DAMPING IN BOLTED JOINTS

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SUMMARY: With the help of modern CAE-based simulation processes, it is possible to predict the dynamic behavior of fatigue strength problems in order to improve products of many industries, e.g. the building, the machine construction or the automotive industry. Amongst others, it can be used to improve the acoustic design of automobiles in an early development stage.
Nowadays, the acoustics of automobiles plays a crucial role in the process of vehicle development. Because of the advanced demand of comfort and due to statutory rules the manufacturers are faced with the challenge of optimizing their car's sound emissions. The optimization includes not only the reduction of noises. Lately with the trend to hybrid and electric cars, it has been shown that vehicles can become too quiet. Thus, the prediction of structural and acoustic properties based on FE-simulations is becoming increasingly important before any experimental prototype is examined. With the state of the art, qualitative comparisons between different implementations are possible. However, an accurate and reliable quantitative prediction is still a challenge.
One aspect in the context of increasing the prediction quality of acoustic (or general oscillating) problems - especially in power-trains of automobiles - is the more accurate implementation of damping in joint structures. While material damping occurs globally and homogenous in a structural system, the damping due to joints is a very local problem, since energy is especially dissipated in the vicinity of joints.
This paper focusses on experimental and numerical studies performed on a single (extracted) screw connection. Starting with experimental studies that are used to identify the underlying physical model of the energy loss, the locally influencing parameters (e.g. the damping factor) should be identified. In contrast to similar research projects, the approach tends to a more local consideration within the joint interface. Tangential stiffness and energy loss within the interface are spatially distributed and interactions between the influencing parameters are regarded. As a result, the damping matrix is no longer proportional to mass or stiffness matrix, since it is composed of the global material damping and the local joint damping. With this new approach, the prediction quality can be increased, since the local distribution of the physical parameters within the joint interface corresponds much closer to the reality.

KEYWORDS: Damping, Joints, Hysteresis, Energy Loss

1. INTRODUCTION

In a research project in cooperation with the automotive industry the prediction quality of models with mechanical joints should be increased. In particular, the aim is a more exact and reliable prediction of acoustic problems in power-trains of automobiles. Therefore, the implementation of damping in mechanical joints in CAE-simulations should be improved. On the one hand, experimental studies should help understanding the physical behavior and, on the other hand, improvements of existing modeling techniques should lead to more reliable and more exact predictions than possible with the present state of research. In this paper the results of experimental and numerical studies will be presented and an approach to identify joint damping will be described.
The quality and predictability of acoustic (or general oscillating) problems with the help of modern CAE-based simulations is depending on several factors, e.g. the chosen approach for the calculation, the knowledge about the structural properties or the implementation of external and internal excitations including possible interactions between each other. Furthermore, the kind of consideration of damping plays a crucial role. Thus, damping (and especially joint damping) was the topic of a lot of research work in the last decades. Ibrahim and Pettit [1] as well as Wentzel [2] give an extensive overview of the problems pertaining to structural dynamics with bolted joints and existing models for the numerical implementation. However, it is still a challenge to perform an accurate and reliable prediction.
Geisler [3] describes in his dissertation an implementation using zero-thickness contact elements with an elastic slip model. With this approach, it is possible to take into account the distribution of the normal pressure in the joint and, consequently, the stiffness distribution in tangential direction. Further on, parameters are identified

from the numerical model with the help of a model correction method based on computed and experimental measured frequency responses. Although Geisler states that his approach gives good results, he also notes that his algorithms show sometimes no convergence. Furthermore, he works with a simple two beam system and an adaption for more complex systems is not possible without additional effort. Additionally, he suggested that the identified parameters should be compared with other approaches, e.g. hysteresis measurements. Because the pressure distribution plays a crucial role, the usage of a simple joint model with a nearly homogeneous normal pressure distribution is to be recommended.
In a VDI guideline [4] an approach based on experimental studies of hysteresis measurements is described. A single screw connection is analyzed experimentally with varying boundary conditions (force amplitude, bolting torque, etc.). Based on the hysteretic measurements, energy loss factors and tangential stiffness values can be identified. Gaul et al. [5-10] are using this approach for their experimental studies and have implemented the identified values into a numerical model for a simulation in frequency domain. A consideration of the pressure distribution within the joint interface is not - or only very coarse - done. The solution is computed by performing a complex modal analysis and, subsequently, by a harmonic analysis based on the complex modal damping values. With this approach, the results of a simulation of an automobile's engine block could be clearly enhanced, but there are still differences between the experimentally measured damping values and the computed damping values up to 60%.

