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# VARIABILITY IN MEASURED RESONANCE FREQUENCIES AND LOSS FACTORS OF A BOLTED PANEL STRUCTURE

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## ABSTRACT

*Structural bolted joints are not perfectly contiguous or rigid. Instead, the joint stiffness, mass, and damping depend on many parameters including the joint thickness, number and size of bolts, the bolt preload, and the motion of the jointed region. Also, damping can vary with motion type (stick-slip for friction and impact for joint opening and closing) and amplitude. Finally, all joint dynamic parameters vary from installation to installation. We measured the variability of bolted joint stiffness and damping for four low-order modes of a bolted Aluminum panel structure – two flexural modes and two twisting/torsional modes. While a large portion of the bolted joint literature features a lap joint or an assembly of two beams, there are few studies dealing with flanged connections of plates. Two flanged rectangular panels were bolted together with two cap screws threaded into one of the flanges. Modal resonance frequencies and loss factors were tracked throughout a decaying vibration response induced by hammer impacts on the structure. Measurements were made for three bolt torques and two flange thicknesses, three impact force strengths, and three repeated assemblies (the panel was unbolted and bolted together again). Relative motion at the joint with respect to mode shape determines whether the joint primarily adds stiffness or mass, as well as the amount of added damping. Not surprisingly, reducing bolt preload reduces resonance frequencies. Only minor variability in resonance frequency and loss factor were observed for most conditions, except for cases with low bolt preload. In these cases mild nonlinearity is evident in the response, where resonance frequency decreases and loss factor increases with increasing vibration amplitude. Also, variability of resonance frequencies and loss factors between installations often exceeds that due to variability in preload and vibration amplitude.*

## 1 INTRODUCTION

Despite many years of research, the dynamic effects of bolted joints on structures are still not well understood. Bolted joints are rarely perfectly contiguous or rigid. Instead, most of the static connection stiffness is concentrated around a frustum in the faying surface region surrounding each bolt. Irregularities of faying surfaces can drastically change the response of bolted systems [1]. The faying surface pressure distributions for the panel described in this paper were measured and documented previously [2]. Various empirical models have been developed to estimate joint stiffness as influenced by frusta cone angles, joint thickness, and contact patch radii [3].

Under dynamic loading, the contact patch total area and shape of bolted joints varies significantly, and dynamic contact force varies by up to 20% of the static contact force [4]. Effective joint stiffness/mass varies with flange thickness, the number and size of bolts, bolt preload/torque (typically set to greater than 50% of the bolt material yield strength), and the motion of the jointed region with respect to a particular mode shape [5]-[9].

Damping behavior is even more complicated and varies with motion type (stick-slip and/or periodic impact) and motion amplitude. Up to 90% of system structural damping can be attributed to bolted joint interaction [10][11]. Most of this damping is thought to be due to dissipation from microslip that depends on oscillating motion at the outer reaches of the contact patch, which is non-stationary [12]-[17]. Numerous models for contact damping have arisen, and initial attempts to align these with more novel experimentally-observed phenomena have become an active area of research [18].

In addition to issues with system variability (from different instances of nominally identical parts), repeatability can vary when testing the same pieces [19]-[21]. Part of this variability is due to the ordering of fastener installation [21]. Also, repeated installations will introduce plastic and other permanent shifts to the microstructures on and just below contacting surfaces, causing an evolution of global dynamic properties. Interface microstructures thus evolve throughout testing [22], but also upon disassembly and reassembly [9], resulting in variations of up to 25% of stiffness and 300% of damping [23].

The benefit of the geometry chosen for this work is that multiple types of surface interaction patterns occur from the various types of dynamic motion. Most of the literature cited above investigates lap-joints, using either the BRB or the S4 beam benchmarks. There is not much literature on the dynamics of flanged bolted connections of plates.

Even when controlling for preload, installation variability of modal parameters in some cases is comparable to the variations mentioned above [24][25][9]. We explore all these issues by measuring the dynamic behavior of a plate structure with a bolted flanged joint.

## 2 TEST STRUCTURE AND METHODOLOGY

Figure 1 shows two aluminum ¼” thick rectangular panels bolted along a common flange. The panels have different lengths to ensure the bolted joint does not lie along node lines (regions of near zero motion) for low order mode shapes. Figure 2 shows a zoomed view of the joint and the experimental configuration. Two flange thicknesses were tested – ½” and ¼”. The bolted joint specifications are typical of marine applications. Threaded holes were drilled into the right panel flange and filled with 3/8-16 18-8 stainless steel (SS) key-locking inserts. The bolts are 3/8-16 PH17-4 SS socket head cap screws. 18-8 1” outer diameter SS washers were placed between the bolt heads and faying surface. Molykote P37 marine grade anti-seize paste used on all threads and contact surfaces.

Aluminum panels, even when bolted, have very low damping loss factors. To minimize the damping added by boundary conditions during testing several suspension methods were investigated. The best methodology was adhering nylon fishing line to the centerlines of the long edges where there is minimal relative motion for bending and twisting modes.

The bolted structure was struck at a corner with an instrumented force hammer and the resulting vibration measured by eight accelerometers. When the system was measured with one or two fewer accelerometers than the eight used for testing, the frequency and damping values were insignificantly different. After attempting several different methods to secure cabling, it was secured to the support structure with a reasonable amount of slack. The cables are not very stiff, and insignificant differences were seen in pre-test damping investigations across all methods to secure the cabling. It is inferred that the added mass from accelerometers and small amount of added damping from the accelerometer cabling does not change the validity of the conclusions addressed later in this document.

In this data set, hammer impacts were used instead of other methods for external forcing, such as broadband or sine sweeps of shaker or acoustic excitation. The advantages of hammer impacts over the other methods include a relatively high amount of energy that can be input into the system, exciting multiple modes at once that can be separately processed with adequate signal processing techniques, and without contacting (and hence contaminating) the system dynamics via physical attachment.

