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An approach for comparing in-service multi-input loads applied on non-stiff components submitted to vibration fatigue

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Abstract

We focus on applications from the automotive industry, on mechanical components submitted to vibration loads. On one hand, the characterization of loading for dimensioning new structures in fatigue is enriched and updated by customers data analysis. In a second hand, the loads characterization also aims to provide robust specifications dedicated to the simulation or test rigs. We aim to provide vibration specifications that are adapted to a calculation time or physical test durations in accordance with the pace imposed by the projects timeframe. The vibration specifications need to be robust by taking care of the diversity of vehicles and markets considered in the projects. In the trucks industry, the dynamic behavior can vary significantly from one configuration to another. For non-stiff structures, the lifetime depends, among other things, on the frequency content of the loads, as well as the interactions between the components of the multi-input loads. In this context, this paper proposes to compare sets of multi-input loads applied on the non-stiff structure, i.e for which the frequency content of loads impacts the damage.

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Keywords: Vibration fatigue, mulnit-input in-servie loads, fatigeu damage spectra, maximum response spectra

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1. Introduction

Durability feature aims to understand and predict failure modes and fatigue phenomena in structures submitted to in-service loads, that are constant or variable amplitude loads. This paper focus on application from automotive industry, especially trucks. We concentrate on load vibration applied on mechanical components.

One of the targets of the loads characterization is to capture damaging events and model them (Johannesson & Speckert, 2014). Loads model is necessary for different reasons: to adapt the rig capability where the components is tested, to understand the scatter of a population of customers, to compare them.

The trucks architecture impacts the load environment of components of the truck (e.g chassis packaging hung on the chassis). The manufacturers aim to get common structures among different trucks silhouettes. For dimensioning new structures, loads from different trucks need to be compared, in order to identify the most severe truck configuration.

An approach to compare variable amplitude multi-input loads applied on non-stiff structures is proposed. After the method description, a two-input example is presented. A cooling package installed on different trucks architecture is used as an illustration.

2. Loads modeling for robust life predictions and endurance rig tests

2.1. Different aspects of loads modeling

Markets or population of markets are usually characterized by the road conditions, driving environment, mission profiles. Measured in-service loads allow us to constantly check the consistency of our proving grounds and durability obstacles, regarding the vibration level our trucks undergo during their life.

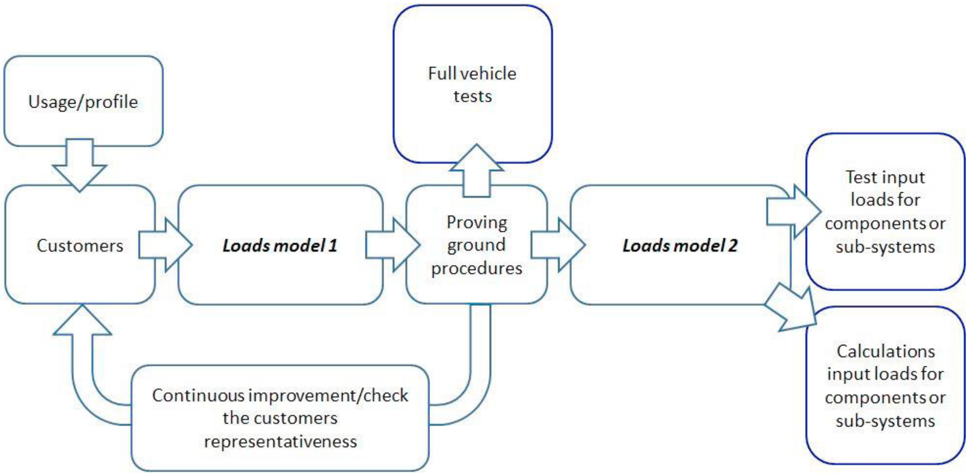

Figure 1: Different aspects of loads modeling

Lower test costs goes together with robust life time prediction. Robustness in simulation needs accurate loads and complex models including environmental parameters. This goes together with powerful calculation capability to get reasonable calculation time.

For non-stiff component rigs, Electro-Dynamic Shaker tests or shake table are used. Input loads are described with power spectral densities. Each direction is applied on the specimen sequentially. These PSD need to be representative to the measured in-service loads, in terms of damage. They are analyzed in each direction, independently of the others: the coupling is usually not considered. In order to lower the test duration, they are accelerated as well. They are determined with, e.g, the tayloring approach. Limitations regarding the representation of singularities like peaks are met in some applications (Colin, 2016).

