Efficient modeling in vibroacoustic systems with consideration of uncertainties and comparison with experimental measurements

P. Langer¹, C. Guist² and S. Marburg³

¹,³Chair of Vibroacoustics of Vehicles and Machines, Technische Universität München, Germany
²BMW Group

This paper focuses on an efficient method for three-dimensional finite element models with respect to computational time and accuracy of the solution. In the finite element method, the accuracy does not only depend on the level of abstraction from a real physical problem, but is also restricted by parameter and model uncertainties, including material, geometry, boundary conditions and mesh construction. In order to ensure an acceptable solution for a particular application, it is necessary to identify and quantify these uncertainties by performing a sensitivity analysis on the solution. In this work, an experimental modal analysis (EMA) was performed on monolithic and assembled structures to evaluate the results from the finite element models, concerning parameter and model uncertainties.

Keywords: Uncertainty quantification, finite element modeling, experimental modal analysis

1. INTRODUCTION

One challenging task in virtual modeling is the simplification process in such a way that accurate results are ensured with minimum effort in modeling and computation [1]. The major issue is that every finite element model possesses some degree of uncertainty due to topological and material parameters, initial and boundary conditions, forcing terms, etc. The results from a finite element method can be reliable only when the method is validated with available data from measurements or analytical models. General recommendations regarding model uncertainties are given in [2], which are important for model formation as well as in the development process. A powerful method in the numerical simulation of engineering problems that considers uncertainty is the stochastic finite element method (SFEM), which is an extension of the deterministic finite element method (FEM) in a random framework [3]. The application of the method on various practical engineering problems has been studied in many works, such as [4–9]. In summary, the sensitivity of the solution to all physical parameters and their uncertainties has to be clear. This can be possible exclusively with highly meticulous identification and quantification of these parameters [10]. A further step is to identify the effect of the abstraction level in the virtual model on the simulated result.

In this paper, monolithic and bolted structures are studied. Ibrahim and Pettit [11] provide a good review about the structural dynamic behavior of bolted structures. In the following, the sensitivity of several influencing variables at the joint area to the eigenfrequencies or resonance frequencies are considered. A bolted beam structure and a simplified component of an engine-transmission assembly are examined. A laser Doppler vibrometer was used for the experimental modal analysis (EMA). The commercial software Abaqus/CAE was used for FEM simulations.

This paper is organized as follows: Section 2 explains the test cases. Section 3 gives an overview of the computational models. Results and discussions from experimental and numerical test cases are presented in Section 4. Section 5 finalizes the paper with a short conclusion.

2. TEST CASES

The specimens tested in this work are presented in Figure 1. Figure 1(a) shows a bolted beam structure. We speak here of beam problems, one dimension is much larger than the other two dimensions. The beam samples have the nominal length \( L = 0.2 \text{ m} \), width \( B = 0.04 \text{ m} \) and height \( H = 0.004 \text{ m} \). In Figure 1(b), a...
bolted crank case and a cylinder block are shown. This model represents a simplified geometry of real engine components.

nominal dimensions: 0.2m x 0.04m x 0.004m

(a) Assembly - bolted beam structure  (b) Assembly - bolted crank case and cylinder block

Figure 1 – (a): Test specimen - steel (C45); assembly - bolted beam structure; nominal dimensions: 0.2m x 0.04m x 0.004m; type of steel screws: M10 x 16-10.9; tensile strength $R_{tu} = 1000$ N/mm, yield strength $R_{e} = 900$ N/mm; (b): Test specimen - aluminum casting; assembly - bolted crank case and cylinder block (CC/CB); type of steel screws: M8 x 186-10.9.

3. MODEL DESCRIPTION

In this section, we discuss the details of finite element and experimental models.

3.1 Finite element model

The finite element model of the bolted beam structure is shown in Figure 2.

The mesh consists of second-order brick elements (C3D20). To minimize the numerical error, no distorted elements were allowed in the meshing process. The boundary conditions are ideal free-free. The connections between the external thread of the screws and the core thread of the beam and between the beam and the relevant screw head have been defined as “tie constraint” as shown in Figure 2. These constraints represent a surface based technique for modeling a contact. In this method, a kinematic coupling of all degrees of freedom between two nodes in different surfaces is considered [12]. The high pressure zone is located between the contact surfaces close to the screw joints.

