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Bolt Preload Loss due to Modal Excitation of a C-Beam Structure

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Abstract
Bolted joints often risk failure due to the loss of fastener preload when subjected to dynamic, multiaxial loads. This process is a complex problem that depends on multiple attributes such as loading direction, rate, contact within the threads and the interface, material properties, and many others. Current literature suggests that oscillatory shearing loads appear to be most detrimental to the loss of preload in threaded fasteners. To investigate the effect of less idealized loading conditions, an experimental setup employing a bolted c-beam structure is used to study loss of preload from various initial preloads during harmonic excitation near specific resonant frequencies of the structure. The preload force is measured using bolts equipped with internal strain gauges and the structure is excited at specific modes via sine dwell excitation with an electrodynamic shaker. The experiments were designed to measure loss of preload as a function of excitation duration and strength. A finite element model incorporating a fully-threaded joint is developed in parallel to investigate the effectiveness of each at measuring and predicting bolt loosening.

Keywords: bolt loosening; preload loss; modal analysis; contact mechanics; shaker excitation

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1. Introduction

Bolted joints are commonly used in many applications to join subcomponents in a mechanical assembly. This fastening technology is easy to assemble and replace parts, and, when used correctly, can properly carry loads across interfaces without permanently altering the clamped structures. During their lifetime, a typical joint experiences different types of loads over many cycles depending on the operating environment. One type of common loading condition to bolted joint assemblies is oscillatory dynamic forces due to vibrations in the main structure. Such dynamic excitation may instigate preload loss by several possible mechanisms [18] such as material wear in the interface, plastic deformation in the fasteners or structures, or in extreme cases, complete failure of the fastener under large strains [19]. The preload loss of interest to this research is the case where loosening in the fastener occurs via backing off of the nut on the threaded bolt, resulting in a gradual or sometimes sudden loss of preload that clamps the two or more members together. This failure mechanism occurs without permanent damage to the structure.

The preload loss due to nut loosening has often been characterized to occur in two sequential stages [7]. The first stage, termed Stage I loosening, corresponds to loosening caused by local plastic deformation and wear in the load bearing threads, which carry the majority of the load during small strains. In this stage, the plastic deformation of threads enlarges the existing gaps between them, reducing the stiffness and load in the joint. This allows slip to occur at the head of the fastener and the contacting threads, further reducing the preload due to the nut backing off the threads. Junker first suggested that this preload loss occurred due to gross slip at these interfaces [1], but there is no rotation between the bolt and the nut during this stage. Jiang *et al.* discovered that preload loss in this stage follows a sudden drop in force, caused by cyclic plastic deformation in the thread contacts [2]. Further studies by Liu *et al* found that fretting wear in the threads causes the gentle decay in preload following the initial drop [3]. Recent work by Zhang further investigated the wear on the first engaged thread, showing that a combination of abrasive ploughing (from preloading) and spalling wear (from transverse cyclic loading) had occurred [4].

Following Stage I, Stage II loosening is defined as gross rotation between the nut and the fastener. Zadoks found that in a transversely loaded joint, slip at both the threads and the bolt head were necessary to cause rotational nut loosening [5]. Later, it was found by Pai and Hess that only partial slip at these interfaces was necessary to induce rotation [6]. Translational motions were found to be the most severe cause of Stage II self-loosening. Zhang used a finite element model of a bolted joint under transverse loading to show that partial slip occurs due to a cyclic bending moment, causing stick-slip between the mating threads [7].

Junker had previously shown that the loss of preload is most likely to occur as the result of shear loading on the joint [1]. This led to other studies which were interested in the loosening of bolted joints as a result of idealized loading conditions, such as pure transverse or axial loading studied in [7,8]. In many real-world applications, bolted joints experience more complicated loading schemes due to a combination of these idealized loading cases. Moreover, since bolted joints exhibit strong nonlinearity, linear models that obey superposition cannot be used to combine the results observed from these idealized loading cases. For example, vibration, especially at frequencies near a resonance, can cause a detrimental loss of preload in a structure [14].

