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Comparison of experimental and operational modal analysis on a laboratory test plate

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ABSTRACT

Operational modal analysis (OMA) is widely used whenever the dynamic characteristics of structures that do not fit into a laboratory are desired. In addition, OMA offers a test of the structure under its real boundary conditions which may sometimes be preferable for validation of numerical models. Theoretically, the natural frequencies and damping ratios should be identically estimated by an OMA test and an experimental modal analysis (EMA) test. However it is still often reported that EMA tests are more reliable. The present paper presents a thorough comparison of EMA and OMA tests of a Plexiglas plate. The experiments were carefully designed, to ensure that the plate was tested under similar boundary conditions. Estimated modal parameters from the EMA test and OMA test are presented and compared, for the first ten modes of the plate. It is found that natural frequencies are deviating by less than 0.3%, damping ratios by less than 7%, whereas cross-MAC values between the mode shapes of the two tests are found to be above 0.99. The experimental test was conducted first by an EMA test, followed by the OMA test and finally another EMA test was conducted in order to catch any time-variance. It is concluded that no significant differences were found between modal parameters obtained by OMA and EMA.

1. Introduction

As simulation of vibration responses based on numerical models are increasingly used to investigate dynamic behavior of structures, experiments are increasingly required for validation or correction of the numerical models. As an example, a dynamic model of a structure with incorrect damping assumptions could give misleading life time estimations [1]. Because of the complexity of damping effects, it is usually not possible to give a proper analytical estimate, hence experimental tests are needed. However, damping can be a notoriously difficult parameter to consistently estimate from experiments, and the accuracy of estimates of damping from, particularly, operational modal analysis (OMA) is currently receiving a lot of research interest.

Experimental modal analysis (EMA) has been used for decades to extract information about the structural damping. EMA is considered reliable because it is based on input-output system identification, which allows validation e.g. of the estimated frequency response functions (FRFs) by coherence functions. However, the strength of EMA is also limiting its applicability as it requires that all inputs (excitation forces) are measured, which is practically unfeasible for many structures.

The last couple of decades, OMA, by which only the structural responses are used, has been widely applied and described in literature. OMA is attractive in many situations because it can be applied to structures in operation, and does not require excitation, which is practical for many large structures, see for example [25,19,24,4,27]. In addition, OMA can sometimes be preferable for validation of numerical models because the effects of boundary conditions are included in the results, as opposed to EMA where the structure is usually tested under free-free conditions. OMA methods are based on some assumptions about the nature of the loads exciting the structure; the loads being the result of random processes, that may be colored as long as the poles of the force coloring are different from those of the structure. In addition the structure should be linear and time-invariant [2–4].

Theoretically, modal parameters should be identically estimated via an OMA test and a classical EMA test [5]. However, it has still sometimes been reported that an EMA test is more reliable because of the available information and controlled environment [6]. Furthermore it has also been reported that the damping estimates from OMA are over estimated, hence suggesting a critical bias error [7,8].

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Little comparison between EMA and OMA has been published where exactly the same structure and experimental setup have been used for both cases. In [9] however, a comparison of EMA and OMA was presented for some case studies of structures with simple to more complex geometries and it was shown that for simple geometries the differences of the extracted modes were small. However, difficulties to extract all modes from OMA tests were reported and in addition it was reported that choosing the locations of the excitations in an OMA laboratory test is non-trivial. For a simple geometry with random spatial excitation, problems with finding all modes from the OMA test were reported, and it was found that the mode shapes from the two tests were not identical. The problem of undiscovered modes increased when the excitation was localized to a single point for the OMA test.

For a more complex structure also investigated in [9], a considerable number of modes were not well estimated or not estimated at all. Damping ratios were not reported and mode shapes were in general showing poor similarity between the EMA and OMA tests. However, it should be noted that only a single reference DOF was used for the calculation of spectra and the measurement time was not reported. Both the number of reference channels and the measurement time are crucial for OMA [6,10].

In [11] a comparison of EMA (using impact testing) and OMA was carried out, which showed large deviations in the estimated damping ratios. However, the methods used for parameter estimation for OMA and EMA were not the same and thus the applied settings for estimation are not comparable. The authors of [11] pointed out that the difference in vibration levels between the two tests could be the reason for the large deviations of the damping ratios.