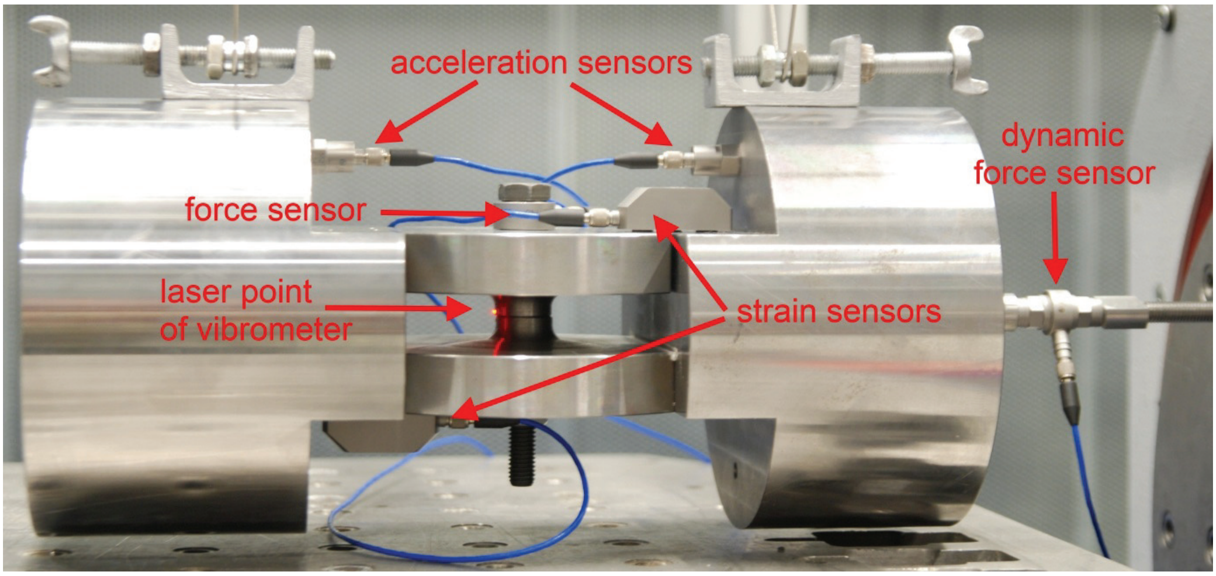

Figure 1 - experimental setup connected to a shaker (right) with acceleration, strain and force sensors

2. PRIMARY CONCEPTION

The VDI approach [4] was the starting point for further investigations in this research project. A similar experimental setup was designed in order to measure the hysteresis that characterizes the energy loss of the joint. The setup consists - analogous to the VDI setup - of two screwed specimens where each specimen is, again, screwed with a counter mass (Figure 1). These masses are necessary for producing a tangential force within the joint of the specimens because of their inertia when the setup is excited with a shaker. With this setup, it is easily possible to test other specimens, e.g. other materials or geometries, since only the specimens have to be replaced. The vertical joint between the masses and the specimens is not critical according to Schmidt et al. [6], because it is stressed in normal direction and so the energy loss is significant smaller than in the tangential stressed joint between the specimens.
The basic idea of this approach is to measure the friction hysteresis depending on the relative displacement Δu of the screwed specimens and the transmitted force F_T in tangential direction (shown exemplary in Figure 2 with a viscous damping hysteresis curve). The area of the hysteresis loop corresponds with the energy dissipated in one period of vibration W_D and a loss factor can be calculated by dividing W_D by 2π times the maximum potential energy U_{max} [5].