For static or quasi static structures, equivalent loads are usually helpful to describe univariate or multivariate loads (Bignonnet & Thomas, 2001) (Le Corre (Genet), 2006). Several proposals have been done to transform complex multivariate loads into deterministic loads and with random processes. The transformed loads need to be equivalent in terms of damage, whatever the structure they are applied on.

2.2. Loads input dependent on the customers usage and the architecture of the truck

Communality of components over different trucks architectures helps to make a substantial reduction of cost. The validation plan focuses on the most severe trucks specifications. A difficulty is to identify which silhouette is the most severe one and for which component.

Trucks architectures need to be sorted by the level of severity, depending on the focused component or area on the trucks, on the failure mode. This can be done with the help of the experience and the background of the project team accumulated from previous projects on close products. We also need numerical criteria to decide which truck architecture is the most severe one compared to others.

For non-stiff structures, loads are analyzed in damage and by frequency. Fatigue Damage Spectra (FDS) and Maximum Response Spectra (MRS) (Halfpenny, Investigation of the durability transfer concept for vehicle prognostic applications, 2010) (Halfpenny, Methds for accelerating dynamic durability tests, 2006) are usually appropriate. FDS and MRS are computed from a one-input acceleration.

In order to compare multivariate loads sets applied on a common non-stiff structure, this can be trickier to handle, as the coupling between the loads input sometimes impacts the damage. A proposal to compare multi-input loads in the frequency domain is described in the following parts. The comparison is done in terms of damage.

2.3. Target and hypotheses,

The target of the following paragraph is to express the stress from a multi-input acceleration applied on a non-stiff structure. We will work under the following hypothesis.

Let's consider a structure called (S),

- H1 : The behavior of (S) is linear,
- H2 : (S) is considered as non-stiff, i.e. the damage depends on the loads frequency.
- H3 : A frequency f0 emerges from the other eigen modes, and is the most damaging at the critical point. The frequency f0 highly depends on the dynamic characteristics of (S). The stiffness of the chassis, where the cooling package is mounted on, has a limited impact on f0.
- H4 : At the critical point, we consider that the stress tensor is mainly unidirectional. The direction of the unidirectional stress is the same, whatever the loads.

2.4. Expression of the uniaxial stress component from the acceleration tensor

Let's call A(t), the acceleration set,

$$A(t) = (a_u(t)) \qquad \text{Eq. 1}$$

The integer value u is described as $1 \leq u \leq U$, where U is the number of loads applied on the structure. In the principal stress axis system, the stress tensor at the critical point A_c of a structure S is expressed as follows,

$$\mathbf{\Sigma}\big(A_c, A(t)\big) = \sigma\big(A_c, A(t)\big)\begin{pmatrix}1\\0\\0\end{pmatrix}$$

Considering the modal superposition method, the unidirectional stress time history at the critical point can be written as follows,

$$\sigma\big(A_c, A(t)\big) = \sum_{i=1}^{m} \sigma_i\,(A_c).\,q_i(A(t))$$

The magnitudes σ_i are the modal stresses. They are dependent on the structure, its material and its geometry at the location of the critical point. The magnitudes q_i are the modal coordinates. They are dependent on the structure as well as on the acceleration inputs. The modal coordinates are time series. The magnitude m is the total number of modes taken into account to calculate σ with the modal superposition method.

By the hypothesis called H3, the resonance frequency is at the origin of most of the damage. Let's consider,

$$\tilde{\sigma}\big(A_c, A(t)\big) = \sigma_{iR}(A_c).\,q_{iR}(A(t))$$

The magnitude σ_{iR} is the modal stress and q_{iR} is the modal coordinate at the resonance frequency. We consider the following approximation, under the hypothesis described in the paragraph 2.3,

$$\sigma\big(A_c, A(t)\big) \sim \tilde{\sigma}\big(A_c, A(t)\big) \qquad \text{Eq. 2}$$

From the hypothesis previously mentioned, the modal coordinates can be expressed in terms of accelerations, from Eq. 1, as follows,

$$\tilde{\sigma}\big(A_c, A(t)\big) = \sigma_{iR}(A_c).\sum_{u=1}^{U} b_u(A_c)\,a_u(t)$$