This research presents a study of self-loosening in bolted joints of a c-beam structure undergoing harmonic, steady-state excitation near different resonant frequencies of the assembly. The work shown here was conducted over the course of seven weeks through Sandia National Laboratories' Nonlinear Mechanics and Dynamics (NOMAD) summer research institute. The objective of the project was to investigate loosening in preloaded bolts both experimentally and numerically with computational mechanics finite element models. An experiment was designed to use electrodynamic shakers to harmonically excite the beam assembly near resonance, and the axial preload force in the joint is measured using bolts equipped with strain gages. Analogous to the experiments, a detailed finite element model of the beam assembly was created, including the small length-scale details of the threads between the bolt and the nut. There has been little research presented in the literature exploring the connection between modal excitation and the occurrence of bolt loosening on generic structures. A 1987 NASA report [9] made use of a cantilevered and bolted assembly of two plates as a means of achieving shear loading with sine dwell, sine sweep and random excitation. Imagine a hole drilled through the two plates. As bending modes are excited the holes become unaligned due to a relative difference in bending radii and a shear forces befall an installed bolt. The work concluded that displacement amplitudes at resonance were the largest predictors of fastener loosening, however there was little emphasis on initial fastener preload as the report did not document this value. With so much intervening research implicating preload as the foremost deterrent of bolt loosening, expanding upon this work is warranted [8].

The paper is organized as follows: Section 2 presents the experimental setup with the electrodynamic shaker and strain gage equipped bolts. Section 3 presents and discusses some of the experimental results obtained on the hardware, as well as a discussion on both experimental challenges and implications of the results. Section 4 provides an overview of computational methods followed by the presentation of computational findings in Section 5. General conclusions and suggestions for future work are offered in Section 6.

2. Experimental Setup

The hardware used to explore the self-loosening phenomena under harmonic excitation near resonance is the so-called c-beam assembly comprised of two identical c-beams bolted together; see Figure 1 for the mechanical drawing with dimensions. Each c-beam was made from 4340 steel alloy and was labeled either B9A or B9B to differentiate their position in the setup. Based on previous research [20], the individual beams were tested to measure the mode shapes and natural frequencies with free-free boundary conditions. Based on these previous results, the material properties for B9A and B9B were calibrated to match the first six natural frequencies of the individual piece-parts. The c-beam assembly has two well defined bolted interface on each end with raised pads to isolate the jointed interface in the structure.

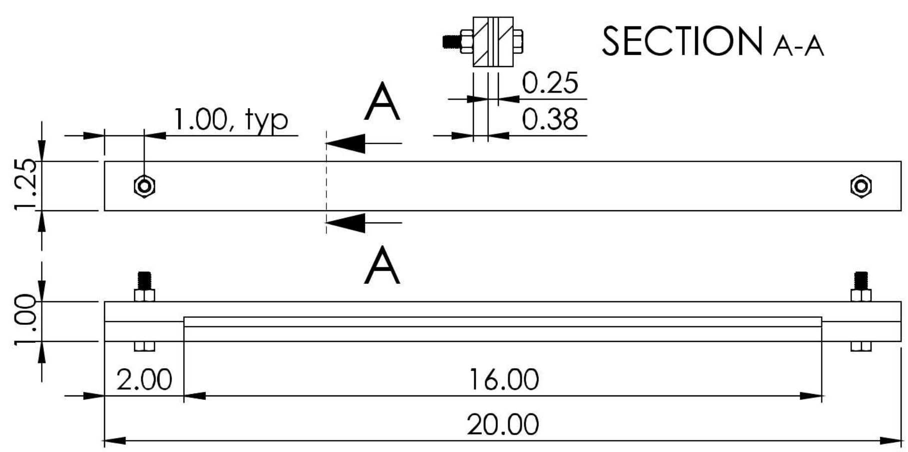


Figure 1: Assembly of bolted, symmetric c-beams (dimensions in inches).

