

Comparing in-service multi-input loads applied on non-stiff components submitted to vibration fatigue to provide specifications for robust design

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Abstract. This study focuses 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 customer data analysis. On the other hand, the loads characterization also aims to provide robust specifications for simulation or physical tests. These specifications are needed early in the project, in order to perform the first durability verification activities. At this time, detailed information about the geometry and the material is rare. Vibration specifications need to be adapted to a calculation time or physical test durations in accordance with the pace imposed by the projects timeframe. In the trucks industry, the dynamic behaviour can vary significantly from one configuration of truck to another, as the trucks architecture impacts the load environment of the components. The vibration specifications need to be robust by taking care of the diversity of vehicles and markets considered in the scope of the projects. 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 an approach to compare sets of variable amplitude multi-input loads applied on non-stiff structures. The comparison is done in terms of damage, with limited information on the structure where the loads sets are applied on. The methodology is presented, as well as an application. Activities planned to validate the methodology are also exposed.

1 Introduction

Early in development projects, loads are needed to enrich the first simulations on the first concepts. At that time, little information on the new concepts, regarding geometry and material, is known (Johannesson & Speckert, 2014). For static or quasi static structures, equivalent loads are usually helpful to describe univariate or multivariate loads (Bignonnet & Thomas, 2001) (Le Corre (Genet), 2006). For non-stiff components, input loads can be described with power spectral densities. Each direction is applied on the specimen sequentially. Limitations regarding the representation of singularities like peaks are met in some applications (Colin, 2016).

This paper deals with loads coming from the roads, applied at different points of a structure placed on trucks, attached to the chassis. This structure is considered as non-stiff, i.e the damage induced by these loads is dependent on the frequency content of these loads.

In order to assure the robustness of the loads, they have to be measured on the most damaging configuration of trucks, among the full scope of trucks configurations in the project. The loads can be measured on a physical truck, when driving on the obstacles of the proving ground. They can also come from simulated trucks, virtually driving on numerical obstacles of the proving ground.

In this context, a tool is needed to identify the most severe trucks configurations, in terms of damage applied on component of interest. This needs to be done without detailed information about the geometry and the material

of the structure itself. The theory is detailed in the following paragraphs, under several assumptions on the structure. The application chosen here is the cooling package, placed at the front of the truck, in the chassis.

2 Expression of the stress from the multi-input accelerations

This paragraph is dedicated to the expression of the stress tensor from acceleration components, under several working hypotheses.

2.1 Hypotheses and target

The aim is to express the stress from a multi-input acceleration applied on a non-stiff structure. The following hypotheses are taken into account.

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 f_0 emerges from the other eigen modes, and is the most damaging at the critical point. The frequency f_0 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 f_0 .
- **H4:** At the critical point, the stress tensor is mainly unidirectional. The direction of the

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unidirectional stress is the same, do not vary with time.

2.2 Expression of the uniaxial stress

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

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

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

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

where $\sigma(X_c, A(t))$ is a scalar. Considering the modal superposition method, the unidirectional stress time history at the critical point can be written as follows,

$$\sigma(X_c, A(t)) = \sum_{i=1}^m \sigma_i(X_c) \cdot q_i(A(t))$$

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 the stress 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,

$$\tilde{\sigma}(X_c, A(t)) = \sigma_{iR}(X_c) \cdot q_{iR}(A(t)) \quad (1)$$

The magnitude σ_{iR} is the modal stress and q_{iR} is the modal coordinate at the resonance frequency. The following approximation is considered, under the hypothesis described in the paragraphs above,

$$\sigma(X_c, A(t)) \sim \tilde{\sigma}(X_c, A(t)) \quad (2)$$

From the hypothesis previously mentioned (H1), the modal coordinates can be expressed in terms of accelerations, from (2) **Error! Reference source not found.**, as follows,

$$\tilde{\sigma}(X_c, A(t)) = \sigma_{iR}(X_c) \cdot \sum_{u=1}^U b_u(S) a_u(t)$$

In order to make the expression of $\tilde{\sigma}$ more tractable mathematically, the following normalization has been chosen,

$$\tilde{\sigma}(X_c, A(t)) = \sigma_{iR}(X_c) \cdot B(S) \sum_{u=1}^U \alpha_u(S) a_u(t)$$

where,

$$\sum_{u=1}^U (\alpha_u(S))^2 = 1 \text{ and } B(S) > 0 \quad (3)$$

Let's consider the magnitude a^* ,

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

Thus,

$$\tilde{\sigma}(X_c, A(t)) = \sigma_{iR}(X_c) \cdot B(X_c) \cdot a^*(A(t)) \quad (3)$$

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 $\sigma_{iR}(X_c) \cdot B(X_c)$ is an amplification factor, dependent on the structure and the location of the critical point.

