Joint damping prediction by thin layer elements

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ABSTRACT

This paper deals with modeling of the assembled metallic structures with dry bolted joints. The damping parameters are found experimentally from a generic isolated joint test bench. The influence of joint parameters, such as normal contact force and frequency dependence is examined. Once the joint parameters are found, they are input into a finite element model. An assembled structure is modeled with thin layer elements on the joints' interfaces. These layers contain the measured stiffness and damping parameters. A simple structure is used to support the experimental and simulation work. The results of the experimental modal analysis are compared with numerical simulations. The sensitivity of the joint parameters, normal contact force distribution, and orthotropic behavior of the thin layer elements are examined.

Nomenclature

σ, ε, τ, γ	stress, strain, shear stress, shear angle
η	loss factor
j	imaginary unit
a	acceleration
M	mass, mass matrix
k, K, K^*	stiffness, stiffness matrix, complex stiffness matrix
E	values of the element's stiffness matrix
D	damping matrix
G	shear modulus
u, d, A	displacement, thickness, area
α	dissipation multiplier
F	force

1 Introduction

Modern FEM software is capable of simulating the dynamics of complex structures. Estimation of eigenfrequencies and mode shapes can be made before the prototype is available. However, the amplitude of the vibration and the resulting acoustic properties depend not only on the eigenfrequencies and modes but are significantly affected by damping. Never the less, in the industry 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, which will allow an engineer to estimate product's behavior in the design phase.

The damping properties of assembled structures are significantly influenced by the 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*.

In order to find the stiffness and damping of the joint, the characteristic part of the joint is isolated and tested experimentally. It has been shown that damping in the joint in tangential direction significantly exceeds damping in normal direction, so in this paper only tangential joint parameters are considered [1]. They are estimated from measured hysteresis curves for various bolt tension and excitation 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 interface is modeled with the thin layer elements and its parameters are included as the material input during the formulation, which after the assembly describes the equivalent stiffness and damping behavior of the complete joint.

It has been shown by the 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 Joint Patch Damping Estimation

In order to determine the stiffness and damping in a joint, a representative part of the structure is manufactured as an isolated joint. Figure 1 shows the drawing of the measurement set-up. It consists of two masses M_1 and M_2 connected by the lap joint, which has the same surface finish and normal force distribution as the joints of the assembly. The masses are supported in their centers of gravity by a thin string. The joint's normal pressure is regulated by a bolt with a force measurement ring. Mass 2 is excited by an electrodynamic shaker with a sine input. Once the vibration reached a constant level, the accelerations a_1 and a_2 on both sides of the joint are measured with piezoelectric accelerometers. These signals are integrated twice with respect to time to receive the displacement. By taking the difference between them, the relative displacement in the joint is calculated. The force transmitted by the joint is calculated as a product of the mass M_1 and its acceleration a_1 [3]. Knowing the transmitted force and relative displacement a force-displacement diagram, also called a hysteresis curve, is constructed. The slope represents the tangential stiffness in the joint. 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 2)

$$\eta = \frac{W_\mathrm{d}}{2\pi U_\mathrm{max}} \ . \tag{1}$$

Once the joint parameters are determined experimentally, they can be input into a finite element model. After the structure is assembled, thin layer elements are used to represent 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 is investigated.

Two versions of the test set-up are used. Besides the test set-up as shown in Figure 1, a modified set-up is used, see Figure 3. This construction is made with a leaf spring, allowing two masses to move with system's resonant frequency, thereby increasing axial excitation level. The experiment without spring requires a more careful alignment of the masses with the shaker, so that the masses move axially with excitation and bending in the joint is minimized. However, this set-up allows to study joint parameters at different frequencies, and not just at the resonance.

