COUPLING PHENOMENA ON HEAVY VEHICLES: MEDIUM FREQUENCY EXPERIMENTAL ANALYSIS AND NUMERICAL APPLICATIONS

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ABSTRACT

Driveline vibrations of a truck are a cause of strong discomfort for drivers, and have to be investigated in early design stages. In order to develop analytical and numerical tools for the prediction of vibration of such a heterogeneous and composite structure, a deep knowledge of the physical phenomena involved is imperative. Few experimental studies have been performed on truck vibrations, and they mostly concerned single components of a vehicle. Therefore an Experimental Modal Analysis (EMA) of a complete truck has been performed in order to observe vibratory phenomena and determine influencing parameters involved in the vibration transmission. Low Frequency and High Frequency ranges were located, and the so called Medium Frequency range was determined. In the latter, interesting transition and interaction phenomena take place, which are thought to have a first order influence on vibration transmission over trucks.

The results of the test campaign are used to validate a pre-design Finite Element model of the complete vehicle, which is subsequently used to investigate the phenomena observed in the experimental phase.

1 INTRODUCTION

Experimental Modal Analysis ([1], [2]), one of the best known techniques to characterize the dynamic behaviour of a structure, is commonly applied in the heavy vehicle industry ([3], [4], [5]). Though, studies usually focus on the dynamics of single parts of a vehicle and no EMA of a whole truck exists, to the authors' knowledge, in the literature; that is why a test campaign was launched, with a twofold objective: give a preliminary idea of the dynamic phenomena found in a heavy vehicle and provide an experimental reference for the validation of a numerical model.

The numerical model will be used for assessments on vibratory performance estimators. The need for a numerical model to perform this kind of assessments is of primary importance for truck analysis, due to the large variability in truck configurations, which makes the study of every specimen unachievable.

Along with EMA, another test campaign has been launched, to perform the so called Operational Modal Analysis, or output-only Modal Analysis. Results were not promising, both from the point of view of their quality and from the point of view of the comparison with modal parameters identified through hammer tests. This bias can originate from incorrect hypotheses formulated on the current excitation, or from modifications that the structure encompasses when excited in its real operating conditions. The authors decided to rely only on hammer test results in view of future exploitation for numerical models validation.

2 TEST SETUP AND CONFIGURATIONS

A heavy vehicle is roughly made up of a chassis, a cabin, a powertrain, axles, and elements suspended to the chassis (Fig. 1). It appears as a composite structure, from the point of view of the architecture and of the dynamic couplings among its components, in other words, it is a composite structure on a macroscopic scale (opposed to the microscopic scale where the composite nature of some materials shows up).

The chassis constitutes the main transfer path for vibration originating from the powertrain. This is the reason why the current analysis focuses on the chassis; all the elements suspended to the chassis are also considered important, because of the way they are supposed to modify the dynamic stiffness of the former. The test campaign has been performed on a Medium duty truck, having a Gross Vehicle Weight (GVW) of 12 tonnes.

Two configurations have been studied:

- Configuration A: unloaded truck;
- Configuration B: truck loaded with superstructure.

The superstructure is used to simulate the truck payload; it is a welded frame that approximately reproduces the torsional stiffness of common truck payloads. The superstructure has been tested itself in free-free boundary conditions, to determine its modal properties.

Impact testing is performed on these two configurations, and respective modal parameters are then compared; comparison between configurations A and B allows drawing conclusions on the effect of chassis stiffness modifications and mass addition.

The truck has been tested while lying on its tyres; indeed, it is assumed that free-free boundary conditions are not attainable (based on the rule of thumb stating that rigid body frequencies should be below 10% of the first flexible mode, [1], [2]), thus the boundary conditions are rather chosen to fit the numerical model conditions. As a matter of fact, testing a massive structure on its suspensions is a common practice in Ground Vibration Testing of aircrafts.