3 Applications

The aim of this paragraph is to use the theoretical background explained above to rank different trucks in terms of damage. The main inputs are the accelerations measured at the attachment points of (S) to its holder. Let's recall that the geometry of (S) is unknown. However, the four hypotheses are valid for (S).

3.1 The cooling package

The cooling package is placed at the front of the chassis, between the two side rails, as shown in Figure 1. The cooling package is characterized by its mass, stiffness and damping. It is placed on rubber bushings, illustrated by springs in the Figure 2. The cooling package is called (S).

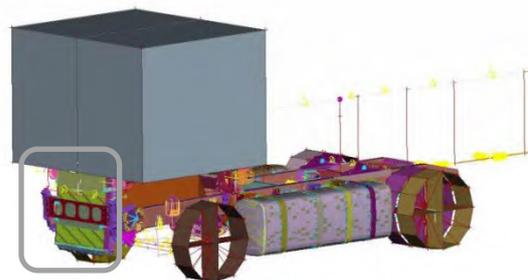


Figure 1 : The cooling package on the truck.

The structure is usually tested on a shake table, with input vibrations like accelerations measured on obstacles from the proving ground. A test procedure describes the number and the type of obstacles of the proving ground to run on the rig, in order to reproduce the complete life, regarding vibrations. The test procedure depends on the targeted customer's usage level. The types of obstacles can be potholes, Belgian pavés, bumps.

Early in projects, the loads can be computed by simulation on complete vehicles, driving on the obstacles

of the proving ground. These obstacles are described numerically in the simulation tool. In this paper, accelerations time signals from complete vehicle models are considered only.

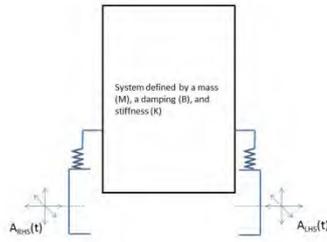


Figure 2 : Model of the cooling package

3.2 Input data

The aim of this paragraph is to present the loads directions and locations. The choice of the linear combinations a* is also described.

3.2.1 The loads

The loads measurements are described as follows.

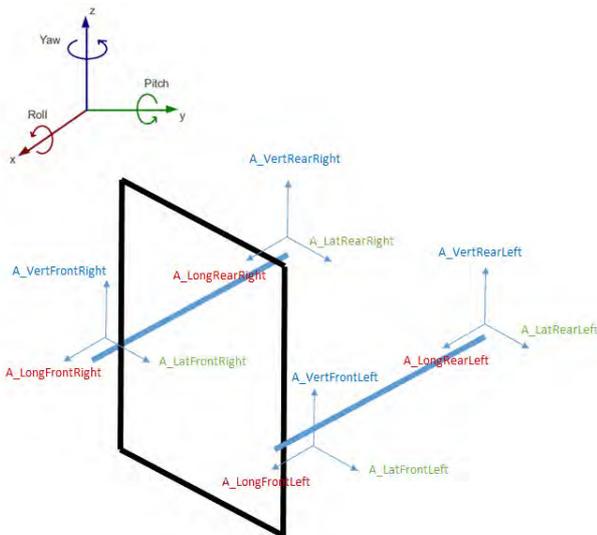


Figure 3: The loads

Accelerations at the attachment points (front) and at the rear of the cooling package are taken into account. Thus, (S) is submitted to 12 acceleration input, as illustrated in Figure 3. Let's consider that the longitudinal loads on the same side, are the same, but they can differ from left to right sides. Lateral accelerations at the left and right side are considered similar as well, but they can differ from front to rear. In this case, the front and rear lateral accelerations are very close: the yaw is mainly due to differences in longitudinal accelerations from left and right sides.

3.2.2 The trucks

Three different trucks will be considered in this study. They are called T27 (6x4 tractor, air suspensions at the front and rear axles), PAVT011 (6x4 tractor, leaf suspensions at the front and rear axles), FH1771 (6x4 tractor, leaf suspensions at the front, air suspensions at the rear). They are modelled in finite elements. The same cooling package is mounted on these three trucks.