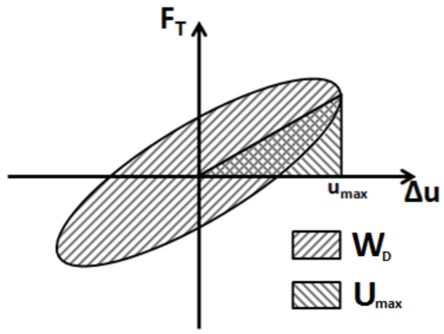

Figure 2 - example of a hysteresis curve with viscous damping with dissipated energy per one period of vibration W_D and maximum potential energy U_{max}, according to [5]

The normal force within the joint interface is directly associated to the tangential stiffness of the connection by Coulomb's Law, where the frictional stiffness is depending on the normal pressure and a material specific coefficient of friction. Hence, it is important to have a homogeneous distributed normal force in the joint. Otherwise you will get unspecific information in the hysteresis measurements, because there are maybe areas with low pressure and therewith a low tangential stiffness. These areas show much earlier slip than areas with bigger normal pressure. Consequently, in these areas energy will get lost, but you do not know exactly where it is dissipated. Hence, the results of an experimental hysteresis measurement with an uneven normal pressure distribution are not really transmittable to other systems. Thus, in contrast to the approach of Gaul et al. [5-10], the normal distribution in the joint interface should be homogeneous - as it is also suggested by Geisler [3]. Therefore, a study of multiple specimens' geometries was accomplished and a specific radiused form of the adapter was found that fulfills this requirement in static load case.
Another difference in comparison to the VDI approach concerns the measuring technology. Due to inaccuracies in the measurement chain the total error of the identified damping values is stated with ±50% in Gaul's implementation. Furthermore, a validation with other kinds of measurement methods was not accomplished. Due to this, additionally to the suggested acceleration sensors, other sensor types should be used in this research project to measure the relative displacement between the specimens. The results from these experimental measurements are summarized in the following chapter.

3. EXPERIMENTAL STUDIES

In the context of the research project, different kinds of sensor types were used to determine the relative displacement between the specimens. The aim was to validate the experimental results among each other and to find the best method to measure the relative displacement between the specimens that is needed to identify the hysteresis. One method to determine the relative displacement - as described in the VDI approach - is to measure the accelerations on both counter masses. Subsequently, the accelerations will be integrated twice to get the absolute displacements and, then, the difference is calculated.
Beside the suggested acceleration measurement, we tried another method including the measuring of the strain with the help of a strain sensor over the gap between a specimen and the counter mass of the other part. The measured elongation corresponds to the relative displacement between the specimens. Additionally, non-contact measurement methods with laser vibrometers were performed. One the one hand, a differential fiber optic interferometer with two sensor heads was pointed directly to the specimens in the vicinity of the joint. The advantage of this method is the optical processing of the difference signal. On the other hand, for the purpose of comparison, two 3D laser vibrometer systems were used to measure the displacement of the upper and the lower specimen separately.
While the absolute displacement signal of all sensor types shows comparable results within the measurement accuracy, the ascertained relative displacement varies with the kind of the sensor (Figure 3). For example, there was a difference up to factor 10 between the relative displacement calculated with the help of the acceleration sensors and the displacement that is optically processed with the interferometer. However, further tests with the interferometer at measuring points on the counter masses verify the results of the acceleration sensors. Hence, it could be concluded that measurements of the relative displacement on the counter masses does not represent the real relative displacement of the joint. Because of the distance to the joint, the relative signal will be disturbed significantly.

Beyond, it turns out that the usage of the strain sensors is unsuitably, since the extra connections between the specimens and the counter masses are stiffening the structural system. As a consequence, the natural frequencies and modes will be influenced and there is also an effect to the relative displacement as interferometer measurements show with and without the strain sensors.