The averaged vibration to force transfer function in Figure 3 shows sharp response peaks at the first two bending and twisting modes of resonance. Figure 4 compares the relative motion between the flange surfaces for these mode types. Bending motion induces periodic 'slapping' of the contact surfaces whereas twisting induces a stick-slip due to oscillating shearing. Both mechanisms induce damping in different ways, and sometimes non-linearity for strong vibration.

To evaluate the linearity of the joint behavior, time-dependent resonance frequencies and damping were measured. For each hammer strike the time signal was filtered around each resonance frequency. Zero crossings in the time signals were used to calculate instantaneous resonance frequencies and the logarithmic decrement of the vibration amplitude was continuously estimated. Time varying loss factor vs frequency curves were calculated for each accelerometer and averaged after Refs. [26]-[28]. These curves are later plotted for different joint and test conditions. While Nyquist plots are often used to show nonlinearity in damping, the method presented in Section 3 matches the plotting style common in the joint dynamics community.

The test conditions were:

- three bolt torques (10%, 33%, 50% of bolt Yield Strength),
- three impact forces (15, 55, and 100 N),
- three repeated impacts for each force level,
- three repeated setup assemblies, and
- two flange thicknesses.

The bolt pre-loads are lower than typical (which is about 2/3 yield strength) to induce more relative motion and damping.

Excitation force amplitude variability for these data sets is very low, as shown in Figure 5. Impacts were controlled via an automated hammer, which has configurable parameters for start and stop motion and step velocity. The three resulting levels of amplitude are well-separated, and highly repeatable across the various test cases.

A separate test was performed to examine the relationship between preload and torque, carried out on flat metal plates of a similar thread configuration. Data from this test are shown in Figure 6. The fasteners were the same 17-4 PH 3/8"-16 screws as those used in the flanged plate dynamic testing, with the bottom plate having the same coil-insert threads and the through-plate having a clearance hole. The material was the same AL T6 6061-T651, and identical lubrication material was also applied in this test. Two through-plate thicknesses (1/4" and 3/16") and two roughness finishes of the AL surface were used at the interface.

An approximation relating preload to torque is given as

$$T = F * D * N,$$

where the product of preload force ($F$), fastener diameter ($D$), and a nut factor ($N$) is equal to the input torque. The slope of the trendline provides the nut factor $N$, which in this case turns out to be $N$=0.135, which is consistent with literature for these types of configurations.

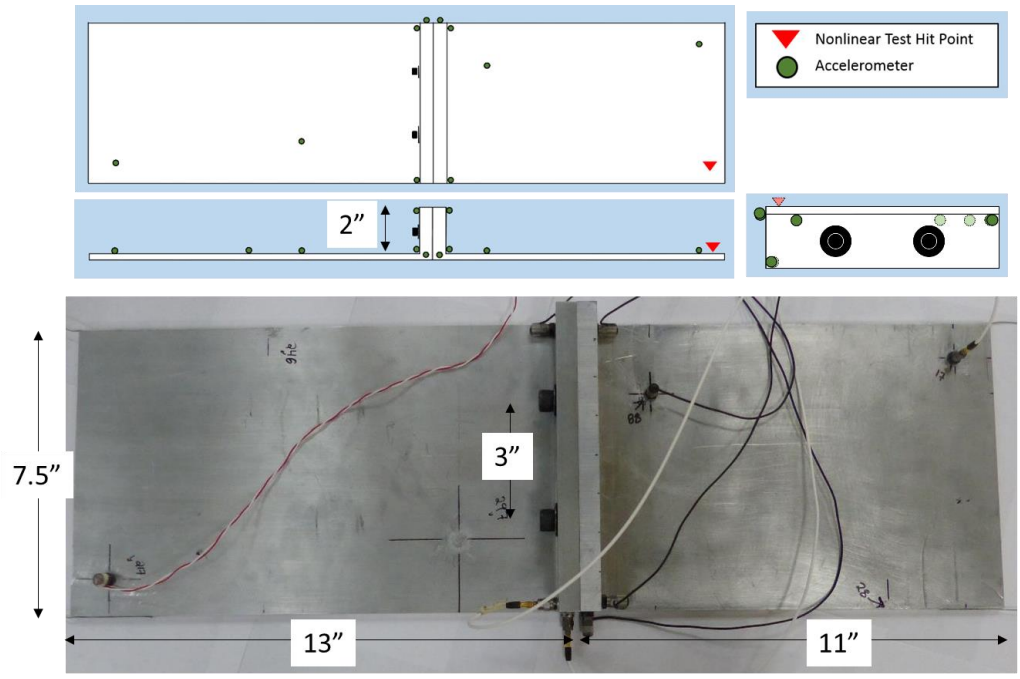


Figure 1. Bolted plate structure with instrumentation.