3.3 6-DOF approach

For the sake of simplicity, a restricted number of multi-input loads and linear combinations a* are chosen. The choice is done thanks to the experience of the test engineers and design experts of the cooling package. The mode shapes of the cooling package can be summarized by six movements, 3 translations in X, Y and Z, and 3 rotations, roll, pitch, yaw, as illustrated in the Figure 3. The following linear combinations a* will be considered.

Nomenclature of load names

- A_VFR Vertical acceleration, front right
- A_VFL Vertical acceleration, front left
- A_VRR Vertical acceleration, rear right
- A_VRL Vertical acceleration, rear left
- A_LaFL Lateral acceleration, front left
- A_LoFR Longitudinal acceleration, front right
- A_LoFL Longitudinal acceleration, front left

	PITCH	LAT	LONG
A_VFR	1/2	0	0
A_VFL	1/2	0	0
A_VRR	-1/2	0	0
A_VRL	-1/2	0	0
A_LaFL	0	1	0
A_LoFR	0	0	$(1/2)^{1/2}$
A_LoFL	0	0	$(1/2)^{1/2}$

	YAW	VERT	ROLL
A_VFR	0	1/2	$(1/2)^{1/2}$
A_VFL	0	1/2	$-(1/2)^{1/2}$
A_VRR	0	1/2	0
A_VRL	0	1/2	0
A_LaFL	0	0	0
A_LoFR	$(1/2)^{1/2}$	0	0
A_LoFL	$-(1/2)^{1/2}$	0	0

In order to compare the trucks regarding their level of severity on (S), these six linear combinations will be analysed. The fatigue damage spectra, also called FDS, (Halfpenny, Investigation of the durability transfer concept for vehicle prognostic applications, 2010) (Halfpenny, Methods for accelerating dynamic durability tests, 2006) are used in this study. This allows comparing them in terms of damage as well as frequencies.

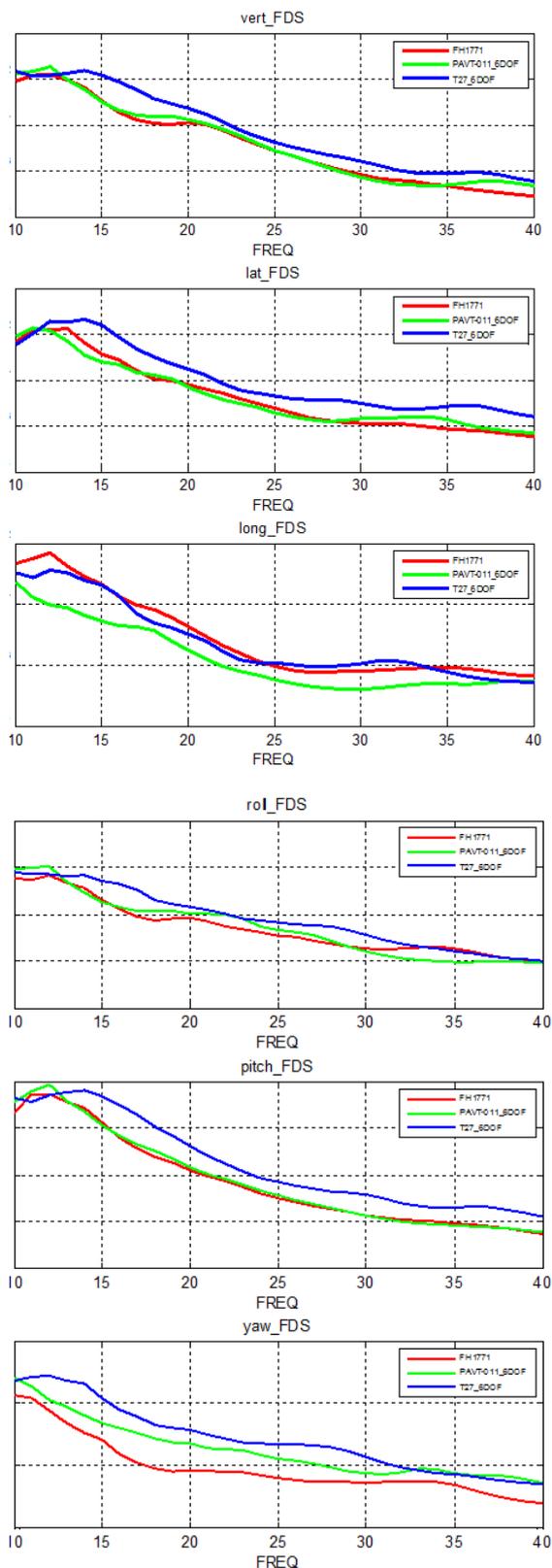


Figure 4: Fatigue damage spectra of the 6 linear combinations

Pitch, Vertical and longitudinal loads combinations are illustrated as examples in

Figure 4. For most frequencies, the truck T27 seems to be more severe, despite for the longitudinal direction.

