Modeling of damping in bolted structures

Dipl.-Ing. **S. Bograd**, Dr.-Ing. **A. Schmidt**, Prof. Dr.-Ing. habil. **L. Gaul**,
Institut für Angewandte und Experimentelle Mechanik, Stuttgart

Abstract

This paper deals with modeling of the structures with dry bolted joints. Material and joint damping are examined experimentally and are transferred into the FE simulation. The goal of the research study is to predict the damping behavior of the assembled structures without performing experimental modal analysis.

Numerous studies have shown that material damping in metals is nearly frequency independent. This is also experimentally proved with three different materials in scope of this project. Damping estimation is performed according to VDI-Richtlinie 3830 Part 5 on a free vibrating suspended beam. Due to the frequency independence of material damping, the principle of constant hysteresis is used during the FE modeling of the structures. This damping is applied globally for every element in the model.

In contrast to material damping, joint damping occurs only at the joint interface, thus it is modeled locally and is applied only to the interface layer. After the damping parameters are identified and input into the FE model, complex eigenvalue analysis is performed and simulated modal damping is compared with modal damping from experimental modal analysis.

1 Introduction

In order to reduce product development time and associated costs, the industry is shifting toward computer aided design and simulation. This means that the dynamical properties of a structure can be examined before the physical prototype is available. This tactic is useful only if the simulation models provide a reasonably good correlation with the future structure. One of the standard approaches that allows the designer to predict the vibrational behavior of a product is the FEM. In last years it has reached a significant improvement and is used successfully for estimation of the eigenfrequencies and mode shapes of complex assembled structures. However, the prediction of the dissipative properties of a structure is normally avoided or is based on the “rule of thumb”. The goal of this paper is to examine an approach for prediction of damping in a structure.

The damping properties of assembled structures are significantly influenced by the design layout as well as the geometry of its joints. Depending on whether the joint is located near the node or the anti-node of the considered mode shape, it can have either weak or strong influence on the modal damping of the structure. For this reason it is important to model the damping in the joints *locally*. On the contrary, the material damping is spread evenly through the structure, and thus should be modeled *globally*.

Free decay measurements will be made for determination of the material damping parameters. In order to find the local stiffness and damping of the joint, the characteristic part of the joint will be isolated and tested experimentally. It has been shown that damping in the joint in tangential direction significantly exceeds damping in the normal direction, so only tangential joint parameters will be determined [1]. They will be estimated from measured hysteresis curves for various bolt tension and load levels. The loss factor is then given by the ratio between the dissipated energy per cycle and 2π times the maximum stored energy. The joint parameters will be included as the material input during the formulation of the joint elements, which after the assembly will describe the equivalent stiffness and damping behavior of the complete joint.

It has been shown by experimental investigations that the dissipated energy per-load-cycle as well as the material damping is weakly dependent on frequency [2]. For this reason, the principle of constant hysteresis, which assumes that the dissipated energy is completely frequency independent, will be applied.

2 Material Damping Estimation

There are many experimental methods for estimation of the material damping, however in practice many of them will be either prohibitively complex or will not give the required accuracy. This paper deals with damping estimation of the metallic structures, which in general have lower damping in comparison with other engineering materials, thus appropriate methods should be considered.

One of the classical methods is to determine damping from the free vibration decay of the torsionally excited specimen. Material sample would be constrained on one side and excited on the other side containing rigid inertia body. During the testing, it is difficult to constrain the specimen at the mounting point, and dissipation due to friction is larger then the material damping itself. This method would be practical for materials with lower stiffness and higher damping ratios like plastics or for measurement performed on thin rodes.

According to VDI guidelines for experimental determination of damping characteristics, damping can be estimated from free hanging beam [3](Figure 1). The beam should be supported