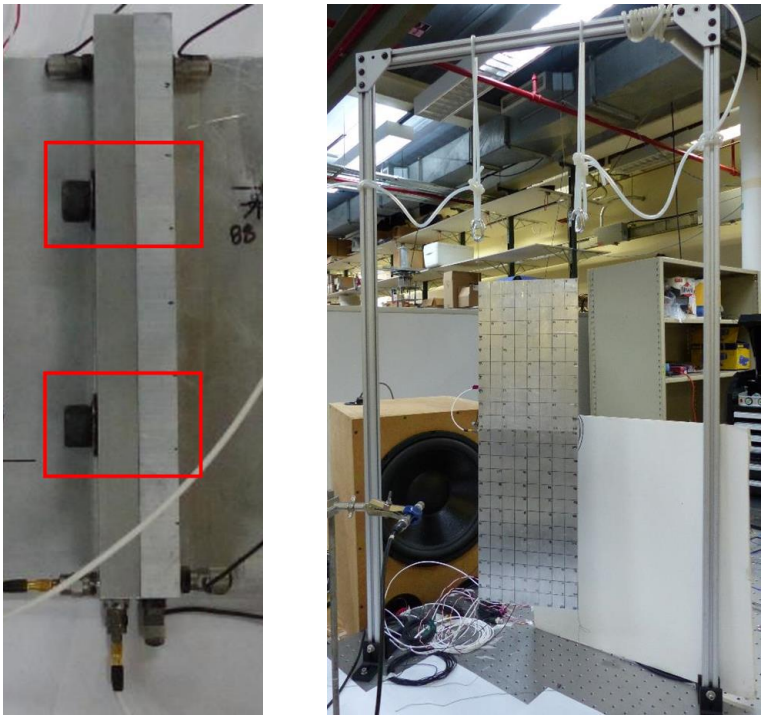

Figure 2. Zoomed view of bolted joint (left); experiment setup with panel suspended by nylon wire adhered to outer edges (right).

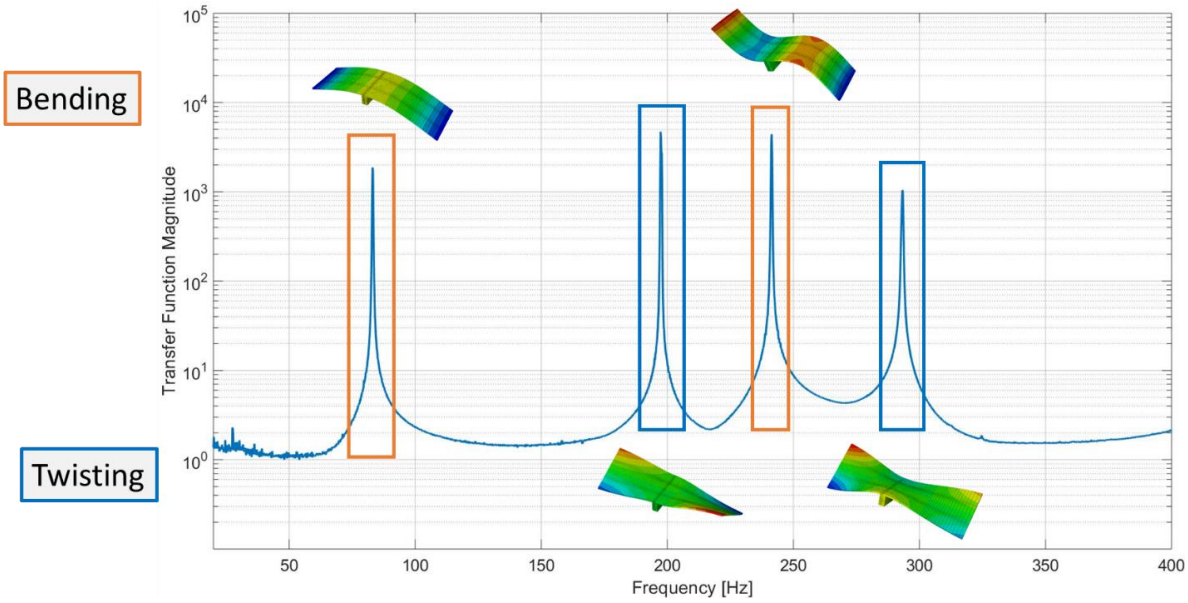


Figure 3. Vibration of bolted structure and first two bending and testing modes.

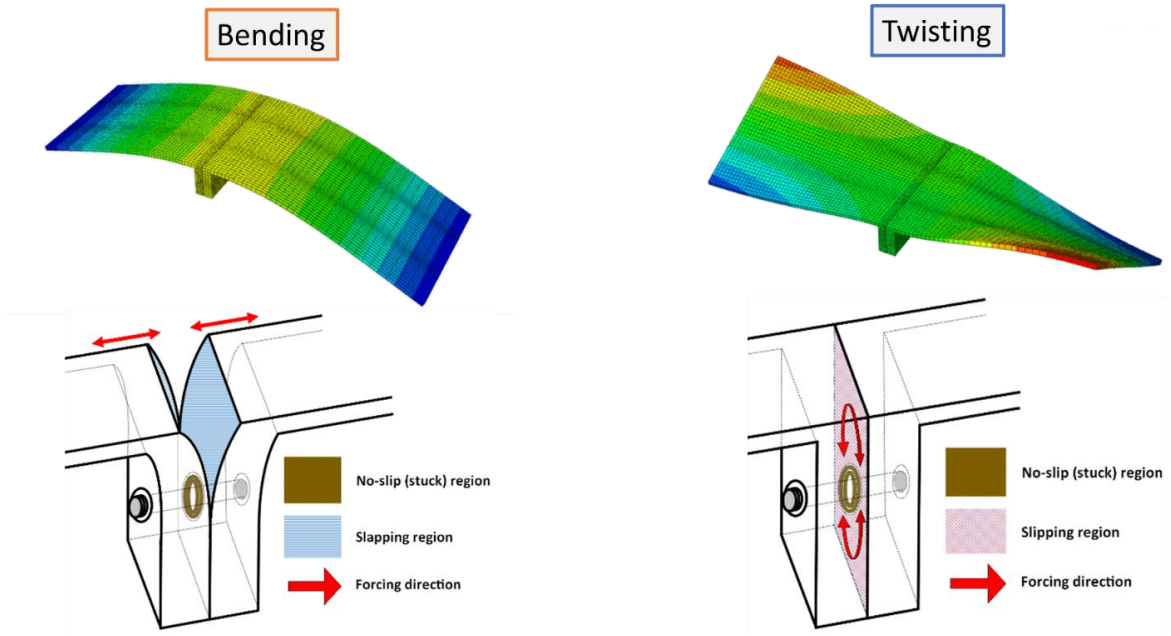


Figure 4. Joint motion for bending and twisting modes.