In [8] a comparison of EMA and OMA was performed on the same data set, acquired from excitation with two shakers and roving accelerometers over a test structure. For OMA the measured input was simply ignored, using only the responses. It was observed that the damping ratios estimated by OMA were higher than the ones from EMA and the same was observed for the natural frequencies. However, in this study the dynamical characteristics of the shakers seems too have been included in the OMA, but not in the EMA, results. For EMA the influence of the shakers is excluded by using frequency responses relative to the inputs, but this is not the case for OMA. Thus the boundary conditions were very different in the OMA test compared to the EMA test, which explains the higher damping in the OMA estimates.

It is worth noticing that one of the attractive features of OMA is that it indeed includes the boundary conditions. This means, however, that in order to compare EMA and OMA, great care must be taken in the design of the experiment to ensure that the boundary conditions are identical in both tests.

The present paper presents an experimental study of EMA and OMA tests on a Plexiglas plate where the EMA and OMA tests have been carefully designed so that they test the structure under very similar boundary conditions. The plate, which is a simple geometry, is designed to have closely spaced modes and is considered a good structure for comparison. Modal parameters are estimated and compared between the EMA and OMA tests.

Damping ratio estimates are the primary focus in the present work as they are known to be the most challenging, although both natural frequencies and modes shapes are also compared for completeness.

2. Experimental setup

The Plexiglass (PMMA) plate that has been experimentally tested can be seen in Fig. 1. The dimensions of the plate are 533 mm × 321 mm × 20 mm. The plate is similar to the so-called IES-Plate proposed in [12], although, for practical reasons, the plate thickness was chosen slightly different from the IES-Plate, by choosing 20 mm thickness, which is a standard thickness in Europe.

The measurement grid consisted of 35 out-of-plane DOFs distributed uniformly over the plate as shown in Fig. 2 and responses from all DOFs were measured simultaneously. The accelerometers and cables significantly mass load the structure, and it is also likely that there is additional damping added. This is, however, irrelevant, as the present experiments are designed so that it is the plate together with the instrumentation that is investigated. The conditions for the EMA and OMA tests should therefore be equal, albeit not being those of the free-free plate.

The following data acquisition (DAQ) and sensor equipment were used:

- 3 National Instrument 4497, 16 channel, 24bit analog inputs cards.
- 35 Dytran 3097A2 accelerometers, 100 mV/g, IEPE, 4.3 grams.
- 1 Dytran 5800B4 impulse hammer, 10 mV/N, IEPE, with plastic tip.
- In-house software for DAQ control, based on MATLAB DAQ-toolbox.

The plate was suspended by springs via a thin fishing line, see Fig. 1, giving it relatively low rigid body modes (<3 Hz). For all tests a sampling frequency of 5 kHz was used.

For the EMA test, frequency response functions (FRFs) were measured by impact testing. FRFs relative to two different excitation DOFs were measured, in order to have multi-reference data for the parameter extraction. DOFs 1 and 29 were thus excited, and time sequences of 20 impacts with 20 s in-between each impact were acquired for each reference point.

For the OMA test, 300 s of data were acquired from all DOFs simultaneously, corresponding to approx. 40000 periods of the lowest natural frequency. This rather long measurement time was chosen to reduce possible effects on modal parameters by limited measurement time, see e.g. [10,4]. The excitation was applied by gently tapping the tip of a pencil randomly around the plate. The idea of tapping rather than scratching was chosen in order to minimize the interaction with the plate as much as possible, as it was initially experienced that scratching the plate influenced its dynamics in a significant way, both on natural frequencies and damping ratios.

Two impact tests were performed; one test was performed before and one after the OMA test, in order to be able to detect any time-variance. The impact test before the OMA test is referred to as *EMA1* and the one after is referred to as *EMA2* in the following. All three tests were conducted consecutively within approximately one hour, under which no interaction with the experimental setup (except the excitation) was allowed. Between the end of the *EMA1* test and the start of the OMA test, 15 min elapsed, and from the end of the OMA test to the start of *EMA2*, 5 min elapsed, all due to initial quality checks. Each of the impact tests took approximately 15 min from start to end.

3. Data processing and analysis

3.1. Impact tests

The acquired time sequences from the impact test were processed to obtain FRFs. FRFs and coherence functions were estimated by averaging the spectra from two selected impacts of the time sequences (the input and all the responses) using the H_1 estimator in the MATLAB toolbox ABRAVIBE [13]. The two impacts