In this work, the numerical modal analysis is performed using two modeling techniques: first, the tie-constraint technique as explained above; second, a two step analysis, in which a static load case before the modal analysis is defined to cover the screw pre-load force. At this step, instead of tie-constraints for the high pressure zone, the contact surface of the beams consisted of normal and tangential stiffnesses.
3.2 Experiment (EMA)/physical model

The results from experiments highly depend on the measuring technique and the precision of the apparatus. Therefore, the technique to apply the excitation force and measure the response of the structure should be carefully chosen. For this work, non-contact measurements are optimal, since they provide the smallest error in the Frequency Response Function (FRF). A common way to excite the structure is by an acoustic source and measuring the sound pressure throughout the structure. In this case, a force response function is achieved [13]. This method, however, has a limited exciting energy and can only be used for small structures. Another possibility is excitation via a magnetic field, in which a force is defined on the structure [14]. This method is limited to conductive specimens. Another option is to creating an excitation pulse with a hammer. Here, a very short pulse duration due to the vertical impact of the hammer on the surface and a sufficient resolution of the pulse signal must be ensured. An excitation of the structure by an electrodynamic shaker must be avoided, if possible, because the connection of the structure with the stingers changes the dynamic behavior of the structure, which influences the measurements. A review of different measurement techniques is given by Ewins [15]. In this work, we use the shaker and the impact hammer for the excitation. The physical model uses experimental modal analysis in which the bolted beam sample is suspended from two elastic strings. The deflection shape is measured by a laser Doppler vibrometer in front of the beam. Figure 3 shows the measurement setup for the bolted beam structure and the bolted simplified engine structure seen from the vibrometer: structure (1), elastic strings (2), shaker (3), impact hammer (4). The eigenfrequencies and the associated bending modes are determined numerically using ME’scope. A similar setup is used for the experimental modal analysis of the simplified engine parts. A shaker excites the structure. The surface velocity is measured contact free by a laser Doppler vibrometer.

4. RESULTS

Initial experimental measurements shows a high influence of the screw tightening torque $M_A$ on the eigenfrequencies of the bolted beam structure. This will be explained in more detail.

4.1 Sensitivity – screw tightening torque $M_A$

Figure 4 shows the sensitivity of the first five eigenfrequencies to the screw tightening torque $M_A$. The eigenfrequencies increase as $M_A$ increases up to 30 Nm. The reason for this is the increased surface pressure in the contact zone. In this area, the surface pressure is significantly enhanced and the coupling for the transmission of vibration between the components is particularly high and better than in other areas. For $M_A = 60$ Nm the eigenfrequencies decrease, except for the first bending mode. One reason for this behavior could be a plastic deformation in the threads and therefore a decrease in stiffness. For the screw tightening torque of 30 Nm, a converged physical model is determined, since it does not cause an increase in the eigenfrequencies and therefore a common value is used in engine development in the automotive industry. Therefore, results
from this case will be compared with numerical solutions. Figure 5 shows the measured eigenfrequencies for the simplified bolted engine structure and their dependency on the screw tightening torque. Three different screw tightening torques of 10 Nm, 20 Nm and 30 Nm and the first resonance frequency are investigated. The increasing resonance frequency at higher screw tightening torques for the engine parts is comparable to that of the bolted beam structure. The deviation of the resonance frequency at \( M_A = 10 \) Nm and \( M_A = 30 \) Nm is 20.4 %. This implies an extremely high sensitivity of the resonance frequency to the tightening torque.

Based on the results in Figure 5 we can determine more general statements on this problem. The higher amplitudes in the resonances with increased screw tightening torques indicate a lower joint damping. With regard to the curve of the resonance frequency at 10 Nm it can be assumed that the joint has a much lower stiffness, since the structure is excited at higher frequency components by a tangential relative movement in the joint. This shows the sudden falling edge after the resonance.