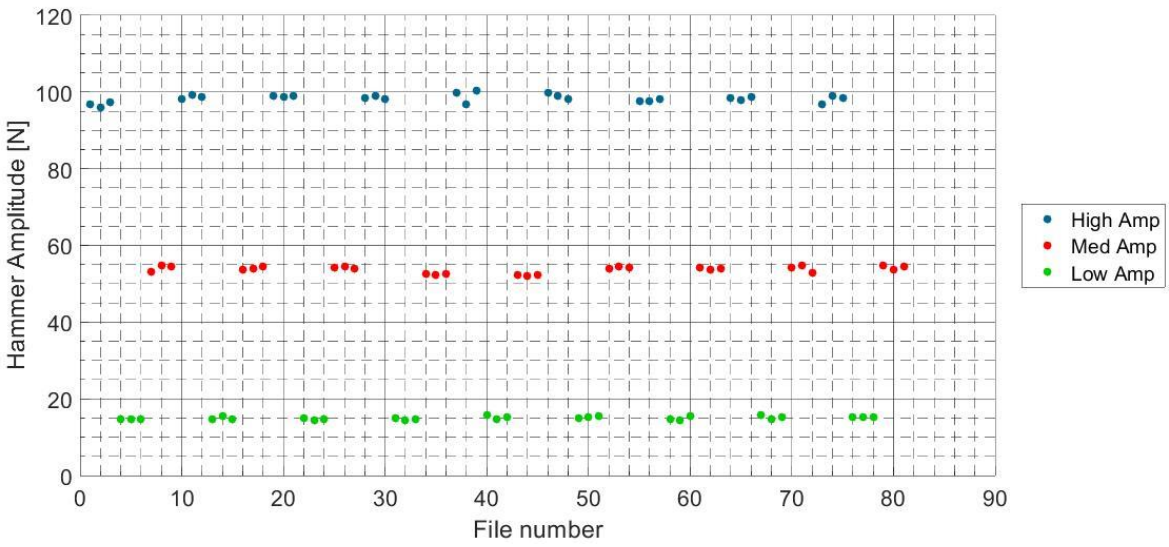


Figure 5. Example output of automated hammer amplitude across a sample range of impact tests.

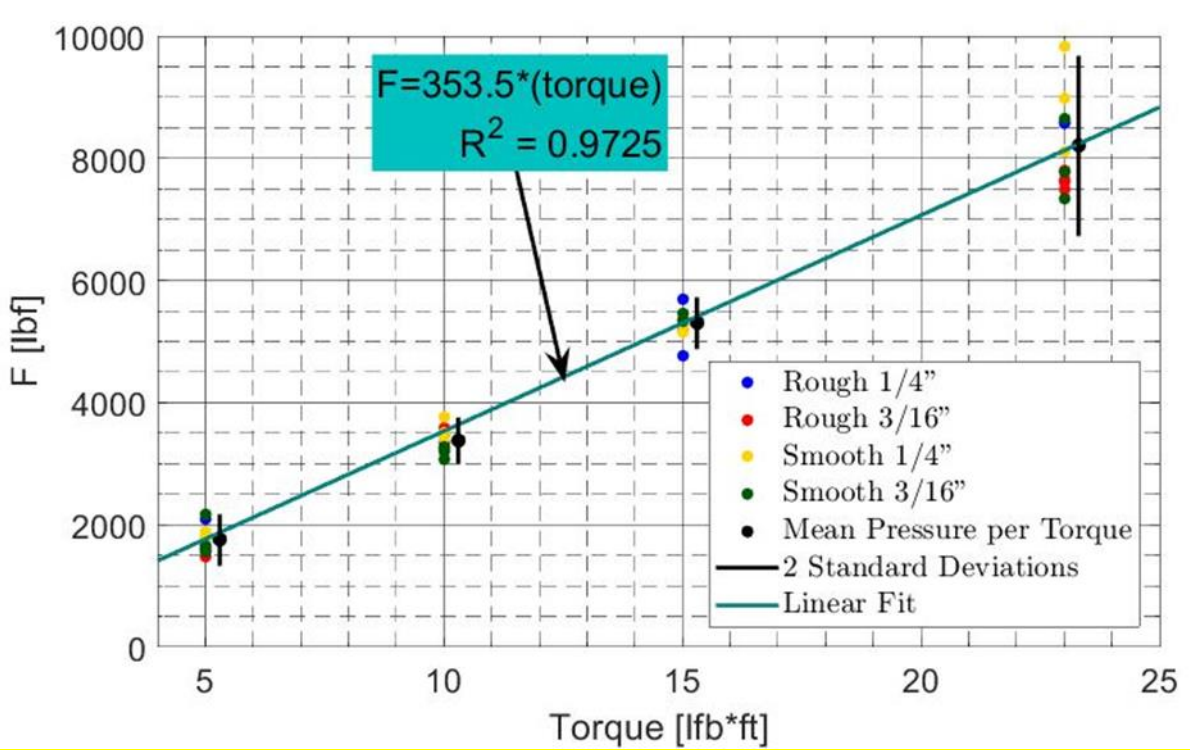


Figure 6. The relationship between target input torque and resulting preload force, showing a linear fit. Standard deviation increases above about 20 lbf*ft, but mean values show an approximately linear relationship between torque and preload with relatively low variability.

## 3 TEST RESULTS

### 3.1 First Bending Mode

The fundamental bending mode can induce repeated separation and 'slapping' at the top and bottom of the joint. The flanges add both mass and stiffness to the shape. Figure 7 compares the time-varying resonance frequencies and damping for the different test configurations and conditions. Unbolted panel damping of 0.001 is shown for context. The ½" flange data are shown in shades of blue and the ¼" data shown in shades of red; we use this color convention for all subsequent modes.