The structure is excited through an impact hammer, heavy enough to inject a sufficient amount of energy into the structure. Measurements are performed on the frequency range [0 Hz

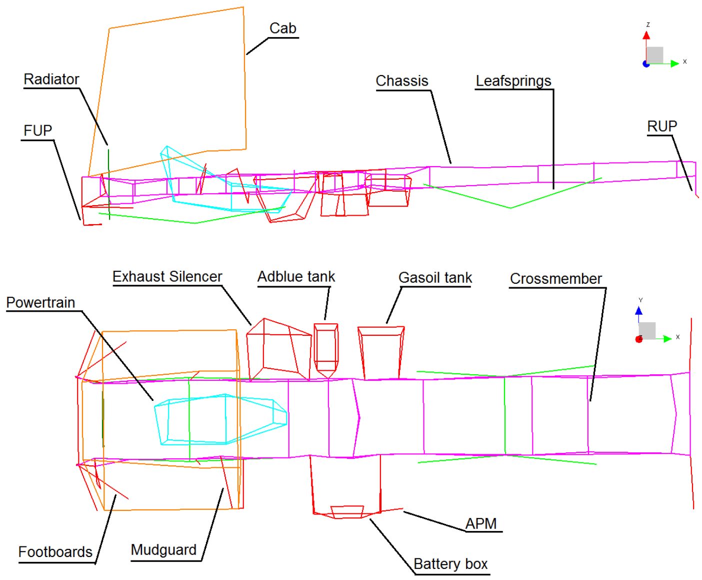


Figure 1: Truck geometry

- 256 Hz]. Pre-test check of input excitation allows limiting the validity of Frequency Response Function's (FRF) to the frequency range [0 Hz - 160 Hz]: a drop of 10 dB in the power spectrum of the injected force highlights the value of this limiting frequency, based on a common rule of thumb ([1], [2]).

A compromise between testing time and testing accuracy leads to choose a frequency step of 0.25 Hz for all acquisitions, thus causing a limitation with respect to the estimation of damping at low frequencies; the authors evaluate that a damping ratio of 1% (a realistic value for such a structure) could be estimated with sufficient accuracy only above 25 Hz. Thus the damping estimation is affected by a certain degree of inaccuracy.

The structure is impacted at two reference points so as to excite the highest possible number of modes. Additional measurements are also performed with input forces on several suspended elements: these sets of measurements serve to inject energy specifically to these components, thus estimating their dynamic characteristics in a boundary condition correspondent to the attachment to the chassis; this boundary condition is intermediate between a fixed interface, and a free interface.

FRF's were acquired by roving tri-axial accelerometers over 143 points (most of whose lied on the chassis). Mass loading from accelerometers was considered to be negligible, because of the lightness of accelerometers (ICP sensors weighting 10 g are used) with respect to the test object.

Modal parameters are identified through the commercial software LMS® Tes.Lab, and a poly-reference Least-Squares Complex Frequency-domain (LSCF) estimation method (PolyMax) is exploited, [6]. The sum of all the measured FRF's and Complex Mode Indicator Functions (CMIF) are fed to the PolyMax algorithm, and physical resonances are detected thanks to a stabilization diagram.

3 DYNAMIC CHARACTERIZATION OF THE UNLOADED TRUCK

The analysis of the sum of all FRF's brings to light a fundamental information: in the frequency range considered, the structure presents well separated behaviours that are typical of the Low Frequency (LF) and High Frequency (HF) ranges. Besides, a transition range commonly called

Medium Frequency (MF) range is observable (Fig. 2). The LF behaviour is characterized by the presence of rigid body modes of the vehicle and of its main suspended constituents (cabin, powertrain); modal density is quite high. The MF behaviour is dominated by elastic deformation modes of the chassis, coupled with all the other components; modal density is lower than in the LF range, while damping is increasingly high. The HF behaviour is characterized by a sort of structural diffuse field; here only local deformation shapes are visible, and modal density is extremely low.

