

TEST-BASED DYNAMIC CHARACTERIZING OF A COMPLETE TRUCK BY OPERATIONAL MODAL ANALYSIS

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Abstract

At the end of its development cycle, a new truck variant is exposed to vibration tests, both in laboratory conditions by means of a 4-shaker test rig and in real road tests on a proving ground. The idea is to extend the use of these test data by applying Operational Modal Analysis and by being able to correlate the analysis results with Finite Element predictions. Basically, the only feasible way to perform a modal analysis experiment on a complete truck is by using test rig or real road excitation. In both cases the forces introduced in the structure are not available or difficult to measure and, hence, Operational Modal Analysis needs to be applied.

To verify the effectiveness of Operational Modal Analysis using test rig data, it was decided to perform time-domain simulations using the complete vehicle model which was subjected to the same excitation as a real truck on the test rig. This paper will compare the directly computed Finite Element modes, the modes found by applying Operational Modal Analysis to the simulated time histories and the modes identified from the real test rig measurements.

1 Introduction

Operational Modal Analysis (OMA) is used to derive an experimental dynamics model from vibration measurements on a structure in operational conditions (as opposed to dedicated laboratory testing). Cases exist where it is rather difficult to apply an artificial force and where one has to rely upon available ambient excitation sources. It is practically impossible to measure this ambient excitation and the outputs are the only information that can be passed to the system identification algorithms. In this case one speaks of Operational Modal Analysis. During the last 15 years or so, Operational Modal Analysis developed and reached a mature state with advanced parameter estimation algorithms, commercial software implementations, and very relevant industrial applications.

This paper discusses such an industrial case. The need exists to experimentally validate a Finite Element Model (FEM) of a truck. This model is called Complete Vehicle Model (CVM) and is represented in LMS Virtual.Lab [1] in Figure 1. It is rather cumbersome to apply a classical modal test to a complete truck using for instance a free-free suspension system and installing at well-chosen locations along the structure some modal shakers. Therefore, the idea arose to use operational vibrations as originating from proving ground tests or as replicated in laboratory conditions where a truck is put with its 4 front wheel on a 4-shaker test rig. In these conditions, the forces are not measured and the truck acceleration responses are the only available information.

The aim of this work is to investigate the applicability of OMA to this data. The starting point of the analysis is the correlation between the mode shapes directly calculated using the CVM and
Operational Deflection Shapes (ODS) obtained from measured test-rig vibration responses (Figure 2). It is observed that this correlation is not excellent. This can have the following reasons:

- The ODS are obtained from the measurement data simply by taking the amplitude and phase at assumed peaks in spectrum plots. Especially in cases with high damping ratios and thus high modal overlap, the ODS will be a combination of mode shapes. For this reason, a true OMA approach where the modes can be isolated by applying curve-fitting techniques to the measured spectra is expected to yield better results.

- In the frequency range of interest, the CVM yielded 52 modes, which are all represented in Figure 2. However, many modes are very heavily-damped (most of the modes have damping ratios between 10 and 60%) and not all of them will be visible in the response. Also, the excitation is applied at the wheel basis. So modes having a low participation factor at these locations will not be well excited.

- Finally, due to modelling inaccuracies, discrepancies may exist between the CVM and the real truck. It is precisely the reason for the use of OMA to validate and possibly update the CVM based on measured vibrations.

This paper will investigate the first two arguments listed above in order to improve the initially disappointing correlation result of Figure 2.
2 Operational Modal Analysis – a very brief review of methods

A whole range of OMA curve-fitting techniques is available [1]. However, an evolution is observed from non-parametric methods to parametric methods [3]. Non-parametric means that signal processing techniques are used (FFT, SVD, …) to enhance certain signal features (frequency domain peaks). In the peak-picking and SVD-based methods, the user has to select modes based on some peaks in spectra. Associated drawbacks are that it is a subjective process, involving the need to inspect many spectra to find all the modes and sometimes the peaks not very well visible (e.g. cases with high damping / large noise levels).