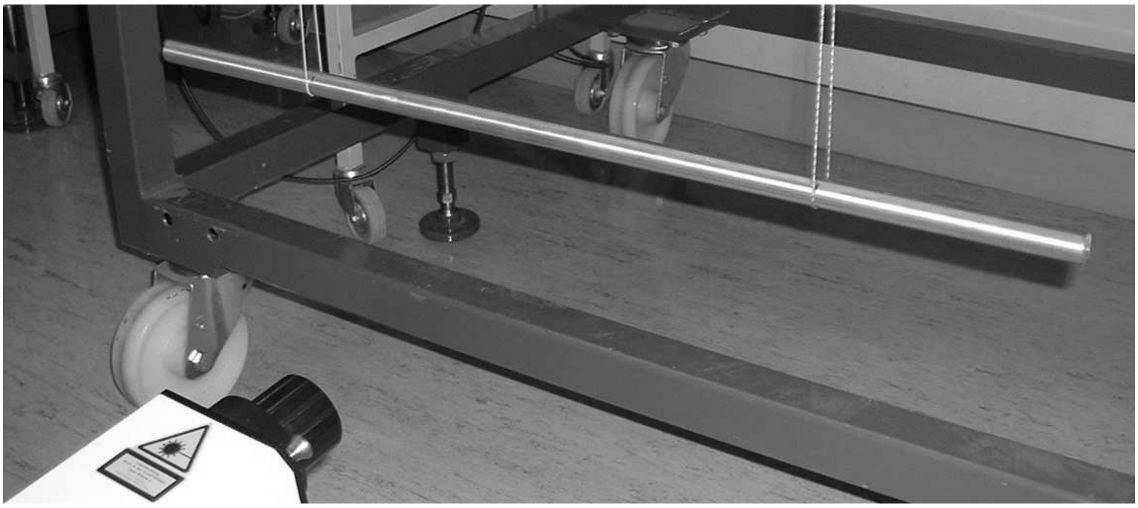

Figure 1: Experimental set-up for material damping estimation

at the nodes of the first eigenfrequency, and is excited with the impulse hammer at the middle of the beam, which corresponds to the anti-node of the first mode shape. In order to minimize the influence of the higher harmonics, the tip of the hammer is chosen for a particular frequency range. Vibration response of the beam is measured contactless with the help of a laser Doppler vibrometer. Then time domain signal is transfered into the frequency domain with the help of an FFT, and the loss factor ϑ is estimated from the frequency response using "half-width" value (Figure 2):

$$\vartheta = \frac{\Omega_2 - \Omega_1}{2\omega_0} \; , \qquad (1)$$

where Ω_1, Ω_2 are angular frequencies corresponding to "half-value" ordinates and ω_0 is the peak of the resonance curve [3]. This method overestimates the damping, in particular in materials where the damping is low. In order to get the best possible estimate from the experimental data, the frequency resolution should be maximized, so that one of the sampling points lies close or at the actual maximum of the response. Good frequency resolution is achieved by longer data acquisition times. This also ensures that the response signal after the impact will decay to zero and during the Fourier transformation the problem of leakage will be avoided [2].

The measurement is performed in three stages. The first step is to survey the scatter of the estimation due to the different measurements and boundary conditions. After initial measurement the test set-up is disassembled, and then reassembled with the same beam two more times during which 5 measurements were performed and the differences were compared. The next step is a comparison between the measurements performed on different samples made of the same material. And finally, investigation of the damping's frequency dependence, which is performed by varying the geometry of samples, is made.

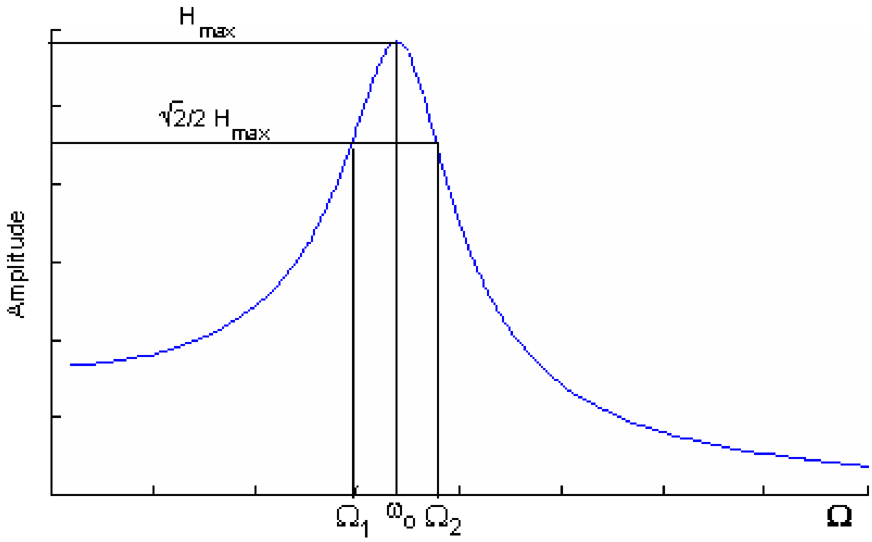

Figure 2: Resonance curve with "half-value" ordinates and corresponding angular frequencies Ω_1, Ω_2, and ω_0

Three materials were used for investigation: aluminum alloy, gray cast iron, and spheroidal cast iron. There were 10 samples from each material, with 5 having diameter 2 cm and length 80 cm, and the other 5 with diameter 5 cm and length 60 cm.

To maximize the frequency resolution, which was 0.0078125 Hz for each measurement, the data acquisition time was 128 seconds. An example of the measurement scatter for an aluminum beam can be seen in Figure 3. Since aluminum has the smallest damping out of the three materials, the measurement scatter should be the largest. In this case the maximum deviation from the mean value is 5.1 %.