![Graph of measured eigenfrequencies](image1)

**Figure 5 – CC/CB - EMA - Dependence of the resonance frequency to the screw tightening torque \( M_A \)**

### 4.2 Repeatability

This section discusses the repeatability of the determined eigenfrequencies. The first three bending frequencies of the bolted beam structure are considered. After each measurement series the screws are loosened and the beams screwed together again. The position of the screws remain unchanged. In a second series of
measurements, the position of the screws are changed. In each series, a number of ten measurements are done. The bolted beam structure is measured after each screwing procedure. Figure 6 shows the results of each measurement series. The arithmetically averaged relative deviations over 10 samples $\varepsilon_m$ of the first three bending modes are calculated with Equation 1:

$$\varepsilon_m = \left( \frac{f_{\text{max}} - f_{\text{min}}}{f_m} - 1 \right) \times 100,$$

where $f_{\text{max}}$ and $f_{\text{min}}$ are the maximum and minimum eigenfrequencies in one measurement series and $f_m$ is the eigenfrequency averaged over ten measurements. The shaded areas in the diagrams indicate the overall measurement uncertainty in each measurement series. The relative deviation increases at higher eigenfrequencies and reaches a maximum $\varepsilon_P$ of 0.51%. This shows a very high reproducibility of the experimental setup. Measurement results show little sensitivity when screw positions are changed. In other words, loosening and tightening the screws do not change the dynamic behavior of the beam structure significantly. The fluctuations in the natural frequencies of a measurement series are also within the given overall uncertainty of the measurement process.

![Figure 6 – EMA - Repeatability determined eigenfrequencies of bending modes of the screwed beam structure](image)

<table>
<thead>
<tr>
<th>Screw Position</th>
<th>1st Bending Mode</th>
<th>2nd Bending Mode</th>
<th>3rd Bending Mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>Arbitrarily</td>
<td><img src="image" alt="Graph" /></td>
<td><img src="image" alt="Graph" /></td>
<td><img src="image" alt="Graph" /></td>
</tr>
<tr>
<td>Equal</td>
<td><img src="image" alt="Graph" /></td>
<td><img src="image" alt="Graph" /></td>
<td><img src="image" alt="Graph" /></td>
</tr>
</tbody>
</table>

4.3 Sensitivity – non-linearity of joints

In this section we discuss the influence of non-linear effects in the screw joint area on the resonance frequency by focusing on varied screw tightening torques. For this purpose, the crank case and cylinder block are screwed together with tightening torques of 10 Nm, 20 Nm and 30 Nm. The structures are excited with a shaker in the frequency range of the first resonance frequency. For the excitation, a periodic chirp signal is used. Figure 7 shows the results of the first resonance frequency with a constant tightening torque and several excitation amplitudes. With a tightening torque of $M_A = 10$ Nm a dependence of the amplitude and the frequency on the first resonance is determined. However, the influence of the joints dampening and the non-linear stiffness are not separately distinguishable. According to Ewins [15], the frequency shift is caused...
by a non-linear stiffness in the joint area. The dependence of the resonance amplitude to the excitation level is related to the contact damping. Figure 7.c shows a minimum frequency shift with different excitation amplitudes.

Figure 7 – EMA - first resonance frequency of the screwed crank case and cylinder block in dependence on the excitation amplitude (Amp); (a): screw tightening torque $M_A = 10 \text{ Nm}$; (b): $M_A = 20 \text{ Nm}$; (c): $M_A = 30 \text{ Nm}$

### 4.4 Comparison of model types

In the following, experimental results of the screwed beam structure are compared with the solution of three different finite element models. Figure 8 shows the solution of the finite element models (a) - (c) and the measurement values for the first three bending modes. In these models, only the modeling of joint surface between the beams are varied. All other model parameters were identical. In the finite element model (a) the high pressure zone is modeled by tie constraints in an area twice of the thread diameter. This is an empirical value commonly used in the industry for a linear contact modeling. No contact models were defined in other areas in the joint surface.