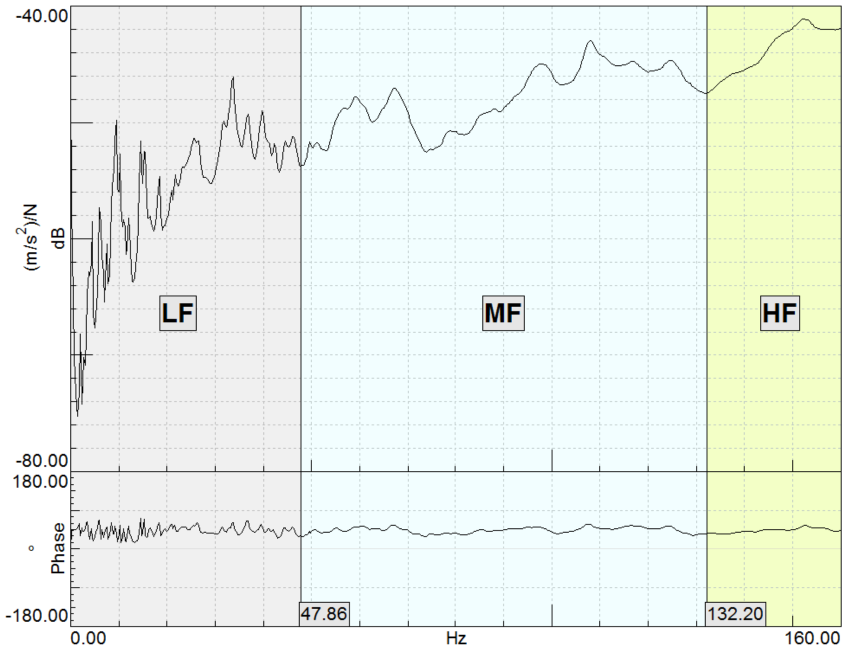


Figure 2: Sum of all the measured FRF's and characteristic frequency ranges.

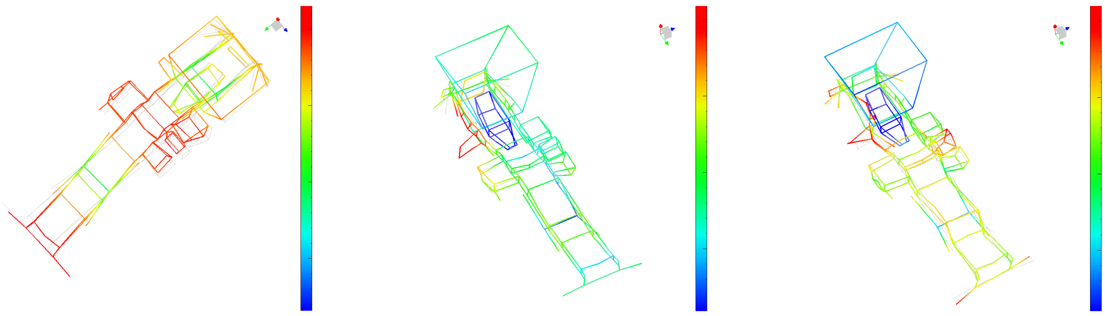
Two main conclusions can be drawn, these being:

- to study whole-body vibrations of a complete truck in an early design stage, all the information can be found in the frequency range [0 Hz - 160 Hz]; phenomena at higher frequency are barely an extension of the HF behaviour, or local phenomena that attain to the analysis of limited parts of the truck;
- a transition frequency can be defined, lying at about 50 Hz, where the behaviour of the structure changes from global to local.

The transition from global to local behaviour, or equivalently from long wavelength to short wavelength wave propagation, happens in the MF domain; here, the interaction between stiffer and more flexible components (the suspended elements and the chassis) is thought to drive the said transition. A number of studies on the said interaction is found in the literature, but they are rather analytic or applied to academic structures. Thus further analyses are needed to highlight the causes and physics of the behaviour for the case at hand; they can be better carried out with numerical tools, as the models that are the final objective of this study.

3.1 Modal parameters estimation

A table of modes (natural frequencies, modal damping and corresponding deformation shapes) is constructed; both global and local modes are identified, depending on the frequency range considered (Figs. 3a, 3b and 3c).



(a) Mode 10: 9.30 Hz, 1^{st} horizontal bending.

(b) Mode 33: 39.64 Hz, local bending of front chassis LHS.

(c) Mode 56: 110.67 Hz, local modes.

Figure 3: Some estimated mode shapes.

A check on the quality of modal parameter estimation shows that modal data are better identified in the low and mid frequency ranges (up to approximately 50 Hz), where identified modes have mostly real deformation shapes.

3.2 Influence of payload on the dynamic behaviour

The dynamic behaviour of the superstructure can be determined with a high degree of accuracy, due to its low modal density. Nevertheless, it is interesting to analyze how the superstructure influences the modes of the complete structure, when lying on it. The superstructure lies on the chassis, and their connection is made through a wooden interface and clamping bars: the transmission of forces turns out to be distributed along a line, instead of concentrated on discrete points.

As one could expect, the main consequence of adding the superstructure to the complete vehicle is a frequency shift (the superstructure represents 30% of the mass of the unloaded vehicle), Fig. 4. Nonetheless, this frequency shift changes at each mode (natural frequencies increase in certain cases and decrease in others, when going from configuration A to configuration B) due to the fact that the added mass has its own dynamics. It is interesting to see the effects on deformed mode shapes: this is done synthetically by calculating the Modal Assurance Criterion (MAC) matrix for the modal shapes of the two configurations.