Further on, the measurements were performed on different samples made out of the same material. The results were also consistent and deviated maximum by 2.5 % from the mean damping value for a material. Frequency dependency was investigated by measurements on samples with different geometries. An 80 and a 40 cm bar with 2 cm diamter, and a 60 cm bar with 5 cm diameter were used to give different eigenfrequencies for the first modeshape. The results for aluminum are depicted in Figure 4. It can be seen from the figure that the damping is nearly frequency independent, which means that the principle of constant hysteresis can be applied for modeling of the material damping. Gray cast iron and spheroidal cast iron showed similarly only a slight frequency dependence. Averaged values of the measured loss factors for all materials are shown in Table 1.

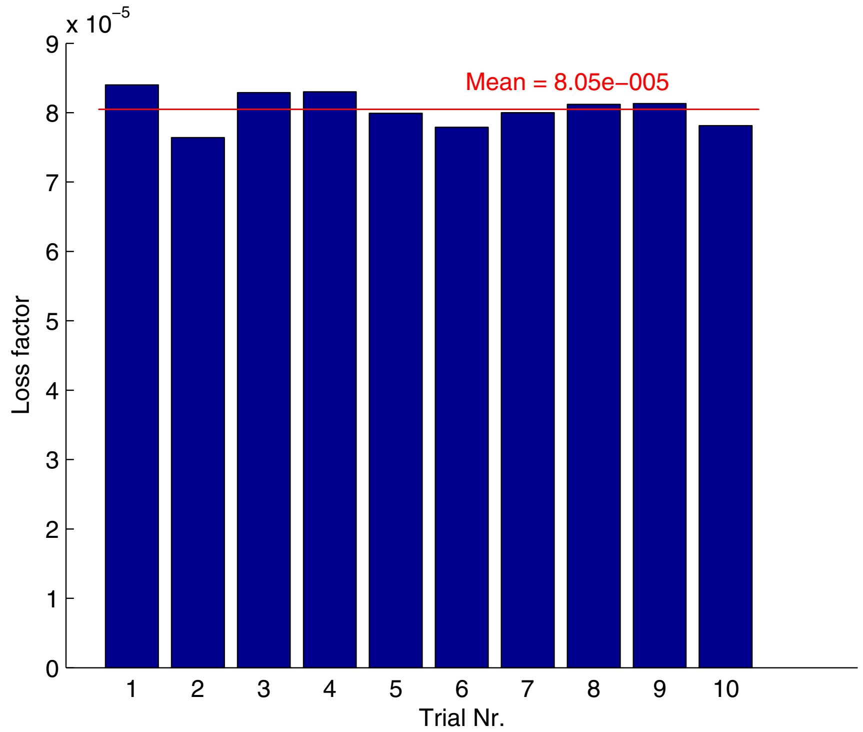

Figure 3: Scatter of the measurement data for aluminum

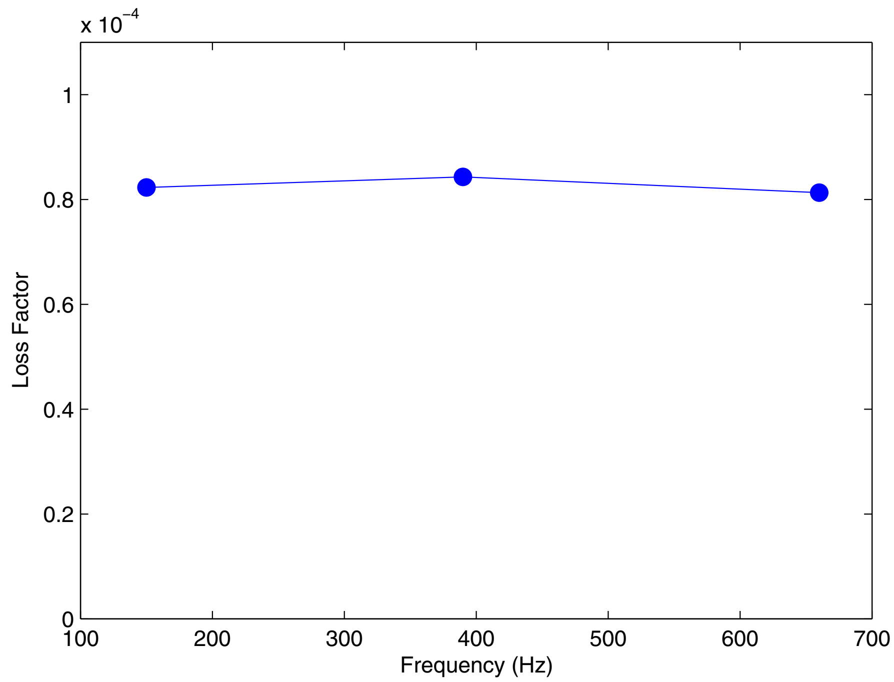

Figure 4: Correlation between frequency and damping for aluminum

Material	Loss Factor ϑ
Aluminum	8.28E-5
Spheroidal cast iron	1.77E-4
Grey cast iron	1.119E-3