A single line corresponds to a complete ringdown test after excitation. Individual tests happen to be grouped on the plot according to assembly. Figure 8 shows a subset of these data for the case of ¼" thick geometry and low torque. The added damping by the joint varies from 0.004 to 0.011. Each damping/frequency curve shows minimal nonlinearity, except for the lightly preloaded joint which shows slightly shifting resonance frequencies but mostly stable damping. Resonance frequencies vary by about +/-1.5% for the thick flange and +/- 3.5% for the thin flange, with the low preload frequencies generally lower. There is no clear trend in frequency or loss factor with respect to flange thickness. However, the thinner ¼" flange shows the most variability with preload. The most interesting feature of these data is the significant +/- 2.5% resonance frequency and 2x damping variability of the lightly torqued thin flange data over repeated assemblies; this exceeds any other variability.

### 3.2 First Twisting Mode

Figure 9 compares the resonance frequencies and damping of the first twisting mode. The resonance frequency variability is +/-2% for the thin flange, and only +/- 1% for the thick flange. Thickening the flange adds more stiffness than inertia, increasing resonance frequencies. As expected, increasing bolt torque/pre-load increases stiffness and resonance frequencies. The added damping for the first twisting mode is much lower than that of the first bending mode – only 0.002 to 0.004, which is only slightly above that of the unbolted structures. There is therefore little slip-stick behavior for this structure and the configurations tested. However, the damping variability over time is higher than that of the first bending mode as shown by the stronger slopes in the individual damping vs. resonance frequency curves.

The subset of cases with ¼" thickness geometry and medium torque are plotted alone in Figure 10. While not true in all tested cases, it was generally found that frequency (stiffness) decreases and damping increases with instantaneous amplitude. This amplitude dependence is plotted for the same subset of data in Figure 11. This observation is consistent with other experiments in the literature [29]. The variability across repeated assemblies for the low pre-load cases is not as pronounced as in the first bending mode, suggesting that the twisting/shearing mechanisms is not as sensitive as the 'slapping' behavior of bending motion.

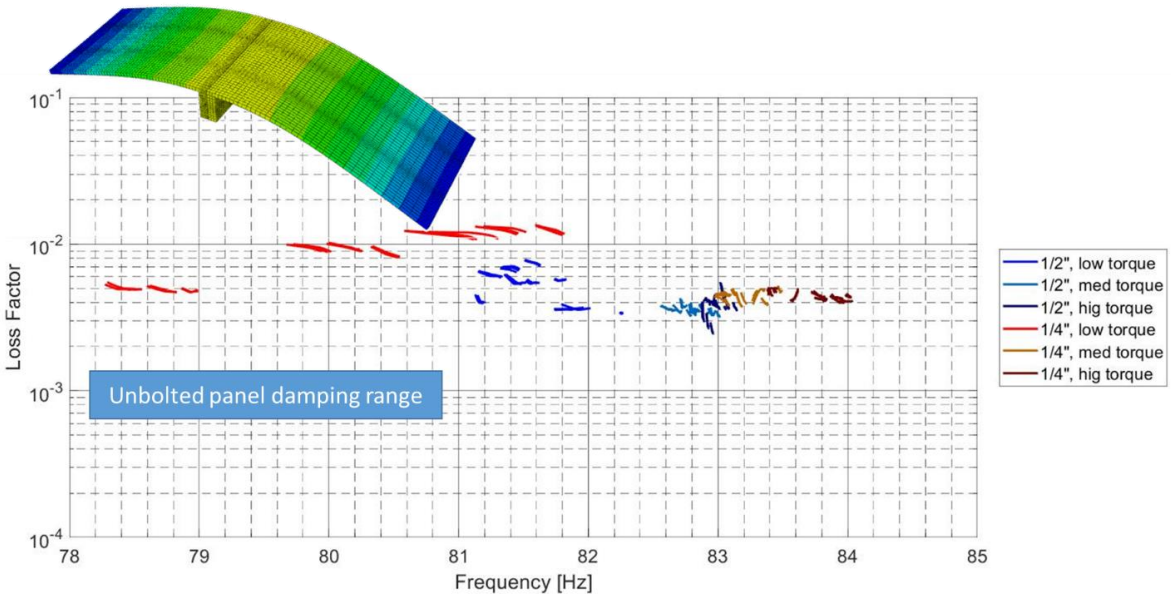


Figure 7. Instantaneous damping as a function of instantaneous frequency for first bending mode.