4 NUMERICAL MODELS

A Finite Element (FE) model has been assembled for a complete truck; it comprises all the main components of the tested truck, with different levels of detail, Fig. 5.

This model must be validated for dynamic calculations. Due to the differences between numerical and experimental meshes, and having experimented some bias when calculating MAC matrices [7], the authors decided to update the FE model based on frequency response, instead of mode shapes; the target of the model update is to match both natural frequencies and response amplitudes.

The comparison of measured and calculated frequency responses is promising, since orders of magnitude and trends of frequency response functions are well predicted. Besides, the calculation confirms that impact test results, instead of operational test data, should be used for comparison with numerical data. Current work aims at updating the numerical model.

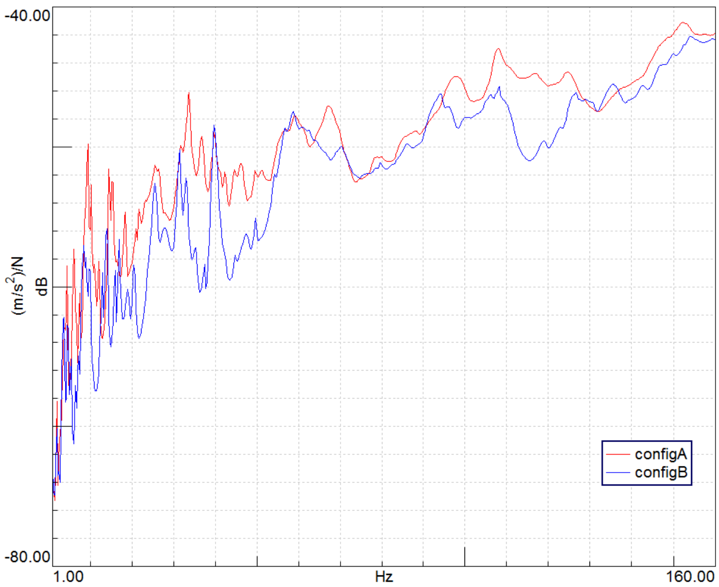


Figure 4: Comparison of the sum of FRF's measured for configurations A and B.

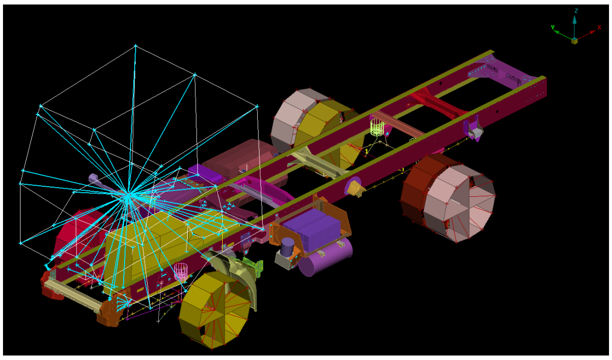


Figure 5: Complete Vehicle Model.

4.1 Applications of the FE model

Once validated, the presented FE model is used to make assessments on the physical phenomena found in the experimental phase.

One interesting application consists in the evaluation of the interface conditions between chassis and suspended elements; possible estimators for the interface conditions are:

- displacements at points of the interface;
- forces transmitted between the chassis and suspended elements at the interface;
- normal modes of the suspended elements in different boundary conditions (free-free, fixed, screwed on a siderail).

Interesting conclusions can also be done by comparing the computed natural frequencies for suspended elements and assemblies. Fig. 7 shows modes of the chassis and of the main elements suspended to the latter; modes of the chassis and suspended elements assembly and of the complete vehicle are also shown. What is shown is barely the mode count, thet helps to identify the frequency ranges spanned by each component.