Parametric means that a model is fitted to the data. Among other, parametric methods have the great advantage that stabilization diagrams can be used to objectively find the modes. Indeed fitting models to data means that models of different order can be fitted and, by consequence, a stabilization diagram is created. An example of a parametric method that received considerable attention recently is the PolyMAX method [4][5][6]. Its main benefit is that it yields extremely clear stabilization diagrams and thus that it implies the potential to be used as an autonomous modal parameter selection method. Another successful parametric method is Stochastic Subspace Identification that estimates a state-space model from estimated output cross-correlation functions [7][8]. Both PolyMAX and Stochastic Subspace Identification are implemented in LMS Test.Lab – Operational Modal Analysis [9].

3 Preliminary simulations

In order to examine the possibilities and limitations of OMA in case of a heavily damped truck, it turned out to be instrumental to consider first a simplified simulation example with only a few DOFs (Figure 3). The left part of Figure 3 indicates what happens when testing the truck on a 4-shaker test rig. The test rig is displacement-controlled so that a prescribed road profile can be realized in laboratory conditions. The model also contains the connection between the rig platforms and the ground. The right part of Figure 3 indicates how the simulations can be performed: imposing a displacement at a certain DOF means that the structure can be “cut” at that DOF. This means here that the platform is fixed vertically and that the imposed displacement is modelled as a force applied to the wheel hub, where the force equals the platform displacement times the tire spring constant (note: it is also possible to add a tire damper, simply by adding a damper force proportional to the platform velocity). These considerations also clarify the CVM simulation approach: a setting similar to the right part of Figure 3 is used both to generate the simulated time responses using a transient analysis and to directly compute the eigenmodes of the CVM.

The simulation model of Figure 3 contains 5 modes. The dampers were selected to have heavily damped modes (damping ratios of about 7, 10, 20, 30 and 90%), representative for the truck modes. Following observations can be made with respect to the application of OMA to this case:

- When using the *analytically computed* spectra, all modes are perfectly identified using Operational PolyMAX, including the one with damping ratio around 90%. However, when estimating the mode shapes using the LSFD (Least Squares Frequency Domain) method [10], the mode shape of the 90% damped mode could not be estimated.
- When using a *realistic number* of simulated time samples to estimate the spectra, very noisy spectra were obtained (even though no measurement noise was added to the data). In this case only 2 modes could be well identified (at 7 and 10 % damping).
- When using a *very high number* of simulated samples, the spectrum estimate approaches the analytically computed and the mode estimation also improved, though the modes at 30 and 90% damping remained very difficult.
From these preliminary simulations, it can be concluded that OMA has difficulties in estimating extremely highly damped modes. The reason is that it is very difficult to distinguish the response to such modes from the effect of the unmeasured randomness of excitation.

Figure 3: Simplified simulation model that is representative for the CVM simulation. (Left) system excited by imposed displacement; (Right) equivalent modelling by a force and change of boundary condition.

4 4-shaker test rig simulation and measurement

In a next stage, the detailed CVM is used to simulate a 4-shaker test rig measurement. Uncorrelated white noise displacements are applied to the wheel platforms. The CVM responses were computed at 78 DOFs, corresponding to the measured DOFs from the real test. During the measurements, the 78 DOFs were not measured at once, but in 3 different runs, keeping 6 sensors in common to each run. These reference DOFs have been carefully selected and are distributed over various components of the truck (cabin, frame, powertrain, rear axle, trailer). They allow combining the analysis results of the 3 runs. Figure 4 compares the simulated autospectra with the measured ones. They agree reasonably well. Note that the autospectra have phases different from zero in Figure 4. This is a consequence of the particular spectrum estimate used: first the auto- and cross-correlation functions are computed from the time data, then an exponential window may be applied to the correlation functions and afterwards a single (i.e. no averaging used) Fourier transform of the positive time lags of the (windowed) correlation functions is computed to yield the so-called half spectra. More information on this approach and the specific advantages in an OMA context may be found in [11].