Table 1: Measured values of the material damping

3 Joint Patch Damping Estimation

In order to determine stiffness and damping in a joint, a comparable part of the structure is manufactured as an isolated joint. A hysteresis curve, that depends on the displacement and transmitted force in tangential direction, can be evaluated. The ratio of the energy dissipated per one period of vibration W_d, which is equal to the area of the hysteresis curve, and 2π times the maximum potential energy U_max gives the loss factor (Figure 5)

$$\vartheta = \frac{W_\mathrm{d}}{2\pi U_\mathrm{max}} \ . \tag{2}$$

Once the joint parameters are determined experimentally, they are input into the finite element model. After the structure is assembled, the elements give the equivalent linearized description of the stiffness and damping in the joint. The influence of the normal pressure and frequency dependence of the joint parameters should be investigated.

The test set-up is shown in Figure 6. It consists of two rigid masses connected to an isolated joint patch. The masses are supported in their centers of gravity by a thin string. The joint normal pressure is regulated by a bolt with a force measurement ring. In order to produce tangential loading in the joint, one of the masses is excited on one end with a shaker. Acceleration is measured on both sides of the joint with a piezoelectric accelerometer, and after numerical integration of the acquired data, the relative displacement is calculated. The transmitted force is estimated as a product of an acceleration and free hanging mass [4].

The result of the measurements with low normal force is shown on the left-hand side of Figure 7. For this case a constant normal force of 200 N with variable excitation levels is used. At lower excitation levels only the micro-slip occurs. When the excitation level is increased, there is both micro- and macro-slip occurring in the joint. The mean pressure distribution for this case was 33 $\mathrm{N/cm^2}$, however normally in technical structures the joint pressure is between 0.5 and 10 $\mathrm{kN/cm^2}$ and macro-slip should be avoided. A more relevant example for a correspondingly high contact pressure can be seen on the right-hand side of Figure 7. Only the micro-slip occurs in the joint and the dissipated energy is clearly smaller then in the previous case. It can be seen that there are almost no differences between hysteresis curves once a certain normal pressure is reached and only the micro-slip occurs. Both damping and stiffness characteristics for different normal forces are comparable. For this case the loss factor calculated with Equation (2) is 0.018.