A photograph of the experimental setup is shown in Figure 2. Both the bolts and nuts used join the individual beams are SAE Grade 9 zinc-plated steel. The bolts have internal strain gauges installed within the shank, produced by STRAINSERT [11], allowing for real-time preload measurements. The schematic in Figure 3 shows the operation of the force measuring bolt. The c-beam structure was preloaded to an initial value, measured via the strain gage, and then freely suspended from a semi-rigid test frame using soft bungee cords. Four triaxial accelerometers (Endevco Corp "Isotron" model #65-10) were placed on the c-beam structure in various locations to measure the vibration response during excitation. A drive-point accelerometer was placed at the location opposite of the input force gage to measure the induced acceleration at the drive-point. A modal shaker (model number: MT-161) and an amplifier (model number: PA-138-1) were controlled using a Siemens LMS SCADAS 24-bit data acquisition system to excite the c-beam structure [15]. A PCB Piezotronics force transducer was bonded to the c-beam assembly in line with a stinger coupling the shaker and structure [17]. The shaker is driven at frequencies corresponding with the first two modes of the c-beam structure. The in-phase and out-of-phase first bending modes studied in this research are shown in Table 1. Bolt preload forces (provided via the strain bolt calibration) were applied equally on both bolted connections at loads between 70 – 1000 lbf (311 – 445 N). Simcenter hardware and software (courtesy of Siemens LLC) were used to collect input force, accelerometer, and strain bolt readings [15]. The raw data was smoothed by averaging samples collected over one period corresponding with the modal frequency.

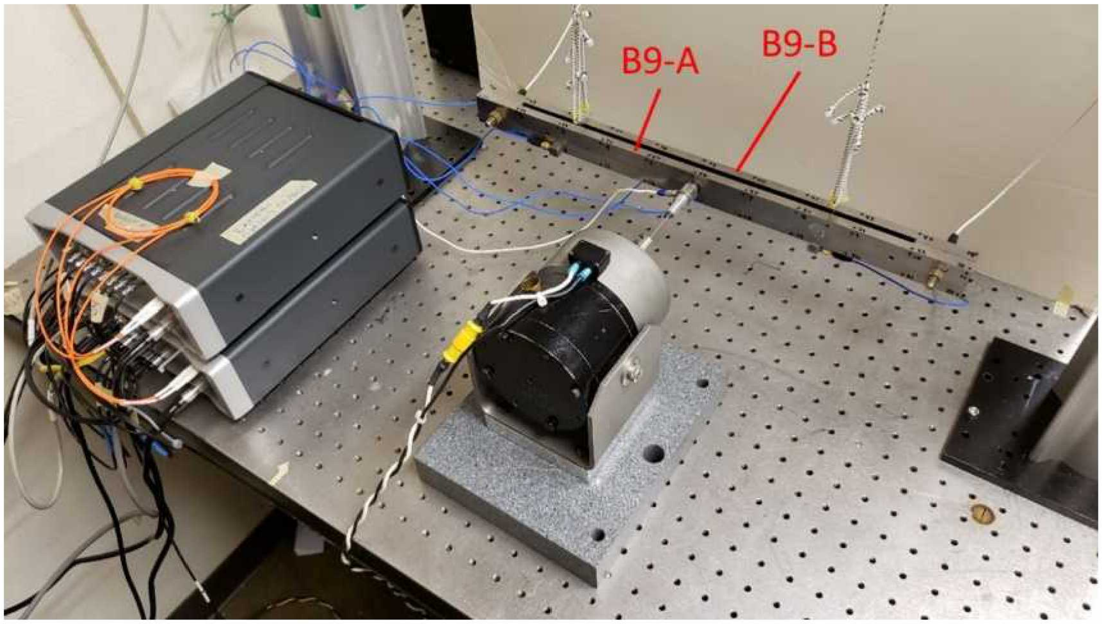


Figure 2: The experimental setup consisted of a shaker connected to an amplifier (not shown) which was fed by the Siemens data acquisition system. The stinger from the shaker held the force transducer on the end which was chemically adhered to the c-beam structure.