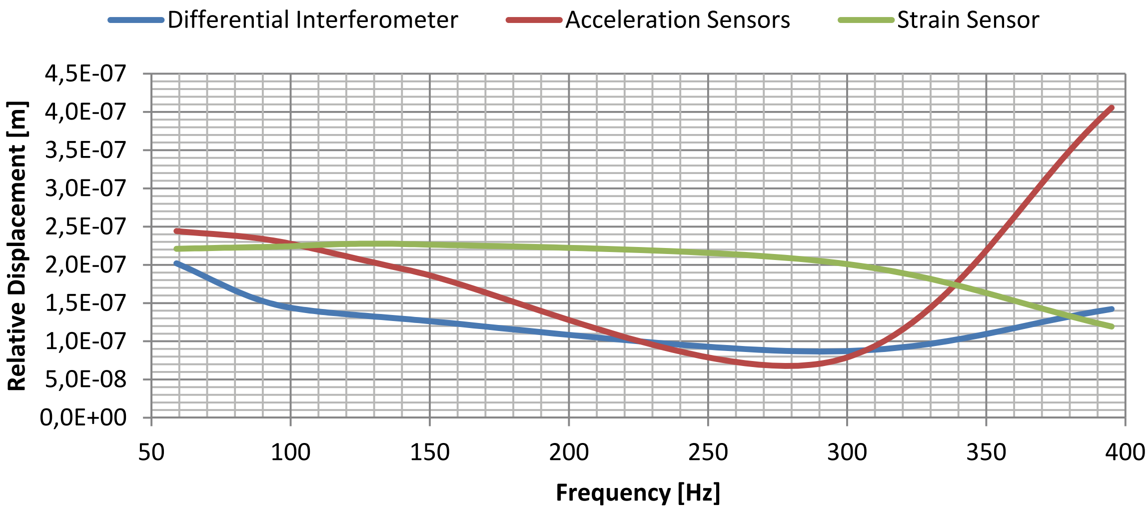

Figure 3 - Determined Relative Displacements of Differential Interferometer, Acceleration Sensors and Strain Sensors

The relative displacements of the differential interferometer and the 3D vibrometer show comparable results. However, it turns out that the distance of the measurement point has also a big influence to the result. Just a few millimeters distance from the joint plane leads to displacements much higher than displacements measured nearer to the joint. Unfortunately, it is not possible to measure the displacement with the vibrometer directly at the joint, because the laser point itself has a certain geometrical dimension. However, the best results could be determined with the non-contact measurement methods.

4. NUMERICAL STUDIES

Parallel to the experimental studies, numerical simulations with the experimental setup are executed in the context of a transient analysis. The joint interface was modeled with a frictional contact model according to Coulomb's law for dry friction and the setup was loaded with a harmonic force analogue to the experiment. The screw pretension was applied with a thermal load. The simulation is divided into two steps. In the first (quasi static) step, the thermal force is applied to simulate the bolt load. This leads to a normal pressure distribution in the joint interface that agrees with the distribution theory of technical literature. In the second step, the bolt force stays constant and the setup is to be excited with the harmonic load that shall produce the tangential load within the joint.

In Figure 4 the normal pressure distribution is depicted for three time steps of such a simulation in time domain that shows micro slip effects. This means that in some areas of the joint the maximum tangential stiffness will be exceeded and so slip will occur there. However, globally the contact stiffness is not exceeded. Consequently, the joint shows no relative displacement in the macro scale, yet. At the beginning of the load cycle, when the harmonic force is zero and there is only an influence of the screw pretension, you can see a nearly homogeneous distributed normal pressure over the radius. In further time steps, you can see a concentration of the normal pressure in the lower and upper part of the joint depending on the harmonic force (which acts in the figures downwards). Nevertheless, the screw pretension is so strong that there are no regions where the pressure becomes zero. In other words, there is no gap between the specimens.

Just as the normal pressure, the frictional stresses in the joint are not homogeneous distributed, while the setup is excited with the harmonic force. Figure 5 shows the frictional stresses when the harmonic load is increased from zero to its maximum. As there is a concentration on the side for the analyzed geometry, the elements in the axis of excitation show significant smaller stresses.