The magnitude b_u depends on A_c,

$$\tilde{\sigma}\big(A_c, A(t)\big) = \sigma_{iR}(A_c).\,B(A_c)\sum_{u=1}^{U} \alpha_u(A_c)\,a_u(t)$$

where,

$$\sum_{u=1}^{U}(\alpha_u(A_c))^2 = 1 \text{ and } B(A_c) > 0$$

Let's consider the magnitude a^*,

$$a^*(A(t), A_c) = \sum_{u=1}^{U} \alpha_u(A_c)\,a_u(t)$$

Thus,

$$\tilde{\sigma}\big(A_c, A(t)\big) = \sigma_{iR}(A_c).\,B(A_C).\,a^*(A(t), A_c) \qquad \text{Eq. 3}$$

The magnitude B depends on the structure and the critical point. The magnitude a* can be described as a linear combination of the acceleration components. The unit vector α_u depends on the structure. The product $\sigma_{iR}(A_c).\,B(A_C)$ is an amplification factor, dependent on the structure and the location of the critical point.

3. An application of a cooling package mounted on the chassis of a truck

The aim of this paragraph is to illustrate the method with an application: the cooling package mounted on a truck. The target is to compare two sets of accelerations in terms of damage and frequency.

3.1. Non stiff structure submitted to variable amplitudes accelerations

The cooling package is placed at the front of the chassis, between the two siderails, as illustrated in the picture, below. The siderails are the two C-beams. The assembly of the cooling package in the chassis is illustrated below.

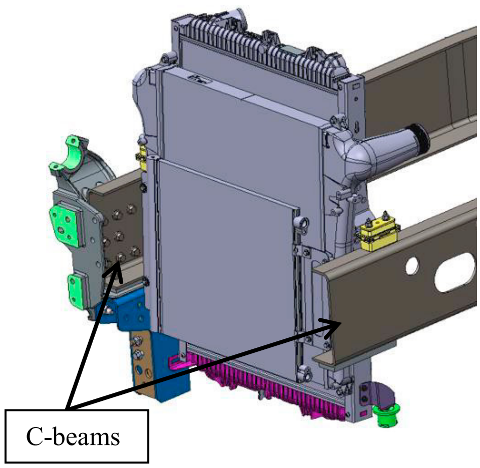

Figure 2: An illustration of the cooling package assembly on the chassis (C-beams, made of steels)

The mechanical behavior of the assembly can be represented as shown in the drawing below. The cooling package is characterized by its mass, stiffness and damping. It is placed on rubber bushings, illustrated by springs of the drawing. Acceleration inputs are the tensors $A_{RHS}(t)$ and $A_{LHS}(t)$.

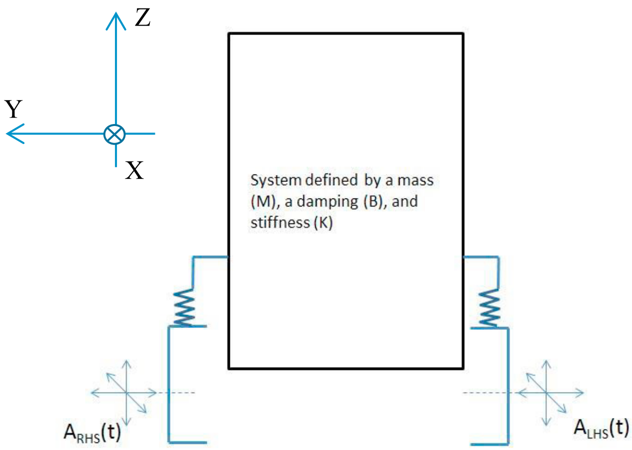

Figure 3 : The cooling package submitted to input accelerations from the chassis.

The cooling package is usually tested on a shake table, with frequency spectra described by the manufacturer, measured on trucks driven on the proving ground. The input accelerations needed for the shake table are usually measured at the side rails.

During development loops, the supplier is responsible for developing the technology for the cooling package. In our case, the trucks manufacturer is responsible for developing the brackets and bushings, in order to mount the CP on the truck. He is also responsible for providing the appropriate accelerations for both simulation activities and physical tests. The key point is to ensure that the provided accelerations sets are the most severe ones, among the diversity of architectures of trucks covered by the project.