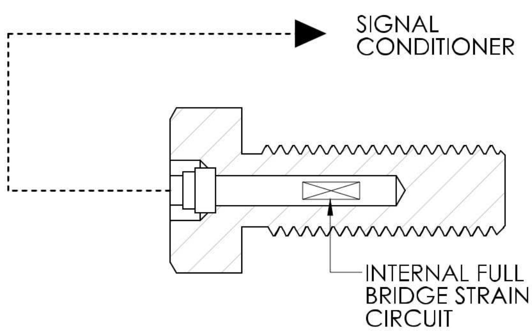
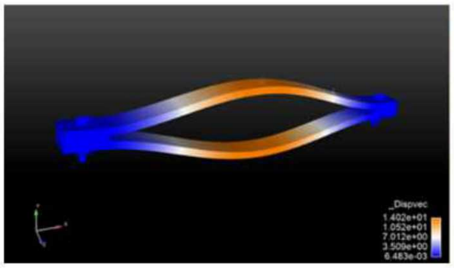


Figure 3: Schematic of the strain bolts used to determine preload [11]. The internal strain gauge calculates the axial strain in the bolt which can then be used to determine the axial force, or preload, observed in the bolt.

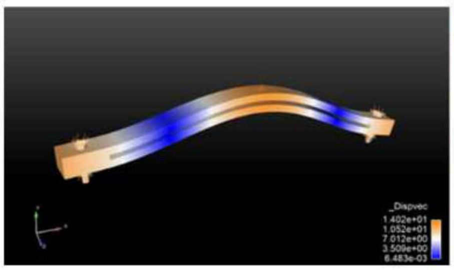
3. Experimental Results and Discussion

An experimental modal analysis was performed to identify the natural frequencies of the bolted assembly prior to performing the shaker-excited loosening experiments. A roving impact hammer test was performed on the preloaded structure at 66 various points on the beams to spatially resolve the mode shapes. The roving hammer impact force was kept to a low level between 8 - 15 lbf (36 – 67 N) to avoid contaminating the measured forced response functions (FRFs) with nonlinearity from the joint. The results for the first two elastic modes are shown in Table 1. The data suggests that as the preload forces increase, the joints stiffen and the modal frequencies shift to higher values until plateauing to a constant value. These two modes were targeted for the bolt loosening experiments since the mode shapes loaded the joints in different ways. The first out-of-phase bending mode applies a bending moment on the bolt head and nut during its deflection, while the first in-phase bending mode applies a shear load in the joint.

Table 1: Modal frequencies as a function of bolt preload.