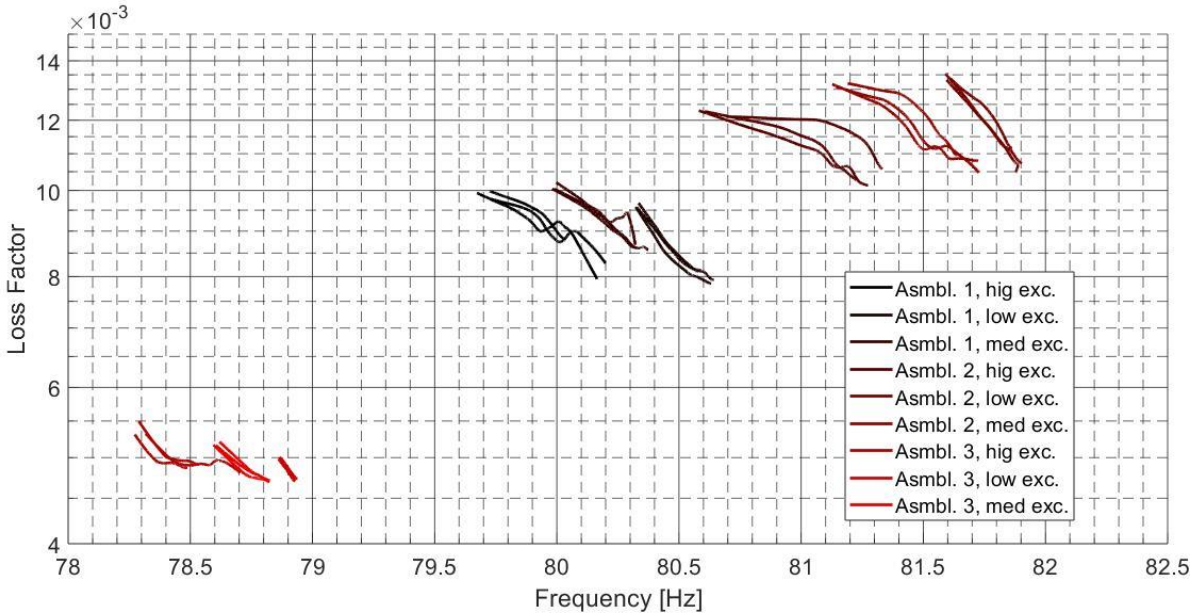


Figure 8. Instantaneous damping as a function of instantaneous frequency for first bending mode; ¼” geometry and low-torque cases only.

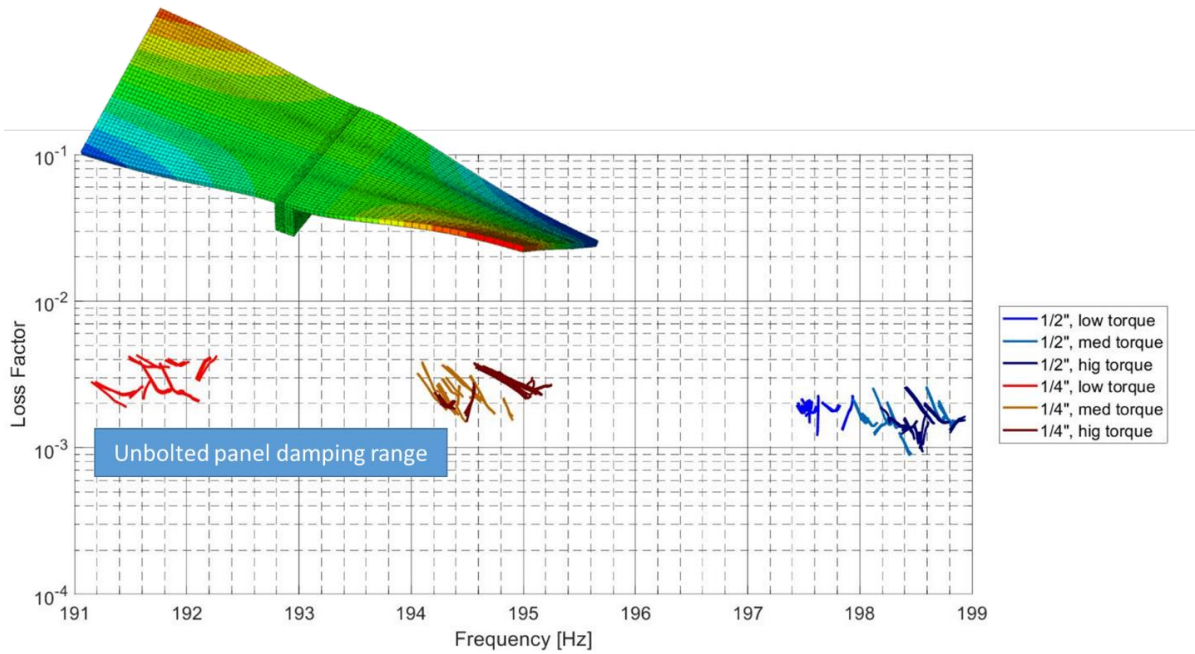


Figure 9. Instantaneous damping as a function of instantaneous frequency for first twisting mode.

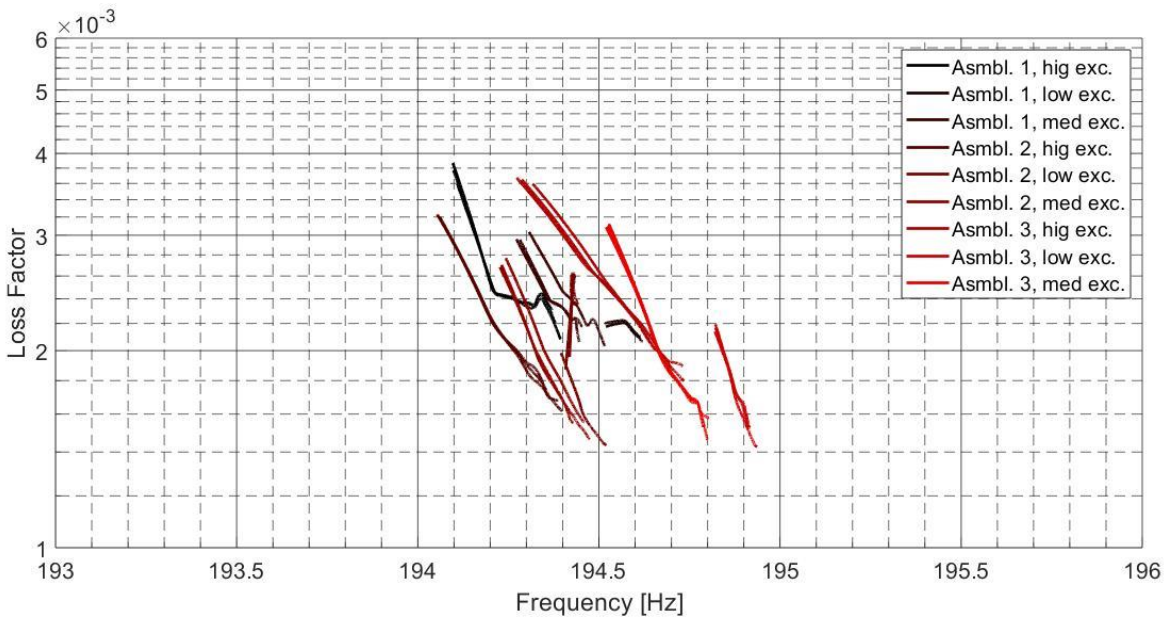


Figure 10. Instantaneous damping as a function of instantaneous frequency for first twisting mode; ¼" geometry and medium-torque cases only.