The result of the measurements with low normal force is shown on the left-hand side of Figure 4. For this case a constant normal force of 16 $\mathrm{N/cm^2}$ 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. However, in a typical application the joint pressure is between 0.5 and 20 $\mathrm{kN/cm^2}$ and macro-slip should be avoided in order to preserve the functionality of the product (e.g. cylinder head). A more relevant example for a correspondingly high contact pressure can

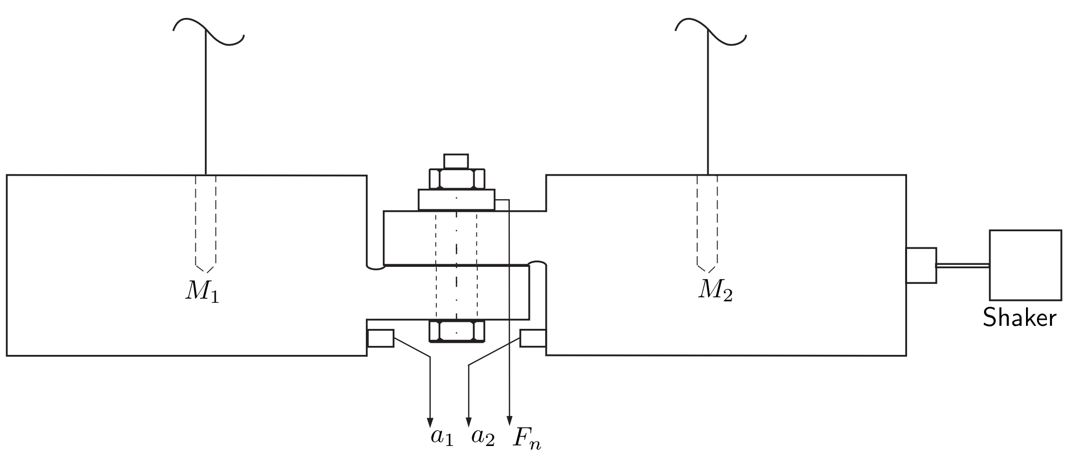

Figure 1: Layout of the measurement set-up for the determination of the tangential stiffness of the joint