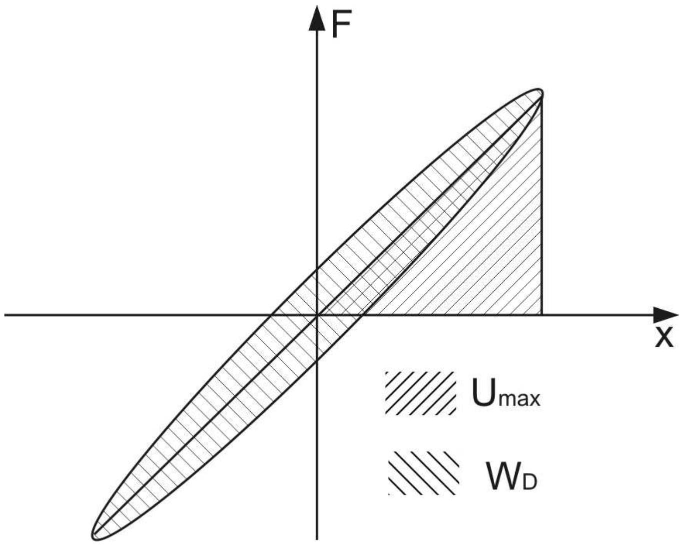

Figure 5: Hysteresis curve

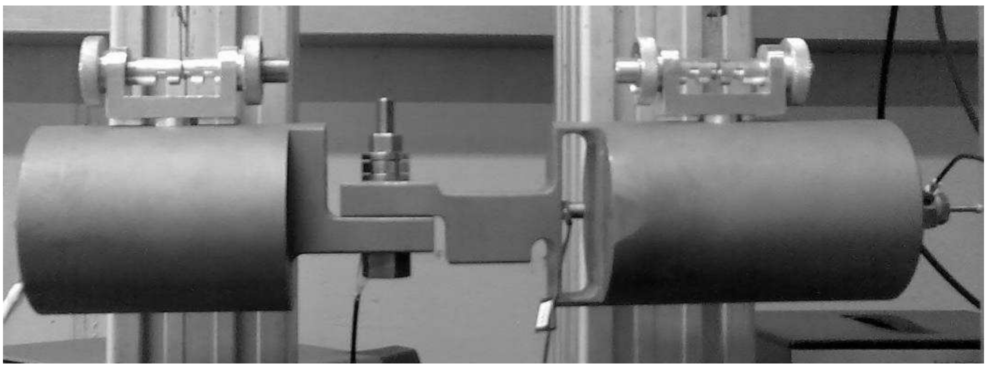

Figure 6: Test set-up for determination of joint patch parameters

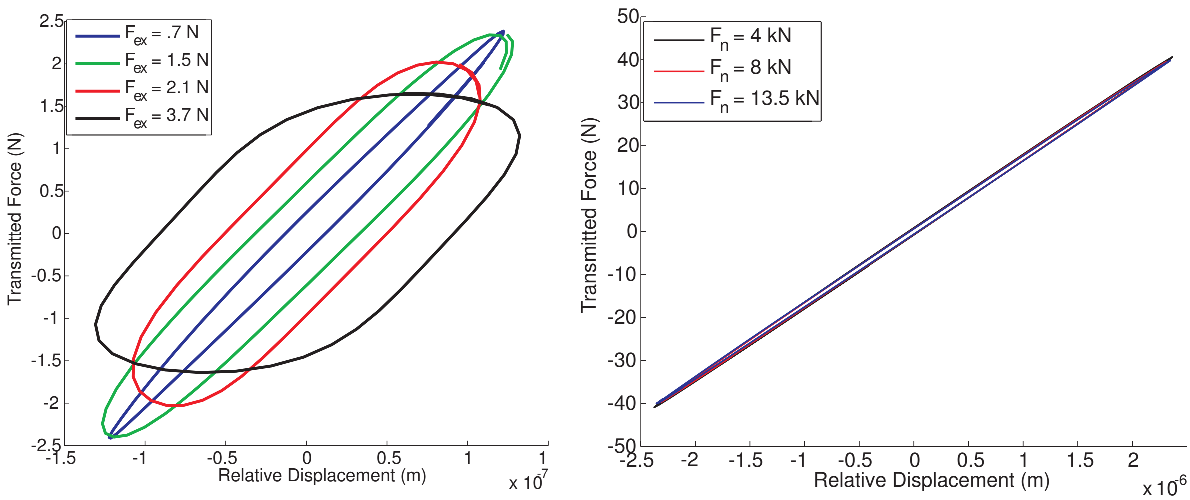

Figure 7: Hysteresis loop with normal contact force of 200 N and varied excitation levels (left); hysteresis loop with varied contact pressure and constant transmitted force (right)

4 Test Structure

A test structure to accompany simulations and experimental investigations is shown in Figure 8. It consists of a 15×12 cm steel plate (thickness 3 mm), which is attached to another plate formed into a U-shape and held together with 5 bolts on both sides. During the testing the structure is assembled with 10 Nm torque, resulting into 9±2 kN normal force. The structure is chosen so that the joint damping would be clearly identifiable from the material damping, and so that the influence of the average normal pressure distribution in the joint could be investigated by varying the number of bolts used for assembly.

An experimental modal analysis with free-free boundary conditions was performed and two exemplary mode shapes can be seen in Figure 9. The mode pictured on the left side has the smallest damping among the measured modes and it is analogous to the damping of the individual parts of the disassembled structure. It can be seen that the deformation occurs on the top and the joint remains mostly unaffected. On the contrary, in the mode shape pictured on the right, there is relative displacement in the joint which increases friction and causes a three-fold increase in damping. This supports the assumption that the joint damping should be modeled locally, where the actual deformation (dissipation) occurs.