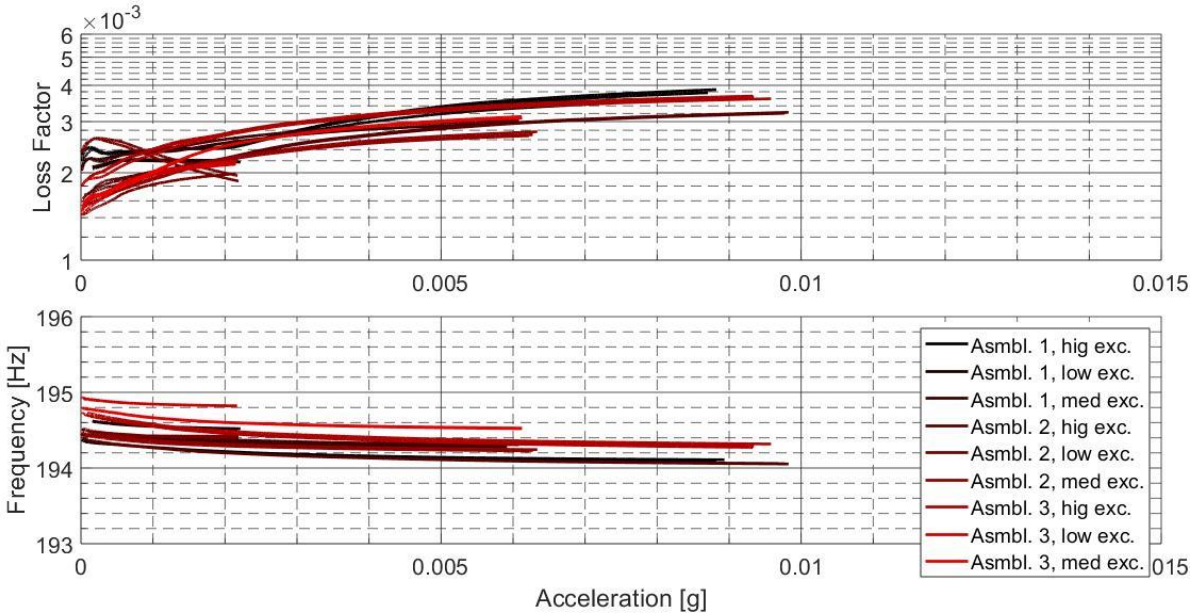


Figure 11. Instantaneous frequency and damping as functions of vibration amplitude for first twisting mode; ¼" geometry and medium-torque cases only.

### 3.3 Second Bending Mode

Figure 12 compares the resonance frequencies and damping loss factors for the second bending mode. The variability is very low, as is the added damping which is almost negligible. The reason for the low variability is the minimal relative motion in the joint for this mode; just a rotational inertia as the joint rotates near the node line of the shape. Since the motion is purely inertial, the resonance frequencies are lower for the thicker flange. While seemingly uninteresting, this behavior points out an important aspect of bolted joint dynamics: *joint stiffness/mass and damping vary significantly with mode shape and depend on relative motion at the joint. Therefore, it is not appropriate to apply general rules of thumb for joint stiffness/mass or damping to structures. These behaviors will always be mode-dependent*.

### 3.4 Second Twisting Mode

Figure 13 compares the resonance frequencies and damping for the second twisting mode. The flange adds inertia to the structure, but unlike the second bending mode there is modest relative motion along the joint, which leads to higher added damping and more variability. The added damping ranges from 0.003 to 0.009, with the highest added damping for the thin flange with low preload (consistent with the behavior of the first bending mode). The resonance frequency variability is about +/-2% for both flange thicknesses with higher frequencies with increasing torque. As with the first bending mode, the variability across installations for the thin flange with low torque exceeds that of the other conditions. There is also more resonance frequency and damping nonlinearity evident for this case.