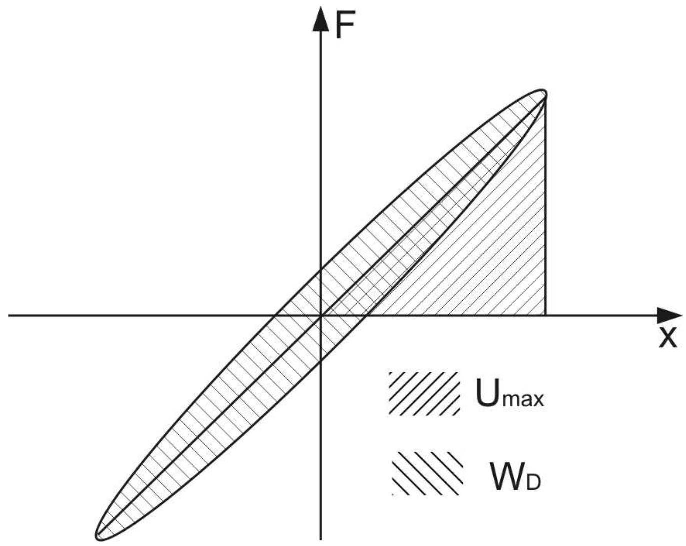

Figure 2: Hysteresis curve

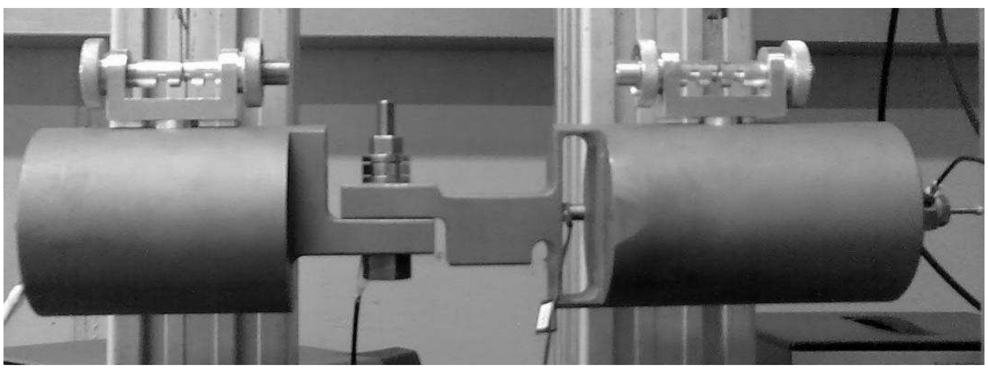

Figure 3: Test set-up for determination of joint patch parameters

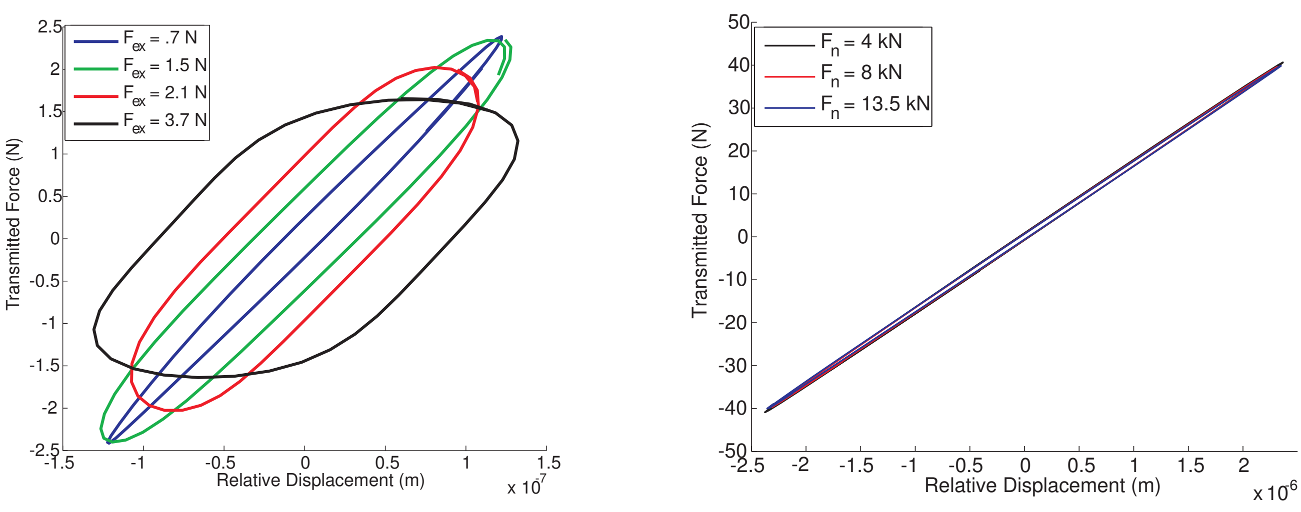

Figure 4: Hysteresis loop with normal contact force of 200 N and varied excitation levels (left); hysteresis loop with varied contact pressure and constant transmitted force (right)

be seen on the right-hand side of Figure 4. Only 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.

When the joint pressure is high, the excitation level does not have a significant effect on damping or stiffness of the joint, as can be seen on the left side of Figure 5. The slope of the the curves and calculated loss factor remains the same with increased excitation levels. The results of the frequency variation are displayed on the right hand side of Figure 5. A slight frequency dependence is seen both in damping and stiffness, however it was not significant and a principle of constant hysteresis can be applied for the modeling of joint damping.