Both Stochastic Subspace Identification and PolyMAX have been applied to both the simulated and measured data. Although both methods are known to yield accurate modal parameter estimates, it is obvious that PolyMAX considerably facilitates the identification process because, typically, much cleaner stabilization diagrams are obtained [12].

Figure 6 shows the correlation (MAC) between direct FEM mode shapes and PolyMAX mode shapes identified from test rig CVM simulations. Although it is clear that many FEM modes cannot be extracted from the simulated data, some modes correlate very well. The right part of Figure 6
compares the PolyMAX and Stochastic Subspace Identification mode shapes. Such a comparison between 2 OMA approaches contributes to an increased confidence in the experimental results.

Figure 7 graphically shows the correlation between 2 mode shapes. In Figure 8, some more direct FEM mode shapes are shown that have a well-correlated experimental counterpart.

Figure 4: AutoSpectra at reference sensors. (Left) CVM simulation (Right) Truck measurement.

Figure 5: Stabilisation diagram obtained by applying PolyMAX to the CVM transient analysis data.
Figure 6: Correlation (MAC) between: (Left) direct FEM modes and test rig CVM simulations and (Right) PolyMAX and Stochastic Subspace Identification mode shapes.

Figure 7: Example of a well-correlated direct FEM mode and a mode extracted from time-domain data using OMA.

Figure 8: Some typical FEM modes that can also be retrieved from time-domain data.
In order to examine the reason why many direct FEM modes are not identifiable using simulated CVM data, the (inverse of the) damping ratios and relative mode participations are shown in Figure 9. The relative importance of different modes in a certain frequency band can be investigated using the concept of mode participation. For each mode, the residues are computed, i.e. mode shapes times modal participation factors (i.e. scaled mode shape components at the input locations which are in this case the wheel hubs). The sum over the outputs of all residue values for a specific input indicates whether that specific mode is well excited by that specific input. The summation over all inputs for each mode represents an evaluation of the importance of each mode. In Figure 9 (Bottom), the highest value has been scaled to 100 (mode 37), so that the relative mode participations are shown. Modes having a high value for both criteria represented in Figure 9 have a high probability to be present in the simulated as well as in the real measurement data. It is clear that not many modes share a low damping ratios and a high participation factors at the input locations and therefore it is not unexpected that not many of the 52 direct FEM modes are observable in the truck response data.

Figure 9: 52 FEM modes: (Top) Inverse of the damping ratios; (Bottom) relative mode participation.

5 Conclusions

This paper discussed the application of Operational Modal Analysis (OMA) to experimentally determine the dynamic properties of a complete truck. These properties would serve the validation process of a so-called Complete Vehicle Model (CVM), which is a Finite Element Model (FEM) of an entire truck. Rather than trying to set-up a classical modal analysis test in the laboratory, it was the idea to use 4-shaker test rig measurements for this purpose. This paper showed that, starting from a relatively low agreement between Operational Deflection Shapes (ODS) extracted from measured test rig data and directly computed CVM modes, it was possible to improve the agreement by using OMA. Furthermore by means of preliminary simulation examples (Section 3) and an in-depth investigation of the controllability and observability of the predicted FEM modes
(Section 4), it was possible to highlight the reason why many of the FEM modes cannot be identified: they are either not well excited by the road input or they are very heavily damped. One could argue that these modes can be discarded as they are not relevant to describe the truck vibration behaviour in normal operational conditions. However, if the goal is to validate and improve a model based on test data, trying to maximize the amount of information that can be found in the experiments is always a good option. Final conclusion is that OMA is a very valuable technology that has its place in the vehicle development process. It enhances the exploitation of operational data that becomes available when performing 4-shaker rig or test track measurements.

Acknowledgements

Part of the work was conducted in the framework of the EC 6-FWP research project NMP2-CT-2003-501084 “INMAR” (Intelligent Materials for Active Noise Reduction, www.lbf.fhg.de/inmar). The support of the EC is gratefully acknowledged.

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