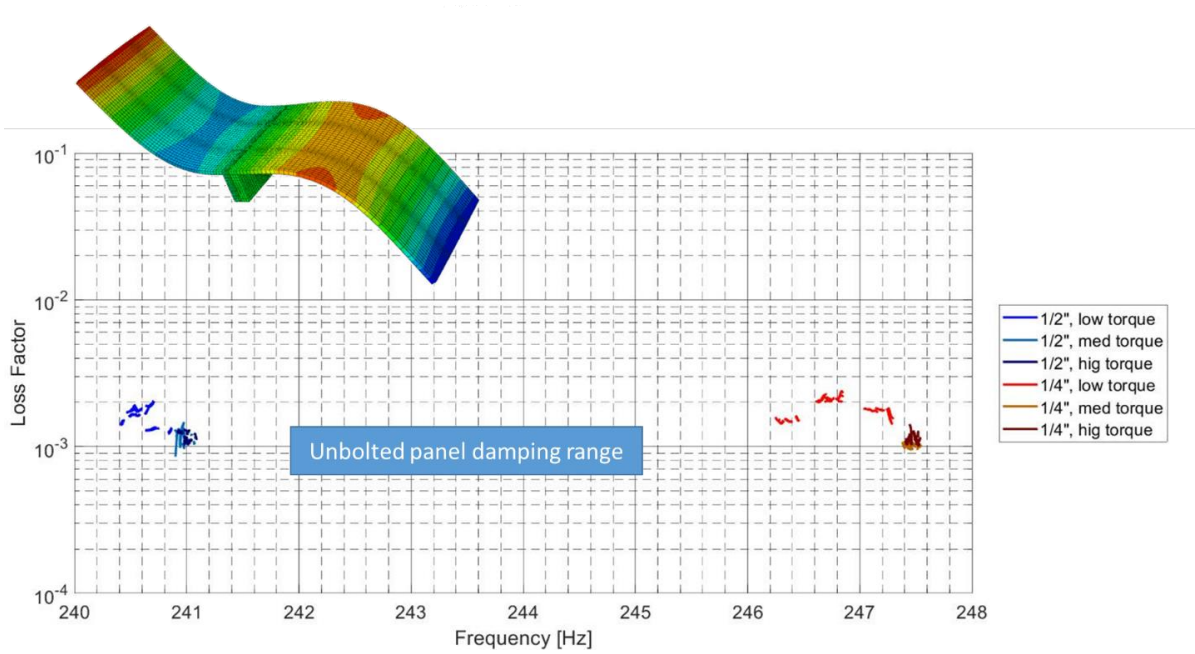


Figure 12. Instantaneous damping as a function of instantaneous frequency for second bending mode.

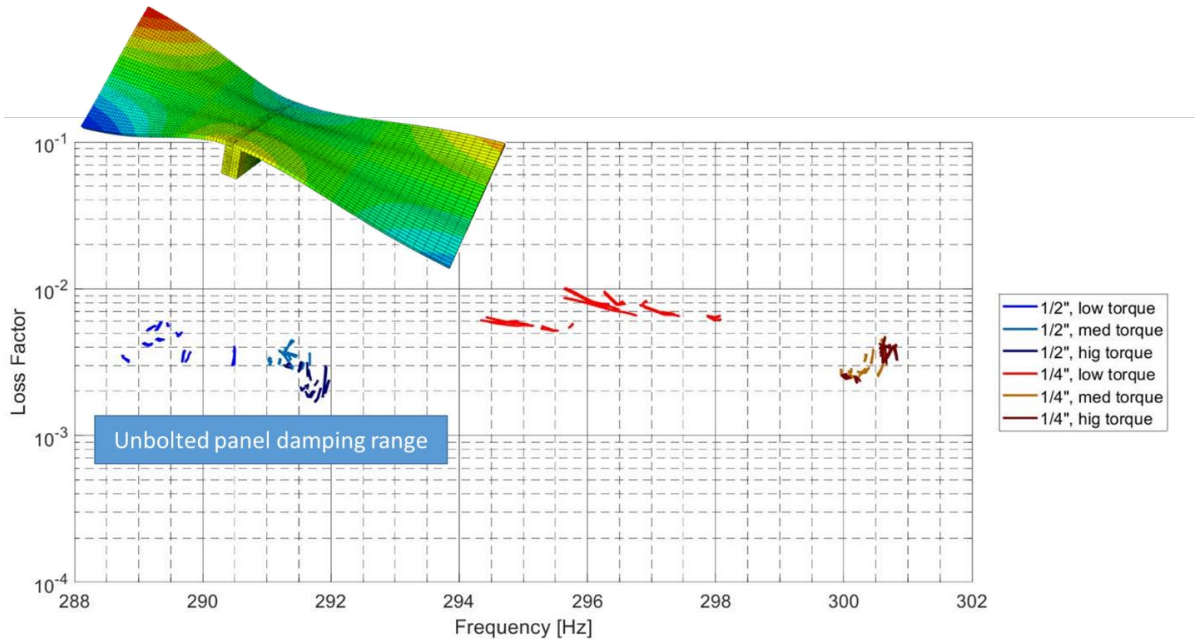


Figure 13. Instantaneous damping as a function of instantaneous frequency for second twisting mode.

## 4 SUMMARY AND CONCLUSIONS

The dynamic behavior of two rectangular Aluminum panels bolted together along flanges was measured over a wide range of conditions, including two different flange thicknesses, three bolt preload strengths, and three separate installations. Time-varying resonance frequencies and damping loss factors were compared for the four lowest order modes of vibration, including two bending modes and two twisting modes.

The general response of the structure depends on relative motion of the mode shape. *Since each mode is unique and the pattern of motion for each includes a unique sum of bending and twisting motion, there are no global rules of thumb for behavior of the modal mass or stiffness of low order modes*. Each of the two types of motion introduces a different damping mechanism. The conditions most ripe for nonlinearity are low torque and thin geometry. In these cases, the nonlinearity is demonstrated in the highest degree in the frequency and damping curves. This variability is manifest in two ways: with both a larger range of frequency and damping per test, but also the spread of the data across tests in general being much wider. Based on the groupings shown, this extreme variability is due to assembly differences.

There is not significant variability in damping values, except for that which arises due to differing assemblies. The modal behavior is mostly linear, except for modes with the thin flange geometry at low torque conditions. Below a certain threshold preload, reducing torque results in reduced resonance frequencies. While a number of hypotheses could be offered by way of an explanation for these results, additional testing will be necessary to further investigate the specific details and nature of this threshold. *The most significant finding is that variability between installations can exceed that due to controlled preload and vibration amplitude variations*.

Modeling joint dynamics remains a challenge. While some significant progress has been made over the past decade, the ability to obtain reasonable results a-priori without using expensive nonlinear models is still the key goal. Initial modeling attempts using plates and beams and solids were only moderately successful. After repeated fine tuning and mesh convergence studies reasonable estimates of resonance frequencies are possible [8]. However, it is extremely challenging to simulate added damping. Further studies, if successful, will be the subject of a future paper.

Finally, there is a growing community of researchers investigating bolted joint dynamics. Among others, notable examples include the Tribomechadyanics group at Rice University (http://tmd.rice.edu/), the structural dynamics groups of BYU/UW-Madison (https://byusdrg.com/), and the International Committee on Joint Mechanics (https://jointmechanics.org).

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