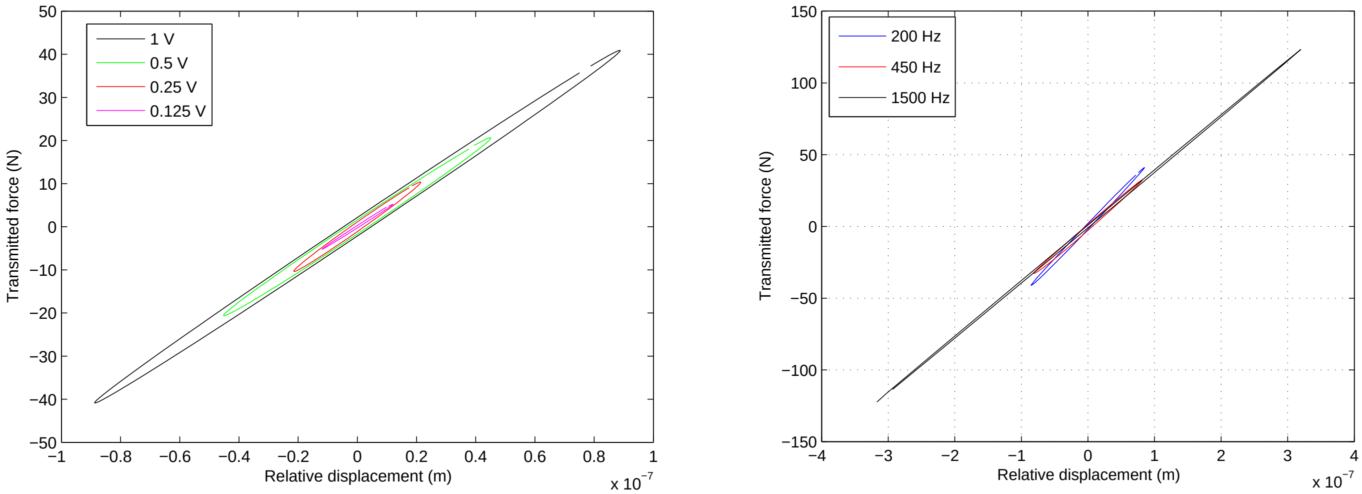

Figure 5: Hysteresis loop with normal contact force of 19 kN and varied excitation levels (left); hysteresis loop with varied excitation frequency (right)

3 Test Structure

A test structure to accompany simulations and experimental investigations is shown in Figure 6. It consists of a 15×12 cm steel plate (thickness 0.3 cm), which is attached to another plate formed into a U-shape and held together with 5 bolts per side. 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 is performed and two exemplary mode shapes can be seen in Figure 7. 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.

Linearized joint parameters which describe the average pressure distribution are used during the simulation. Originally the modal analysis on the structure was performed with closely spaced bolt distribution, so that only the micro-slip could occur. The effect of the non-constant pressure distribution with wider spacings between the bolts 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 1 shows the results of these experiments. A lower number of bolts causes a decrease in structure's stiffness and thereby an 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 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 a decreasing number of bolts, it can be seen that damping is a much more sensitive parameter, making its accurate estimation more difficult.

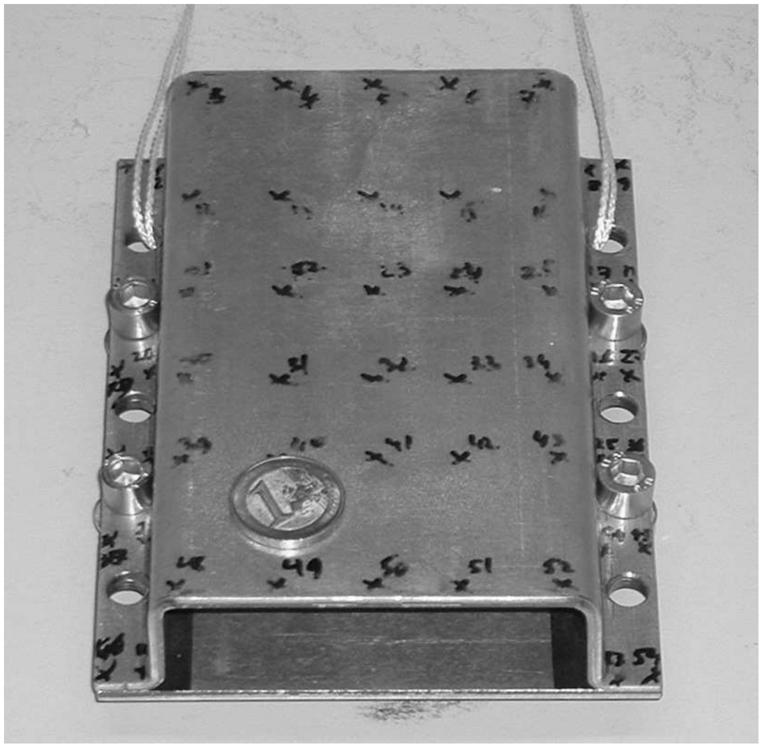

Figure 6: Test structure

4 FE simulation

It has been shown by experimental investigations that joint damping is nearly frequency independent [2, 4, 5]. Similar results have been shown for material damping in metals, where the main cause of dissipation is inner friction in the material [6]. These results can be also concluded from the investigations performed for this project (Section 2) [7]. 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 [8, 9, 10]. Some investigations to this model have been made already, and show good correlation with experimentally determined joint parameters [11, 12, 13].