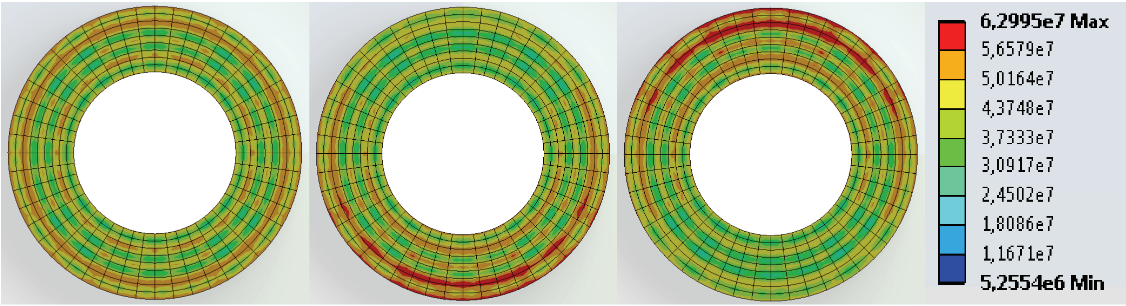

Figure 4 - Normal Pressure [Pa] at harmonic force of 1kN and screw pretension of 10kN
Left: at the beginning of the load cycle (force=0) / Middle: at maximum tension /
Right: at maximum pressure of the harmonic force

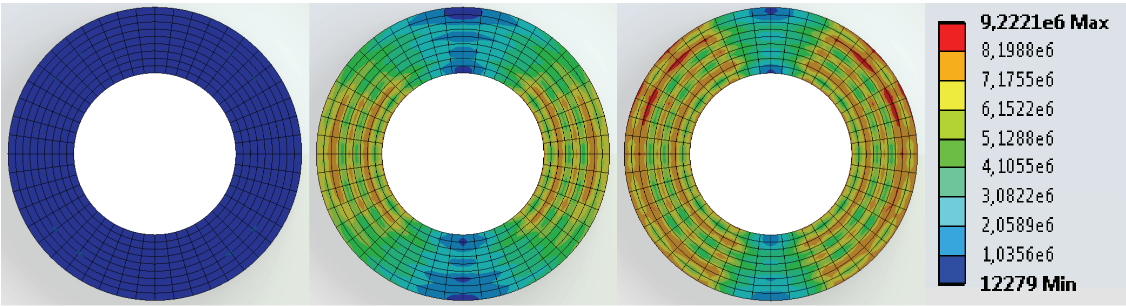

Figure 5 - Frictional Stress [Pa] at harmonic force of 1kN and screw pretension of 10kN
Left: at the beginning of the load cycle (force=0) / Middle: increasing the harmonic force /
Right: maximum frictional stresses

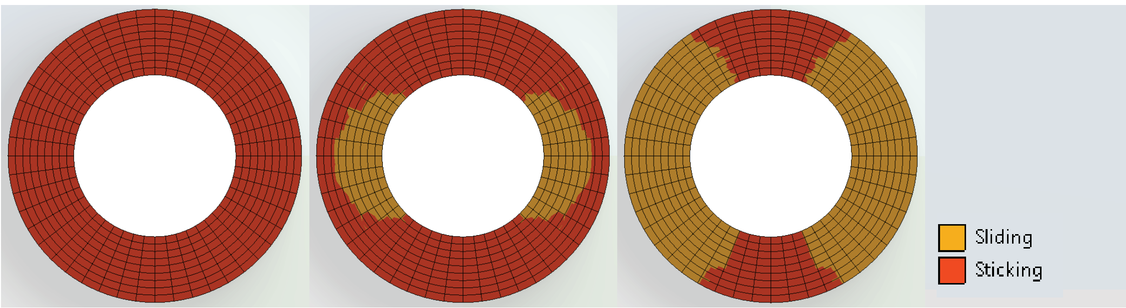

Figure 6 - Slip Status at harmonic force of 1kN and screw pretension of 10kN
Left: at the beginning of the load cycle (force=0) / Middle: increasing the harmonic force /
Right: at maximum frictional stresses