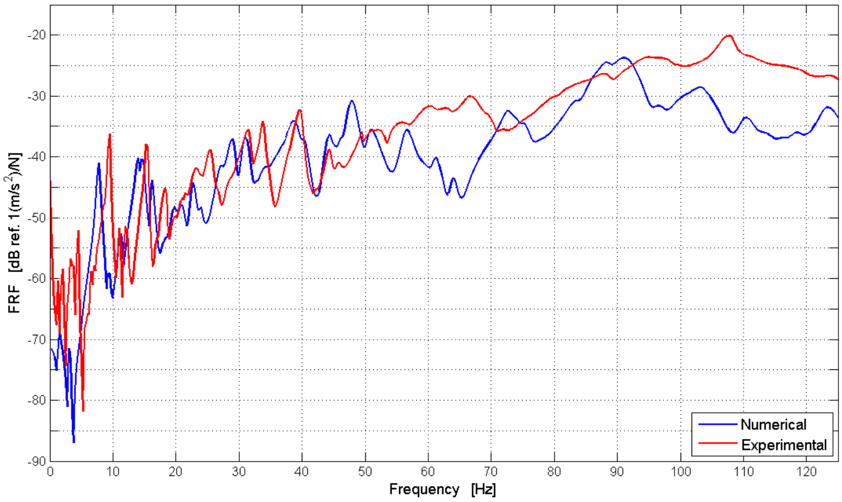


Figure 6: Point FRF's on a selected point of chassis: comparison between experimental and measured curves.

One can see that the first modes of the chassis appear at quite low frequency, while the first modes of the suspended elements appear at high frequency. One thus observes the interaction between a flexible component (the chassis) and stiffer ones (suspended elements), that has been mentioned in Section 3, the notion of stiff and flexible being here intended to be iso-mass. By deeply inspecting mode shapes related to the shown natural frequencies, one is able to understand how coupled modes originate.

4.2 Future developments for numerical models

The objective of the project which is the framework of this study is the development of numerical tools for the prediction of vibro-acoustic estimators in truck cabins. The named tools should be used at an early design stage, so one should be able to perform rapid calculations thank to these tools. Conventional Finite Element is seemingly not the most adapted technique to fulfill the aimed task, so the authors identified better methodologies to develop the said tools: the Wave Finite Element method ([8]), in conjunction with suitable reduction techniques, will be employed to build reduced models for the calculation of the frequency response of a simplified structure representing the vehicle. The FE model presented here will represent a reference for the reduced models.

5 CONCLUDING REMARKS

An experimental modal campaign on an industrial vehicle is described and the estimation of modal parameters carried on. Physical phenomena linked to the onset of the so called Mid-frequency range are highlighted thanks to the measurements. The results of the experimental campaign can serve as a reference for comparisons with numerical models developed in the truck industry, most of all when concerning frequency ranges and expected physical phenomena. Measurements allow identifying local vibration phenomena, and investigating the influence of mass and stiffness changes.

The estimated modal parameters are used as a basis for modal update of a FE model. First correlations are promising, but an effort has to be made to further improve the predictability of the numerical model. One possible application of the numerical model is showed, and its usefulness in interpreting the experimental phenomena is illustrated. Future reduced numerical

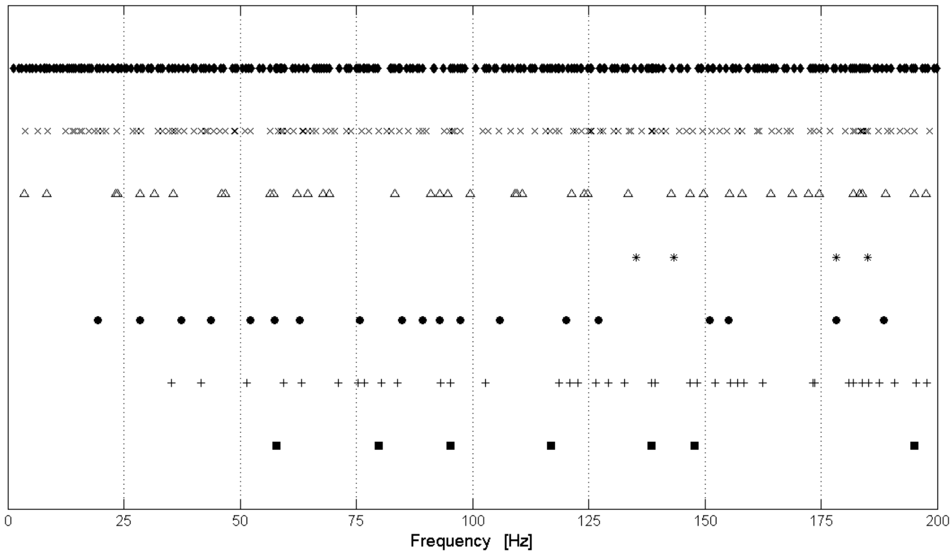


Figure 7: Mode count for components and assemblies. $\square + \circ *$: suspended elements in free-free BC, $\triangle$: frame in free-free BC $\times$: frame assembly in free-free BC, $\diamond$: complete vehicle lying on its tires

models are briefly introduced and their relationship with the FE model explained.

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