If the damage at the critical point is more influenced by modes with longitudinal mode shape, the choice of the T27 is questioned. In general, this is not the case for the cooling package application.

The advantage of this approach is to refer to mode shapes of the structure, easily identified in a modal analysis of the virtual model. However, the method can be too restrictive as the mode shapes can also be combinations of these 6 movements.

3.4 Unit vectors uniformly spread on the unit hyper sphere,

In this part, the unit vectors are generated from the uniform distribution, according to the Equation (3). As a test, a set of 200 linear combinations have been generated and uniformly spread over the unit hyper sphere. The comparison of the trucks is based on these combinations. Results are expressed in terms of fatigue damage spectra. In Figure 5, only FH1771 and PAVT011 are illustrated, as an example. For the same frequency, a factor of almost 100 in damage can be observed between the 10th and 90th percentile.

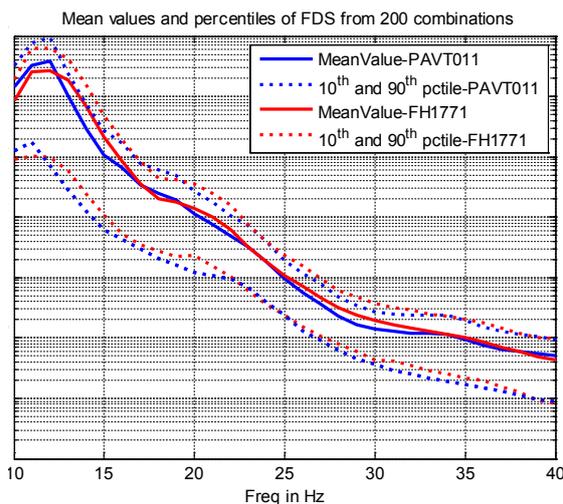


Figure 5 : Scatter of linear combinations of sets of loads from 2 trucks, regarding their fatigue damage spectra

In order to rank the trucks, the relative position of the FDS of each truck, for each combination and frequency, is of interest. For each frequency, we have counted the number of combinations each truck is the most severe.

In the Figure 6, the trucks PAVT011 and T27 are compared. The dark grey part of the histogram represents the number of linear combinations a* for which the truck T27 is the most severe one. The light grey part represents the number of linear combinations a* for which the truck PAVT011 is the most severe one. The figures are given in percentage of the total number of linear combinations. In overall, T27 covers 90.2% of the combinations and frequencies when compared to PAVT011. The truck T27 covers 87.9% of the

combinations and frequencies when compared with FH1771, as shown in Figure 7.