You can determine the slip status of an element - whether it is sticking or sliding - if you compare the maximum tangential stiffness with the actual frictional stresses. If the stresses are smaller, then the contact pairs are sticking. Otherwise sliding will occur. Of course, as long as not the complete joint shows sliding state, the slip is limited by elastic deformations. Such a slip status distribution for the presented simulation is shown in Figure 6. You can see that the slip starts on the side and at the inner boundary of the joint when the frictional stresses increase due to the dynamic load. By increasing the harmonic force, the slip reaches the outer boundary of the joint. If the harmonic force would be increased further, all elements would show the sliding state and the whole joint would start to slip, which is also known as macro slip. Then, the deformation is irreversible and only limited by the friction. One must note that the results are strongly depending on the chosen geometry. Another configuration, e.g. a bigger contact interface, would possibly lead to a decreasing normal pressure at the outer boundary. Under these circumstances, slip could start for example at the outer boundary.
Besides, the influence of the distance of the measuring point to the joint interface was examined, which is very considerable, as you can see in Figure 7. Three curves show the relative displacement over the time measured at different locations at the joint. The red curve shows the relative displacement calculated by subtracting the

absolute displacements three millimeters above and respectively below the joint interface (measuring points *B1* and *B2*). These measuring points agree with the experimental measuring points for the differential optic interferometer (see chapter 3). Further on, the blue curve is calculated by the difference of adjacent nodes on the inner boundary of the joint (measuring points at *C*) and the black curve shows the relative displacement of adjacent nodes on the outer boundary (measuring points at *A*). While the red curve is nearly a sine curve, the blue curve has a rectangular form. Furthermore, the amplitude of the red curve is about five times as big as the blue one. The black curve is always zero, since at this location the sliding state is never reached (see also Figure 6). This means, that the measurement of the relative displacement aside the joint interface gives results that do not represent the actual behavior. Reasons for the difference between the red curve and the other ones are elastic deformations. This is underlined by the phase shift between the red and the blue curve. While the red curve agrees with the harmonic load representing elastic deformations, the blue curve shows changes only when the red curve reaches its maximums and minimums. In other words, the start of the slip in the blue curve is recognizable, when the harmonic force reaches a force, where the maximum tangential stiffness is locally exceeded. The slip ends after the maximum of the harmonic force is reached and the tangential stress will decrease.

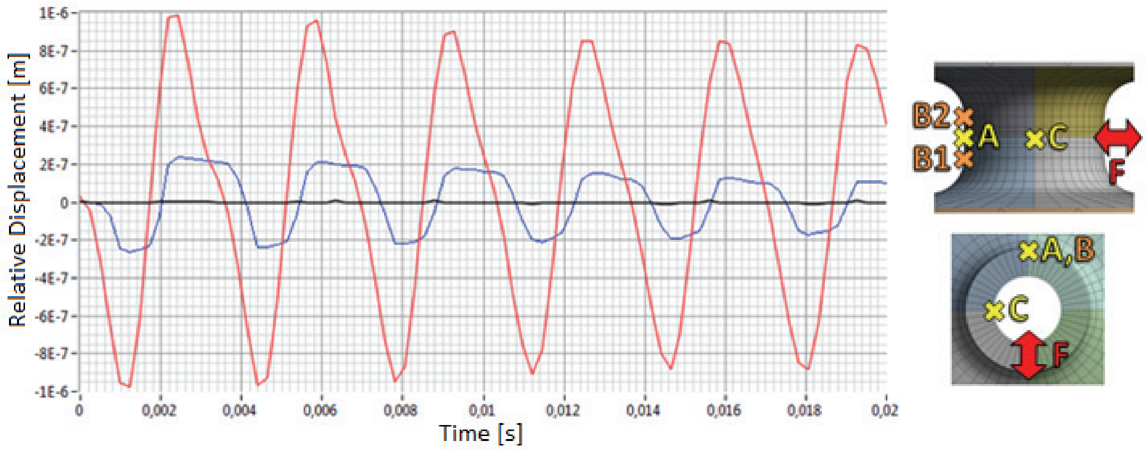

Figure 7 - Relative displacement at harmonic force of 1kN and screw pretension of 10kN
Black: Relative displacement at measuring points *A*, Blue: at *C*, Red: difference of *B1* and *B2*

Recapitulating, the primary conception of a homogeneous normal pressure distribution seems to be invalid, since it is varying with the dynamic load. Furthermore, the distribution of the frictional stresses leads to micro slip, where some areas - depending on the geometry - show relative motion and others not. In comparison, experimental results agree with results from the simulation at measuring points *B1* and *B2*. However, it seems to be obvious that the results from these measuring points did not represent the actual relative motion within the joint. Moreover, the experimental measuring of the relative displacement is hardly possible, since it is varying in the joint interface and elastic deformations falsify the results significantly. This means, that it is hardly possible to make a clear statement about the damping factors and tangential stiffness's with the help of experiments, because the boundary conditions within the joint are still varying. As a conclusion a more local consideration within the interface is necessary. This approach is described in the next chapter.