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

$$\underline{M}\,\underline{\ddot{u}} + \underline{K}\,\underline{u} = \underline{0}\,, \tag{2}$$

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 a numerical modal analysis with a standard FE software. Using the principle of constant hysteresis, the damping is incorporated into stiffness matrix by augmenting it with the product of experimentally determined dissipation multiplier α_i and complex stiffness matrix containing joint elements

$$\underline{K}^* = \underline{K} + \mathrm{j}\,\underline{D} = \underline{K} + \mathrm{j}\sum_{i=1}^{n} \alpha_i \underline{K}_i^{(\mathrm{Joint})}\,, \tag{3}$$

where for the considered systems with low damping $\alpha_i \ll 1$. This equation can be solved for complex eigenvalues and eigenmodes with some commercial FE packages, and in this case is performed with MSC.Nastran. Modal damping for the structure is calculated by the software and can be compared with an experimentally acquired damping.

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 be up to 1000:1 without causing numerical problems during the calculation [14].

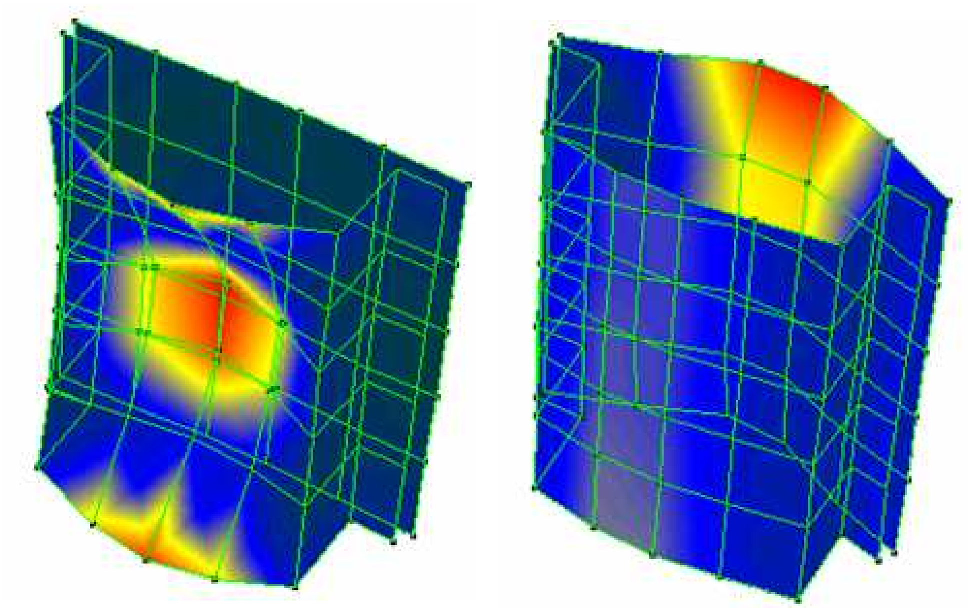

Figure 7: Mode shapes with the smallest (left, modal damping is 0.062 %) and the largest (right, modal damping is 0.191 %) modal dampings in measurement range

	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 1: 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

Good results for a joint's stiffness modeling were shown by implementation of thin layer elements in numerical modal analysis in FEM [15, 16]. In cited works, the 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 stiffness of the generic joint from Section 2 should be transferred into the FE model as a parameter of the thin layer elements. A schematic of an arbitrary joint modeled by a thin layer is depicted in Figure 8. The force F acting on both sides of the joint produces a shear stress τ in the layer. This stress can be expressed as

$$\tau = G\gamma \approx G\frac{u}{d}, \tag{4}$$

where G is the shear modulus and γ is the shear angle. The shear stress can also be estimated by the ratio of the applied tangential force F and area of the contact A

$$\tau = \frac{F}{A}\,. \tag{5}$$

By combining Equations (4) and (5) the force can be calculated

$$F \approx \frac{GA}{d}u = ku\,. \tag{6}$$

The stiffness k in the Equation (6) is obtained from the generic lap joint experiment.