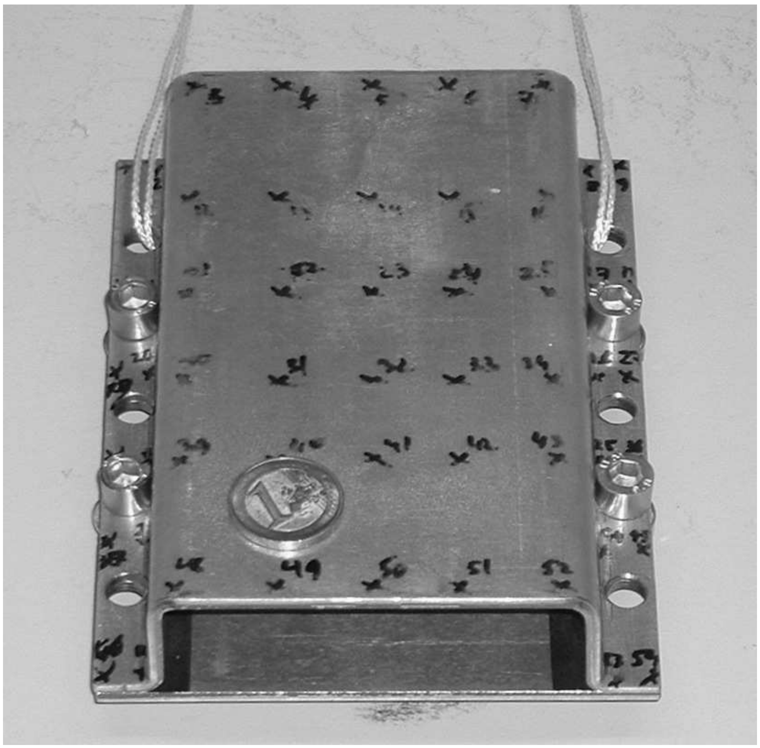

Figure 8: Test structure

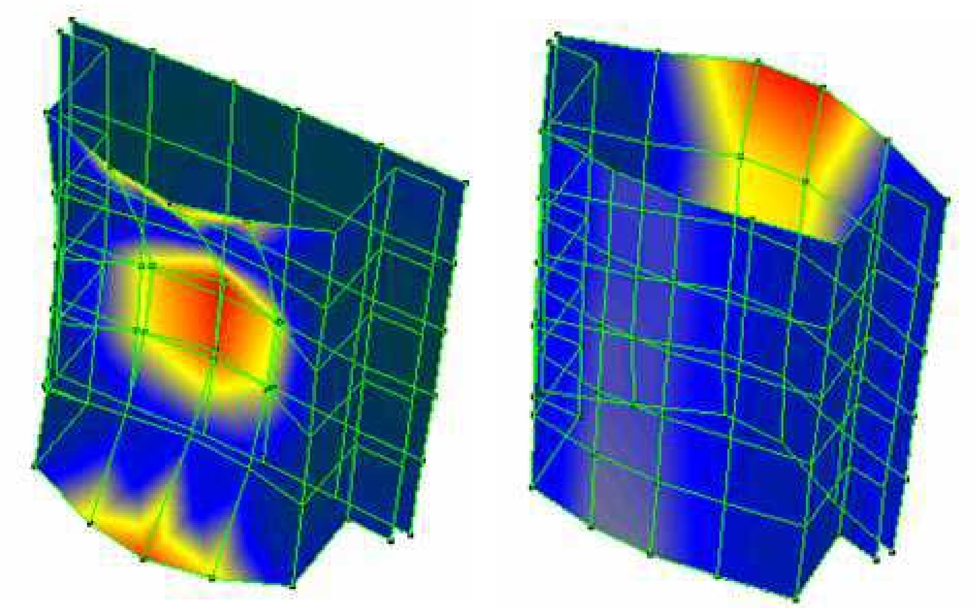

Figure 9: Mode shapes with the smallest (left, modal damping is 0.062 %) and the largest (right, modal damping is 0.191 %) modal dampings in measurement range

Linearized joint parameters which describe average pressure distribution are used during the simulation. Originally the modal analysis on the structure was performed with closely spaced bolt distribution, resulting in nearly constant pressure in the joint. The effect of the non-constant pressure distribution and the ability of the constant hysteresis model to describe such cases adequately has to be investigated. For this purpose testing was also performed on a model with 3 and 2 bolts per side. Table 2 shows results of these experiments. A lower number of bolts causes decrease in structure's stiffness and thereby increase in relative deformation in the joint. For the 3 bolt configuration the maximum eigenvalue shift is lower than 3 % and the damping remains within 39% of the original set-up. However, when the results of 2 bolts

	5 Bolts		3 Bolts			2 Bolts		
Mode Nr.	Freq (Hz)	Damping (%)	Freq (Hz)	Damping (%)	Diff (%) Fr/Dmp	Freq (Hz)	Damping (%)	Diff (%) Fr/Dmp
1	1063	0,1099	1060	0,1270	-0,3 / 16	1030	0,2190	-3,1 / 99
2	1348	0,1911	1320	0,2660	-2,1 / 39	1100	1,6900	-18,4 / 784
3	1441	0,1066	1430	0,1470	-0,8 / 38	1260	0,6910	-12,6 / 548
4	1558	0,1466	1520	0,1890	-2,4 / 29	1380	0,1670	-11,4 / 14
5	2149	0,1428	2100	0,1700	-2,3 / 19	1800	1,5300	-16,2 / 971
6	2307	0,0766	2320	0,0966	0,6 / 26	2280	0,3410	-1,2 / 345
7	2447	0,0863	2450	0,0974	0,1 / 13	2410	0,1380	-1,5 / 60
8	2559	0,0619	2550	0,0653	-0,4 / 5	2550	0,1610	-0,4 / 160

Table 2: Eigenvalues and corresponding modal damping values from experimental modal analysis for different bolt combinations; % difference in frequency and damping between the 5 bolts set-up and 3 and 2 bolts set-ups is shown

structure are examined, the structure's stiffness changes up to 18% causing the damping for some modes to increase by a factor of 10. Also, while both frequency and damping change with the decreasing number of bolts, it can be seen that damping is a much more sensitive parameter, making its estimation more difficult.