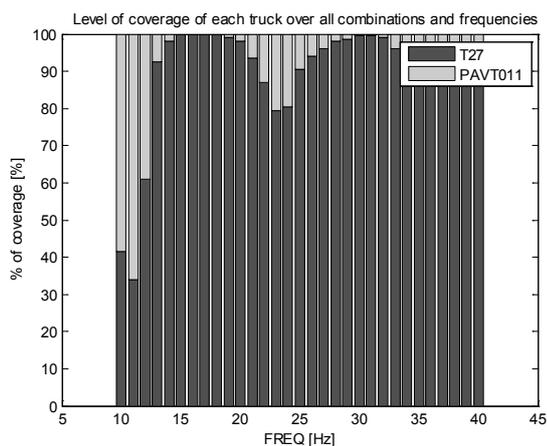


Figure 6 : Damage comparison between T27 and PAVT011

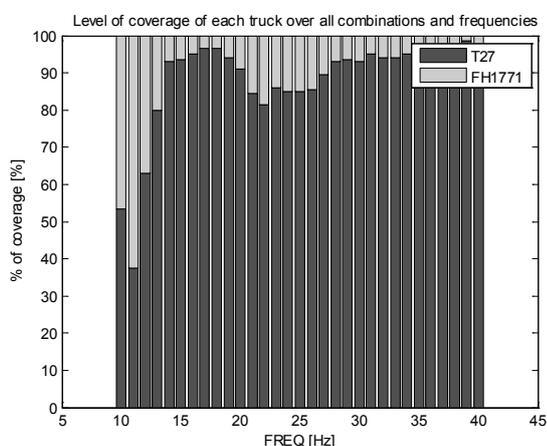


Figure 7 : Damage comparison between T27 and FH1771

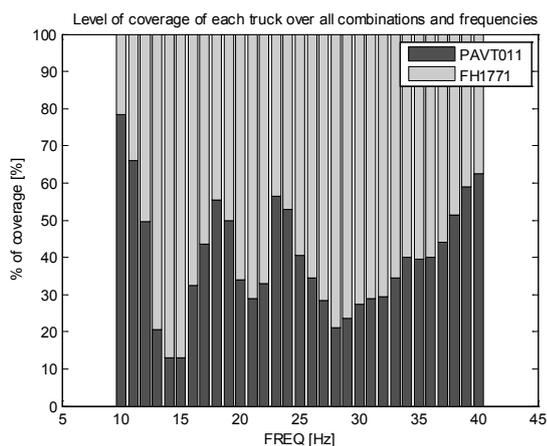


Figure 8 : Damage comparison between PAVT011 and FH1771

In the Figure 8, the same graph is presented for the vehicle FH1771 and PAVT011. The truck FH1771 covers 60.2% of the combinations and frequencies.

By experience, eigen-modes of most of the cooling package installations are higher than 13Hz. Thus, the

following conclusion can be drawn. The truck T27 seems to be the most severe truck, followed by both FH1771 and PAVT011.

A special comment can be added regarding the ranking on the FH1771 and PAVT011. In this case, it's more likely to find a cooling package (S) for which the truck PAVT011 would be more severe than FH1771. More information on the cooling package, e.g the impact of each acceleration component on the stress, would help to rank. Another possibility could be to adjust the severity of the loads of the chosen trucks, e.g FH1771, in order to cover more combinations. As these loads are to be used as input of a calculation or a rig, a possibility is to either extend the duration of FH1771 loads application, or increase the amplitude of FH1771 loads (in this last case a calculation is needed to check if the structure doesn't encounter plasticity). If these loads are used as input of durability calculations, another proposal could be to lower the damage criterion when computing damage calculation.

These proposals aim to mitigate the risk that the chosen truck is not the most severe one, among the trucks we have data from.

An extra comment regarding the comparison between FH1771 and PAVT011 can be drawn: it has been observed that, for 73% of linear combinations and frequencies, the damage ratio between the 2 trucks are between 0.5 and 2. It confirms that these 2 trucks are close regarding the damage. It can be hard to distinct them with this method only.

The same exercise has been done with 50 linear combinations, instead of 200. It shows that the truck FH1771 covers 59.8% of the combinations and frequencies, when compared with PAVT011. The truck T27 covers 73.5% of frequencies and combinations when compared with FH1771 and covers 74.5% when compared with PAVT011.

4 Conclusion

A method for storing trucks configurations regarding damage on a common structure has been detailed. Accelerations on same locations in the trucks have been considered. Several hypotheses have been done, in order to express local stresses from accelerations, with limited detailed information on the geometry of the structure and the material.

The main result is a severity indicator, representing the percentage of linear combinations of input loads and frequencies, where a truck is more severe than others. Fatigue damage spectra have been used to compare the loads. This severity indicator is global. In order to reduce the probability that the chosen truck might not be the best choice, a possibility is to increase the severity of the loads of the chosen truck, by increasing the duration of loads, or its amplitude.

The robustness of the severity indicator can be increased with the contribution of FEA analysis, or with the knowledge of the experienced designers and analysts. They are able to give an approximation of the eigenmodes, and, by that, orientate the severity indicator

to more precised ranges of frequencies. If the severity indicators of 2 trucks are too close, as it's the case for PAVT011 and FH1771, FEA analysis are eventually helpful to distinct them.

A special care is needed to check if the working hypotheses are valid in the context of the study. This can be done with the help of virtual simulation again. Moreover, the influence of number of linear combinations on the severity indicator can also be investigated.

A perspective is to check the validity of the choice of the most severe truck by measurements. Stress measurements at critical points of a chosen cooling package, on trucks or on rigs, can be helpful to confirm or contradict the choice. If the choice is refuted by measurements, an investigation of the appropriate a^* is necessary.

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