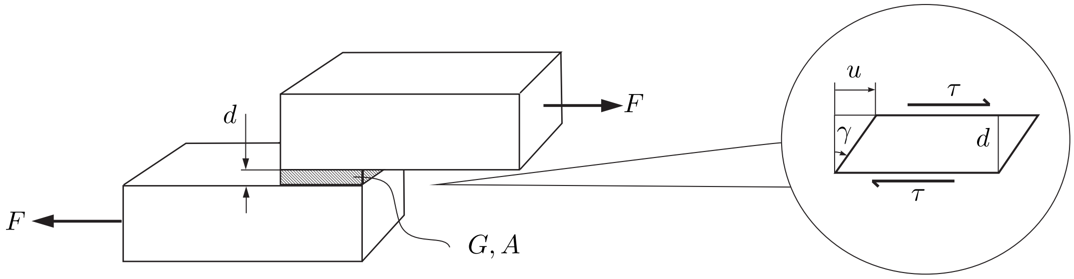

Figure 8: Schematic of a joint with a thin layer

It is important to recognize that the characteristics of the joint's interface that is represented by the thin layer elements is *orthotropic*. There are significant differences in tangential and normal behavior of the joint.

From Equation (6) with known stiffness k of the lap joint, the shear modulus can be approximated as

$$G = \frac{kd}{A}\,. \tag{7}$$

When changing the thickness d of the elements, the shear modulus must be updated accordingly. Normal stiffness is not determined experimentally, but can be either estimated from contact analysis or like in this case is based on the experience.

The orthotropic matrix input into the software is of the following form

$$\begin{bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \sigma_{xy} \\ \sigma_{yz} \\ \sigma_{zx} \end{bmatrix} = \begin{bmatrix} E_1 & E_{12} & E_{13} & 0 & 0 & 0 \\ & E_2 & E_{23} & 0 & 0 & 0 \\ & & E_3 & 0 & 0 & 0 \\ & & & E_4 & 0 & 0 \\ & & & & E_5 & 0 \\ & & & & & E_6 \end{bmatrix} \begin{bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \\ \varepsilon_{xy} \\ \varepsilon_{yz} \\ \varepsilon_{zx} \end{bmatrix} \tag{8}$$

The off-diagonal terms are zero for physical reasons (there is no transversal contraction invoked by the contact interface). Also, since the interface has no stiffness in x and y direction (parallel to the joint's surface), the terms E_1 and E_2 disappear. E_3 represents the normal stiffness, whereas $E_5 = E_6$ define the tangential stiffness of the joint. Since the joint exhibits no stiffness for in-plane shearing, E_4 also is zero. For numerical reasons all diagonal terms in Equation (8) must be different from zero. Thus, in the calculation, the values for E_1, E_2 and E_4 are set to some small values that are at least by a factor of 10^{-2} lower than the corresponding normal and tangential stiffnesses of the joint. For this simulation $E_3 = 184N/cm^2$ and $E_5 = E_6 = 2000N/cm^2$.

The simulated structure can be seen in Figure 9. It is modeled with isotropic hexahedral elements, with thin layer elements having a 40:1 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

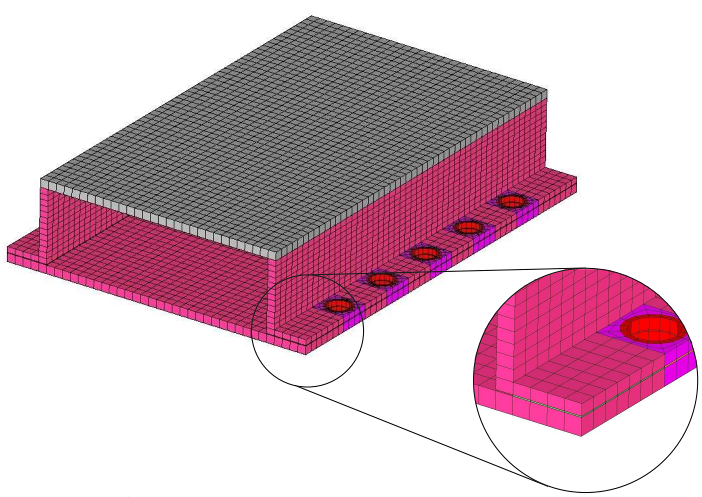

Figure 9: FE Model of the tested structure with thin layer elements between two parts

of the structure, and joint damping is realized by application of damping to thin layer elements.