5 FE simulation

It has been shown by experimental investigations that joint damping is nearly frequency independent [2, 5, 6]. Similar results have been shown for material damping in metals, where the main cause of dissipation is inner friction in the material [7, 8]. These results can be also concluded from the investigations performed for this project (Section 2) [9]. So during the FE modeling the principle of constant hysteresis will be used. Such model makes sense only in frequency domain, while in time domain it leads to non-casual material behavior [10, 11]. Some investigations to this model have been made already, and show good correlation with experimentally determined joint parameters [12, 13, 14].

During the calculation of the vibrational characteristics of the structure with the FEM the following equation of motion for an undamped system is used

$$\underline{M}\,\underline{\ddot{u}} + \underline{K}\,\underline{u} = \underline{0}\,, \tag{3}$$

where $\underline{M}$ is a mass matrix, $\underline{K}$ is a real valued stiffness matrix, and $\underline{u}$ is the displacement vector. Eigenvalues and eigenmodes can be determined by performing the numerical modal analysis with a standard FE software. Using the principle of constant hysteresis, the damping will be incorporated into stiffness matrix by augmenting it with the product of experimentally determined dissipation multipliers α_i and β_i and complex stiffness matrices containing respective material and joint elements

$$\underline{K}^* = \underline{K} + \mathrm{j}\,\underline{D} = \underline{K} + \mathrm{j}\sum_{i=1}^{n}\alpha_i\underline{K}_i^{(\mathrm{Material})} + \mathrm{j}\sum_{i=1}^{m}\beta_i\underline{K}_i^{(\mathrm{Joint})}\,, \tag{4}$$

where for the considered systems with low damping $\alpha_i\,,\beta_i \ll 1$. This equation can be solved for complex eigenvalues and eigenmodes with some commercial FE packages, and in this case was performed with MSC.Nastran. Modal damping for the structure can be estimated from the solution.

Thin layer elements are used for the simulation of joints in the FE model. Experimentally determined contact stiffness and dissipation parameters are used as material properties of these elements. Thin layer elements are normal hexahedral elements in which length or width to thickness ratio can up to 1:1000 without causing numerical problems during the calculation [15].

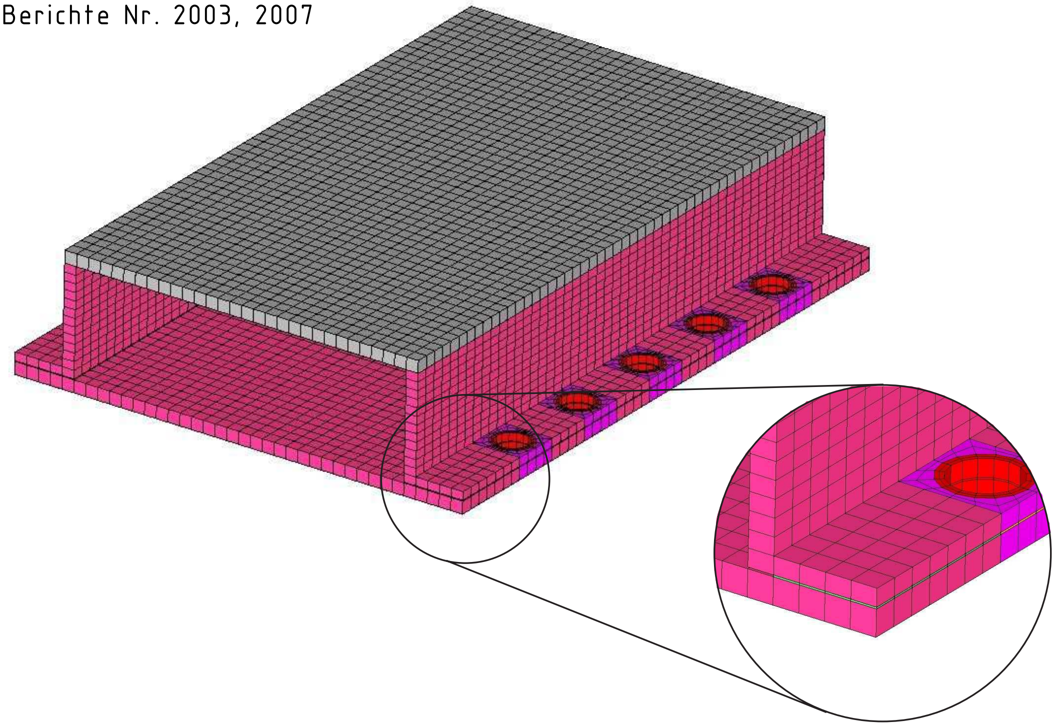

Figure 10: FE Model of the tested structure with thin layer elements between two parts

Good results were already shown by implementation of thin layer elements in numerical modal analysis in FEM [16, 17]. In cited works, authors used model updating of the contact stiffness with already existing experimental models. In this project the goal is to predict the stiffness as well as the damping in the joints based on the experimental data from the isolated joint experiment.

