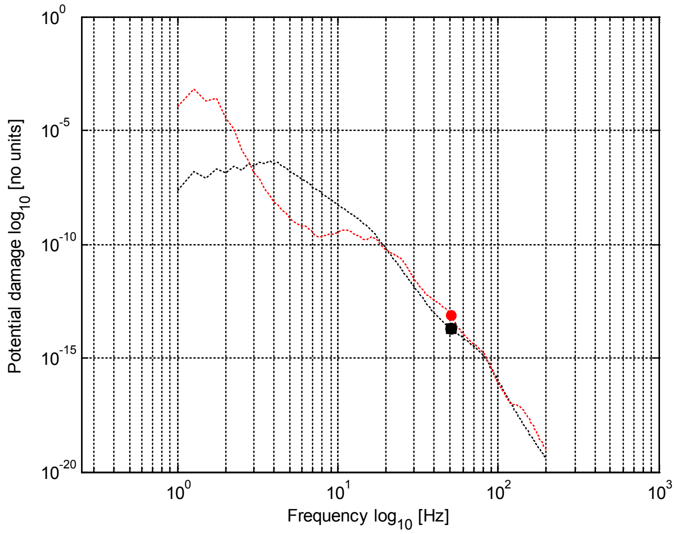

Fig.13. Comparison between the potential damage values with m= 4, for a sdof with natural frequency equal to 50 Hz and ξ equal to 0.005. In black the response to the norm excitation (2 h of exposure) and in red the response to the equivalent track excitation (14 h of exposure)

6. Conclusion

The developed approach, based on FDS, demonstrates how the dynamics can influence the combination of multiple loading conditions into an equivalent one, how it can influence the synthesis of accelerated tests and how, by FDS itself, it is possible to take into account it. It is an alternative view of the same problem the authors analyzed and observed in the past without assuming a dynamic behavior of the payload.

The test case shown in the paper demonstrates instead how the FDS approach could be useful to design test conditions, off laboratory standards, as well as to accelerate tests when test laboratory (i.e. MIL standard test by shaking table) cannot be performed.

Acknowledgements

This research activity was partially financed by Italian PRIN funding source (Research Projects of National Interest - Progetti di Ricerca di Interesse Nazionale) by a financed project entitled SOFTWIND (Smart Optimized Fault Tolerant WIND turbines).

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