In the finite element model (b) the calculated diameter $d_{ers}$ for modeling the high pressure zone was used. Using Eq. 2, $d_{ers}$ is obtained as [16]:

$$A_{ers} = \frac{\pi}{4} \cdot \left[ \left( d_w + \frac{l_k}{a} \right)^2 - d_h^2 \right] \quad \text{with} \quad d_{ers} = d_w + \frac{l_k}{a},$$

where $A_{ers}$ is the cross section of the high pressure zone and $d_h$ is the diameter of the through hole. The diameter $d_{ers}$ for the high pressure zone is calculated according to $a$, a material specific constant, $d_w$, the screw head diameter and $l_k$ the clamping length of the clamped parts. The minimum effective screw head diameter $d_{win}$ for a M10 screw is 0.011 m. With a screw head diameter $d_w$ of 0.0168 m, $d_{ ers}$ is 0.01777 m. This value is used in the finite element model (b) for modeling the high pressure zone with tie constraints.

ABAQUS/CAE offers the possibility to consider stiffness effects due to a pre-load force in the eigenvalue computation. To do this, a static load case has to be defined prior to modal analysis. The model solution with modified stiffness matrix is fixed and transferred to the numerical modal analysis. To achieve this, the geometric non-linearity in the calculation process is permitted.
Instead of a tie constraint in the joint area, the finite element model (c) has a contact model with a tangential and normal stiffness for the whole contact zone. The values for this parameter have a small influence on the solution. However, significant variations in the calculated natural frequencies can be obtained, if the tangential stiffness is not defined in the finite element model. The normal stiffness is required for the counterforce to the screw pre-tension force.

Here, from the screw tightening torque of 30 Nm an analytically calculated pre-tension force is required for all seven screws. With the minimum and maximum possible friction coefficients $\mu_K$, $\mu_G$ and the uncertainty in the screw tightening torque of 30 Nm $\pm$ 3%, screw pre-tension forces $F_{\text{VMmax}}$ and $F_{\text{VMmin}}$ are calculated. With these parameters, $F_{\text{VMmin}}$ and $F_{\text{VMmin}}$ are determined to be 19 kN and 11 kN. In the finite element model (c) average values are used.

![Figure 8 – Screwed beam structure: first three eigenfrequencies of bending modes from simulation (FEM) and experiment (EMA); (a)-(c): different modeling strategies of bolted connection in finite element models](image)

It is becoming clear that finite element model (a) is stiffer than the experimental sample. The finite element model (b) shows a good agreement with experimental results for the investigated bending modes. The finite element model (c) with the highest model depth and modeling effort shows deviations of 3.84 %, 1.59 %, and $-0.96 \%$ for the first, second and third bending modes, respectively. In summary, the finite element model (b) has nearly the same accuracy as experimental results, whereas the model (c) has less deviation from the measurement results with respect to the uncertainty of the screw pre-tension load with $\pm$ 26 %. For calculating the eigenfrequencies of bending modes, the diameter $d_{\text{ers}}$ should be used for modeling the high pressure zone.

5. CONCLUSION

The first three eigenfrequencies of bending modes of structures with expanded joint areas can be numerically calculated with an accuracy of $\pm 3.5 \%$. With more accurate numerical results, the diameter of the high pressure zone in the joint area should be calculated with Equation 2. The two-step analysis with a static load case before the modal analysis requires more computational time, although it does not improve the results sufficiently compared to a one-step numerical modal analysis.

The reproducibility of measurement results of screwed structures not depends on how tight the screws bind the structures together. The uncertainty of these variable screwing situations is ten times lower than the
overall measurement uncertainty. The eigenfrequencies show a high sensitivity to the screw tightening torque $M_A$ and thus the preload force. The eigenfrequencies increase as $M_A$ increases up to 30 Nm. The reason for this is the increased high pressure area in the contact zone between the screwed structures. In these areas, the surface pressure is significantly increased and the coupling for the transmission of vibration between the components is particularly high and better than in other areas. The first resonance frequency is more sensitive to non-linearities in the joint area for a low screw tightening torque. In the example discussed here, these influences are determined at $M_A < 10$ Nm.

REFERENCES