The simulated structure can be seen in Figure 10. It was modeled with isotropic hexahedral elements, with thin layer elements having a 1:15 length to thickness ratio. In Nastran hysteretic damping can be input as a structural element damping coefficient on a material property card. In this case material damping is applied to the elements belonging to the components of the structure, and joint damping is realized by application of damping to thin layer elements. In scope of this project, due to the time constrains, it was not realizable to transfer the experimental joint parameter data into the FE simulation. Material damping was taken as the damping acquired from individually tested components, and joint damping was estimated by model updating of the simulated structure with experimental modal analysis of the assembly.

Comparison between the experimental and simulated results for frequencies and damping for the 5 bolts structure can be seen in Table 3. A good correlation for eigenfrequencies of measured and simulated data is achieved. The maximum deviation is 3.2 %. The highest deviation for damping is 17 % which, taking into consideration that damping is a much more sensitive parameter, is also a good result.

Simulation for a 3 bolt structure was also performed, and the results are quantitatively similar to a 5 bolts assembly. However, trying to simulate a 2 bolts structure brings out limitations of the method. The relative friction in joints is too large and non-linear effects start to occur.

Mode Nr.	Experimental Freq (Hz)	Simulated Freq (Hz)	Difference (%)	Experimental Damp	Simulated Damp	Difference (%)
1	1063	1070	0,7	0,110	0,107	-2,9
2	1348	1370	1,7	0,191	0,161	-15,9
3	1441	1425	-1,1	0,107	0,116	8,4
4	1558	1605	3,0	0,147	0,149	1,3
5	2149	2218	3,2	0,143	0,153	7,1
6	2307	2256	-2,2	0,077	0,090	17,5
7	2447	2447	0,0	0,086	0,089	2,7
8	2559	2553	-0,2	0,062	0,067	9,0

Table 3: Eigenvalues and corresponding modal damping values from experimental modal analysis and simulation for the 5 bolts structure

Also, it has been shown by the FE joint contact calculations, that contact pressure due to the bolt loading is concentrated in the immediate vicinity of the bolt, approximately twice the bolt radius [18]. So averaged pressure distribution used for the simulation does not represent the physical system well, when the bolt spacing is large. To approximate this effect, simulations were performed, where the joint was split into two regions with two different joint parameters: in the vicinity of the bolt and between the bolts. This technique raises the correlation with experimental data for 3 and 5 bolt structures to about 2 % for frequencies and 14 % for damping. There are also improvement for a 2 bolts structure, however it is still not possible to achieve results comparable to the other assemblies. The differences remain at maximum 7 % for frequency and 75 % for damping.

6 Conclustions

This paper gave an overview of an approach for prediction of damping in assembled structures with help of the Finite Element modeling. For this purpose there were experiments performed in order to find the damping and stiffness parameters of the joint and dissipative characteristics of the material used for the structure's construction.

An approach was tested on a simple steel structure, which consisted of two parts joint by 5 bolts on two sides. In order to examine the effect of pressure distribution in the joint, measurements on structures with 3 and 2 bolts were performed as well. The results of experimental modal analysis have shown that the damping in the structure is dependent on the deformation in the joint occurring due to a certain mode shape. This means that the joint damping should be modeled locally, and material damping globally.

Experimentally determined joint parameters were not utilized during this work yet, and parameters for simulation were based on model updating with experimental structure. Good correlation between experimental and simulated results has been achieved for the structures with 5 and 3 bolts. However, non-linear effects start to occur in a 2 bolts structure, making it difficult to predict the damping with a linear constant hysteresis approach.

The future work on this topic will concentrate on determination of experimental joint parameters and their application in the simulation. Also, it would be interesting to perform further research on parameters which affect the damping and its sensitivity and define what can be considered as a “good” correlation for damping between experiment and simulation.

7 Acknowledgment

The support of the project FVV Nr. 877: 'Experimentelle Ermittlung von Kennwerten zur Werkstoff- und Fügestellendämpfung sowie deren Berücksichtigung in Finite-Elemente-Berechnungen' by the Forschungsvereinigung für Verbrennungskraftmaschinen (FVV) which enabled us to perform these investigations is gratefully acknowledged.

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