5. CONCLUSION AND OUTLOOK

The primary intention was to measure experimentally the energy dissipation depending on homogeneous distributed boundary conditions. Unfortunately, as already mentioned, one can assume that a homogeneous distribution is hardly to reach. Besides, the experimental measurement is very sensitive in regard of the measuring position. Consequently, such a global view to the joint is not really useful. A more local consideration within the joint interface seems to be appropriate.
The idea of this approach is to find a model, where the pressure distribution, the tangential stresses and the coefficient of friction can be identified reliably. One point concerns the improvement of the experimental setup in order to minimize the measuring error due to the distance to the joint and due to elastic deformations. Maybe, therefore, a fundamental change of the experimental setup is necessary.

Another way would be to identify the damping parameters from a numeric reference model depending on the local boundary conditions of each element in the joint, respectively. Outgoing from the simplified experimental specimen with a single (tangential loaded) joint the physical behavior is investigated. A physical model can be identified and implemented into a numerical model in time domain. In order to consider the strongly nonlinear dissipative effects within the joint, different kinds of damping models and effects (e.g. Coulomb model, viscous friction, or stiction effects) can be implemented in order to find the best equivalent model in comparison to the experimental studies. In the context of a validation, the developed numerical reference model is verified by comparing displacements all over the whole experimental setup. The aim is to find a numerical model that shows the same behavior than the experimental model using a model updating strategy.

The tangential stiffness at a specific position within the joint interface is depending on the normal pressure at this position. However, the energy dissipation in the joint is depending on the amount of slip that is distributed over the joint interface. It is locally influenced by the normal pressure and tangential stress at each position, respectively. But, it is also depending on the global utilization level of the tangential stiffness in the whole joint influenced by the individual geometry of the structural system. In other words, it can be described as a function of the ratio of tangential stress to tangential stiffness locally, and as function of the ratio of areas with sticking state to areas with sliding state, globally.

With the help of the numerical reference model it is possible to identify the tangential stiffness and damping parameters performing a parameter variation (e.g. variation of bolting torque, the force amplitude, or the coefficient of friction). The results of these studies are analyzed for each element in the joint depending on the mentioned local and global parameters and stored for later usage in numerical predictions with other structural systems.

The next step is to implement the identified parameter relationships in thin layer elements. As described in Gaul's approach [5-10] the maximum tangential stiffness can be included using an orthogonal material law. Damping can be considered by arranging a damping matrix with portions of the global material damping and the local joint damping. As a result, the damping matrix is no longer proportional to mass or stiffness matrix. Consequently, in frequency domain the equation of motion cannot be solved on the conventional way, but it can be solved for example by performing a complex modal analysis.

Gaul sets the stiffness and damping parameters globally within the joint to a specific value depending only on the material. In comparison to his approach, the parameters are distributed in the new local approach according to their local and global influencing boundary conditions, which have to be determined in advance by performing a static simulation with the applied bolting torque and external and internal forces.

6. SUMMARY

With the help of experimental studies, it could be shown that measuring the relative displacement of the specimens is very sensitive to the measuring point. The bigger the distance to the joint, the bigger is the measured relative displacement. Moreover, the relative signal will probably be falsified by elastic deformations as could be shown in numerical simulations of the experimental setup.

Further numerical investigations show that it seems to be unrealistic going out of a spatially homogeneous normal pressure distribution and homogeneous frictional stresses during a dynamic excitation - at least for the investigated specimens. Because of this, it will come to micro slip effects that are strongly depending on the geometry of the joint interface. Since it is unknown, where exactly the energy is dissipated, you will get an averaged result and a transfer to other joints including different boundary conditions is not possible.

Therefore, a new approach with a more local consideration of the damping and stiffness parameters within joints has to be developed. This approach should lead to models with spatial distributed stiffness and damping values that is more realistic than a homogeneous distribution.

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