A comparison between the experimental and simulated results for frequencies and damping for the 5 bolts structure can be seen in Table 2. A good correlation for eigenfrequencies of measured and simulated data is achieved, with the maximum deviation of 2.7 %. The highest deviation for damping is 87.7 % which, considering that damping is a much more sensitive parameter, is also a good result.

Mode Nr.	Experimental Freq (Hz)	Simulated Freq (Hz)	Difference (%)	Experimental Damping (%)	Simulated Damping (%)	Difference (%)
1	1063	1057	-0,5	0,110	0,107	-2,5
2	1348	1339	-0,7	0,191	0,204	6,9
3	1441	1406	-2,4	0,107	0,114	7,1
4	1558	1567	0,6	0,147	0,178	21,6
5	2149	2155	0,3	0,143	0,179	25,1
6	2307	2244	-2,7	0,077	0,072	-6,1
7	2447	2428	-0,8	0,086	0,065	-24,9
8	2559	2531	-1,1	0,062	0,026	-58,0
9	3372	3363	-0,3	0,116	0,110	-5,3
10	3713	3742	0,8	0,076	0,009	-87,7

Table 2: Eigenvalues and corresponding modal damping values from experimental modal analysis and simulation for the 5 bolts structure

A harmonic analysis with FE model is performed and the results are compared with the frequency response functions obtained from the experiment. A comparison of the response at a driving point is shown in Figure 10. There is a good correlation of the curves up to the 7th mode. Slight discrepancies are seen for mode 3 and 6, where the difference in frequencies between experiment and simulation is 2.4 and 2.7 % correspondingly.

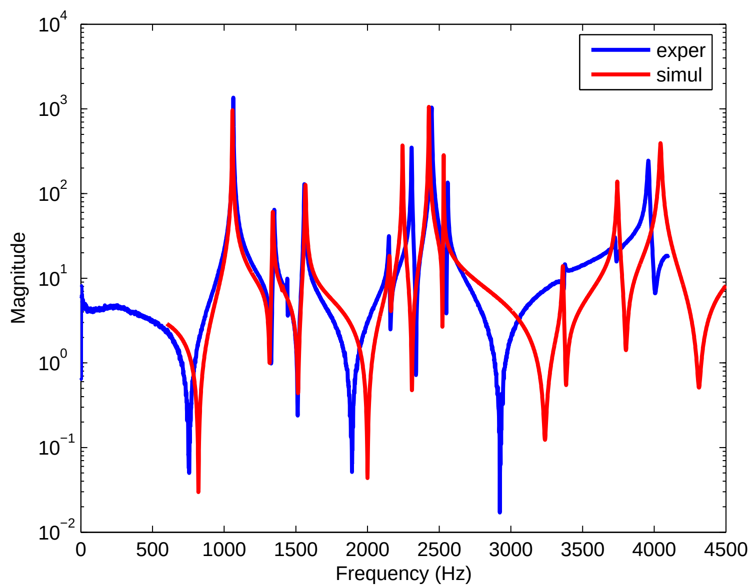

Figure 10: Comparison of simulation and experimental frequency responses for driving point measurement

A simulation for a 3 bolt structure is 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. 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 [17]. So an 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 are performed, where the joint is split into two regions with two different joint parameters: in the vicinity of the bolt and between the bolts. This technique slightly raises the correlation with experimental data for 3 and 5 bolt structures, but more work needs to be done in this direction.

5 Conclustions

This paper gives an overview of an approach for the prediction of damping in assembled structures with help of a Finite Element simulation. For this purpose there were experiments performed in order to find the damping and stiffness parameters of the joint. Then the joint's interface was modeled with thin layer elements, which contain the acquired experimental parameters.

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, where as material damping should be modeled globally.

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 approach shows good results for the joints with homogeneous pressure distribution, but is not suited for the joints were the pressure is concentrated around some regions, so the limits of the method should be examined. The future work on this topic will also concentrate on the sensitivity analysis of joint parameters and the determination of normal stiffness. An application of the method to more complicated structures is being performed as well.

6 Acknowledgment

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