For a given cooling package on a truck, the accelerations A_{RHS} and A_{LHS}, as illustrated in the picture above, can be described with its three-input component, as follows,

$$A_{RHS}(t) = \begin{pmatrix} a_{RHS}^{x}(t) \\ a_{RHS}^{y}(t) \\ a_{RHS}^{z}(t) \end{pmatrix}$$

$$A_{LHS}(t) = \begin{pmatrix} a_{LHS}^{x}(t) \\ a_{LHS}^{y}(t) \\ a_{LHS}^{z}(t) \end{pmatrix}$$

Components of acceleration tensors are time series. The cooling package excited by the accelerations A(t) is (S). Let's consider the critical point A_c on (S). For the sake of simplicity, we consider the following notation,

$$A(t) = (a_u(t))$$

The integer value u is described as $1 \leq u \leq U$, where U is the number of loads components applied structure, i.e 6 in this application.

Let's consider two accelerations tensors A_I and A_{II}, measured on two trucks configurations.

$$A_I(t) = (a_{I,u}(t))$$

$$A_{II}(t) = (a_{II,u}(t))$$

The same cooling package is installed on these two trucks. The cooling package fulfills the hypothesis presented in the paragraph 2.3. If S undergoes $A_I(t)$,

$$\tilde{\sigma}(A_c, A_I(t)) = \sigma_{iR}(A_c).B(A_c).a^*(A_I(t), A_c)$$

with

$$a^*(A_I(t), A_c) = \sum_{u=1}^{U} \alpha_u(A_c)\, a_{I,u}(t)$$

If S undergoes $A_{II}(t)$,

$$\tilde{\sigma}(A_c, A_{II}(t)) = \sigma_{iR}(A_c).B(A_C).a^*(A_{II}(t), A_c)$$

With

$$a^*(A_{II}(t), A_c) = \sum_{u=1}^{U} \alpha_u(A_c)\, a_{II,u}(t)$$

The difference between A_I and A_{II} in terms of severity can be deduced from the fatigue damage spectra comparison of the linear comparison $a^*(A_I(t), A_c)$ and $a^*(A_{II}(t), A_c)$. The linear combination described by the coefficient α_u is similar for both $a^*\big(A_I(t)\big)$ and $a^*\big(A_{II}(t)\big)$.

- If the geometry is known, the coefficients α_u are known, as well as f0. Thus, we can compare the FDS of the linear combinations $a^*(A_I(t), A_c)$ and $a^*(A_{II}(t), A_c)$, around f0.
- If the geometry is not known, neither the coefficients α_u nor the eigen frequency f0 are. In order to compare the severity of $A_I(t)$ and $A_{II}(t)$, different linear combinations of α_u are studied, randomly and uniformly distributed.
- From the experience of the designers, the interval $[f_{min}, f_{max}]$, where the natural frequency most probably is, is known. The analysis can be reduced to this interval.

3.2. Comparison of two sets of two-input accelerations in terms of damage

Let's consider the vertical accelerations acting on the cooling package,

$$\boldsymbol{A_{RHS}}(t) = \begin{pmatrix} 0 \\ 0 \\ a_{RHS}^z(t) \end{pmatrix}$$

$$\boldsymbol{A_{LHS}}(t) = \begin{pmatrix} 0 \\ 0 \\ a_{LHS}^z(t) \end{pmatrix}$$

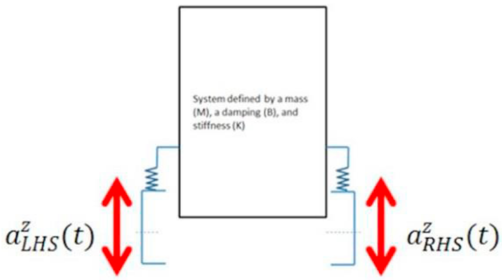

Figure 4: A cooling package submitted to vertical accelerations only

Then,

$$A(t) = (a_u(t))$$

With,

$$a_1(t) = a_{LHS}^z(t)$$

$$a_2(t) = a_{RHS}^z(t)$$

Two sets of accelerations have been measured on two different trucks, with the same cooling package. The target is to know which one is the most severe, regarding the studied cooling package.

The linear combination $a^*(A(t),A_c)$ can be written as follows,

$$\mathrm{a}^*(\mathrm{A}(\mathrm{t}), A_c) = \sum_{\mathrm{u}=1}^{2} \alpha_{\mathrm{u}}(A_c)\, \mathrm{a}_{\mathrm{u}}(\mathrm{t}) \qquad \text{Eq. 4}$$

with,

$$\sum_{\mathrm{u}=1}^{2} \left(\alpha_{\mathrm{u}}(A_c)\right)^2 = 1$$

Let's consider the following expression for the parameters α_u :

$$\alpha_1(\mathrm{A}_C) = \cos(\theta(\mathrm{A}_C))$$
$$\alpha_2(\mathrm{A}_C) = \sin(\theta(\mathrm{A}_C)) \qquad \text{Eq. 5}$$

with $0 \leq \theta(\mathrm{A}_C) < \pi$.

A cooling package geometry that undergoes the acceleration A(t), is characterized by a parameter $\theta(\mathrm{A}_C)$ at the critical point.

From the equation Eq. 3, the uniaxial stress at the critical point depends on a*. The target is to compare the damage of two sets of accelerations, applied on the same cooling package. The acceleration set A_I is more severe than the acceleration set A_II if and only if the damage induced by $\tilde{\sigma}\left(A_c, A_I(t)\right)$ is greater than $\tilde{\sigma}\left(A_c, A_{II}(t)\right)$. Thus, the following conclusion can be drawn: the acceleration set A_I is more severe than the acceleration set A_II if and only if the damage computed from $\mathrm{a}^*\left(A_c, A_I(t)\right)$ is greater than $\mathrm{a}^*\left(A_c, A_{II}(t)\right)$.

3.3. An example with sets of measured accelerations

An example is illustrated below. A given cooling package is mounted on two different trucks, called KM-2 and KM-82. The KM-2 is a tractor 6x4, dedicated to construction usage. The truck KM-82 is a construction truck as well, a rigid 8x8.

$$A_I(t) = A_{KM-2}(t)$$

$$A_{II}(t) = A_{KM-82}(t)$$

Accelerations measurements are illustrated in the pictures below. Vertical accelerations were measured at the siderails, on the corrugation obstacles (washboards), on the proving ground.

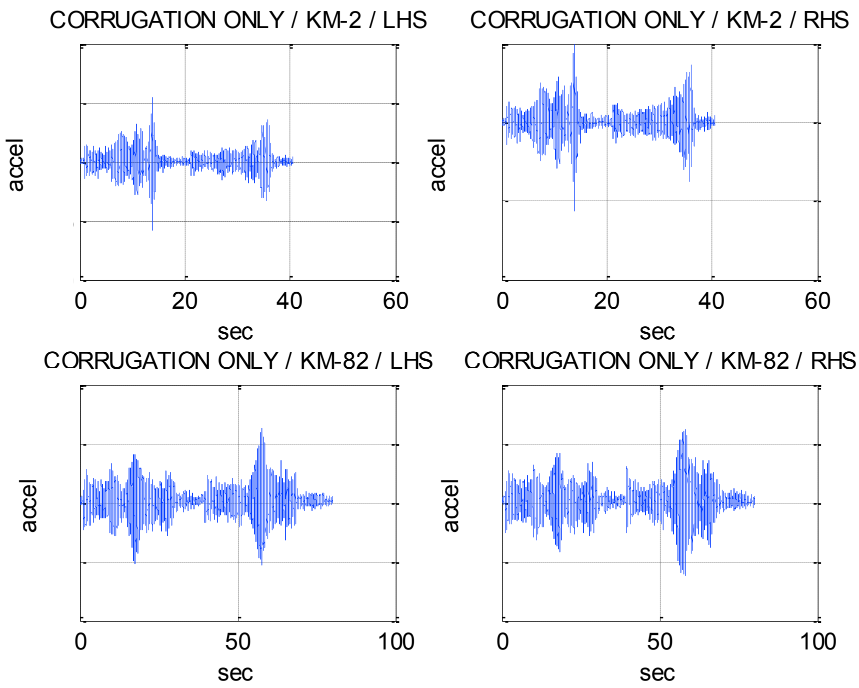

Figure 5: Illustration of measured accelerations time series, in g, for both trucks KM-2 and KM-82, left hand side and right hand side.

From Eq. 4 and Eq. 5, four values of θ are considered, evenly spread in the interval [0,π]. The fatigue damage spectra and maximum response spectra are presented in the graphs below.

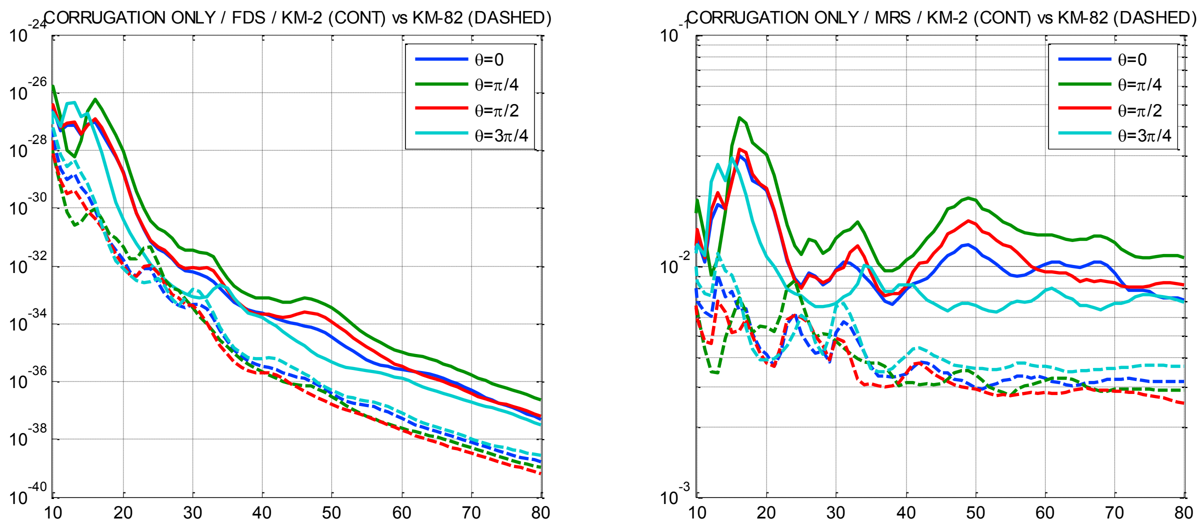

Figure 6: Fatigue Damage Spectra (left hand side) and Maximum Response Spectra (right hand side). Basquin slope : 5. Software : Matlab R2012b. Data from the KM-2 are drawn with continuous lines, data from the KM-82 are drawn with dashed lines.

Different points can be highlighted from this application:

- The KM-2 seems to be more severe than the KM-82, for the cooling packages with a natural frequency f0 above 32Hz.
- Below this frequency, the conclusion regarding the severity of A_I and A_{II} depends on θ and on the frequency f0. More details about the geometry of the structure at the critical point, e.g with the help of finite element analysis, are required, in order to go further in the analysis.

Along the development loops and the maturity level of the new design, the level of the details is higher. In collaboration with the durability analysts working in durability simulation, the severity comparison gains in precision and robustness. The choice of multivariate input loads for the shaker tests is relevant, the level of confidence increased.

The complexity of the implementation and the analysis increases when considering the full accelerations vector applied on the cooling package, i.e six components. Unit vectors can be randomly generated with a calculation tool. The analysis is easier as soon as the level of knowledge regarding the cooling package is higher: the natural frequency as well as predominant accelerations components impacting the damage.

4. Conclusions

Comparing multi-input vibration loads by their level of damage on non-stiff components is essential for different aspects: to build test specification components that are common for different truck architectures and vibration environments, to compare customers severity and get more knowledge for future measurements, to better evaluate the risk of failure.

Main difficulties can be highlighted. The damage depends on the amplitude of loads and its frequency content. The damage is dependent on multi-input loads applied on the components. Most of the time, especially during front loading phases of a project, the component is not known. The loads severity needs to be evaluated with limited information about the component.

In this paper, a method for comparing multi-input accelerations applied on non-stiff components regarding the damage, has been proposed. Hypothesis linked to its mechanical behavior were taken. Results on a cooling package application have been presented in terms of fatigue damage spectra for different linear combinations of the multi-input accelerations. Results analysis can be quite tricky to handle, especially if the order of the main natural frequency is unknown. Hopefully, the order of magnitude of the first natural mode does not change drastically, from one cooling package to another. A two-input loads has been presented. The results representations and analysis get harder if the number of acceleration directions increase.

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