Mode 1	
Preload [lb_f]	Frequency [Hz]
70	275
160	281
500	287
700	288
1000	288

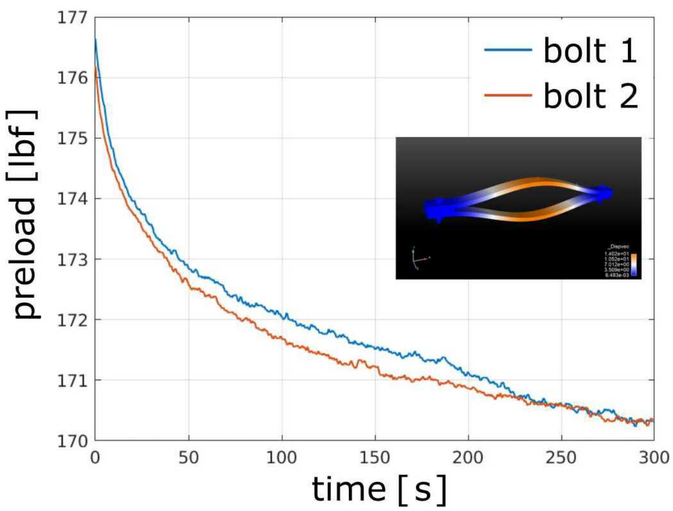


Mode 2	
Preload [lb_f]	Frequency [Hz]
70	325
160	337
500	350
700	350
1000	350

After characterizing the vibration modes with the roving hammer impact tests, several sine dwell experiments were performed on the c-beam assembly up to a maximum of 10 lbf (44.5 N) input force, 30 minutes in duration, and at the preloads and modes listed in Table 1. These experiments were performed to scope the test parameters required to observe loosening in the test setup. A majority of these experiments revealed preload losses of less than 5% of initial preload following five minutes of excitation. It was discovered that only mode 1 produced complete loosening (visual rotation of the nut off the beam) if the preload level were low enough, e.g. finger tightened. Consider the preload values listed in Table 1 in relation to the maximum load rating of the strain bolts of 5,000 lbf (22.2 kN). Fernando discusses, experience guided, installation preload values of at least 65% of bolt yield load for fasteners in normal vibratory environments [8]. The scoping tests revealed that the available 25 lbf (111 N) modal shaker to be insufficient for eliciting bolt loosening at more than 1% of bolt yield.

After performing the scoping tests, the structure was harmonically excited at the frequency of the in-phase bending mode for five minutes with the shaker attached at the center of the beam, as shown in Figure 2. The root mean square (RMS) of the input force supplied to the structure was 4.2 lbf (19 N). Figure 4 shows a plot of the bolt preload forces in each as a function of time for the five-minute experiment; each bolt was torqued to approximately 5 in-lbf (0.6 N.m) resulting in a nearly 176 lbf (782 N) preload force. It should be reiterated that these preload levels are small (i.e. loosely tightened) since initial scoping experiments revealed that higher levels of preload did not produce any bolt loosening measurements. The raw time histories of the bolt strain gages were oscillatory, so a moving average was used to smooth the raw data, such that the lines in Figure 4 represent the average of the bolt preload over time. This method averages a complete period of raw data centered on a sample, then was slid over by one sample, and repeated. The preload measurements reveal that over a five-minute period, the preload drops exponentially and then begins to decay in a linear fashion. Overall, the bolt preload dropped by approximately 5 to 6 lbf (22 to 27 N), or about 3% of the initial preload. Previous research in the literature found that the loss of preload begins with an exponential decay and moves into a period of linear decline before reaching Stage II loosening [6]. The trend observed in Figure 4 resembles the Stage I trend observed by Zhang [4].

In an attempt to observe Stage II self-loosening in the joints, longer duration tests were conducted using mode 1 excitation at 5 lbf RMS (22 N). Figure 5 shows the results of both fifteen-minute and thirty-minute tests. These preloads were normalized by their maximum value (170 lbf (756 N) nominal) to better show the trend observed in each test. The preload shows a similar trend as the shorter test as the preload initially drops exponentially then decays



linearly. The difference in severity of the decreasing trends in Figure 5 requires some explanation. With only small losses in preload observed for the shorter run (the preload was still in the desired nominal range) the longer duration test followed without adjusting the preload of the bolts. The bulk of Stage I loosening had occurred during the first 5 minutes of the test. Essentially the longer duration test picked up where the shorter test left off; the linear tail of the Stage I loosening trend. This observation agrees with the division of bolt loosening into two primary stages. During Stage I, plastic deformation of the threads results in a sudden loss of preload until plateauing as the imperfectly matched threads "settle in" to place. The plateauing is the strongest indicator that Stage I loosening was observed. The bulk of collected data follows the Stage I loosening trend observed by Pai *et al* [6], however, further experimental measurements are needed to validate this observation.

The experimental setup observed small losses in preload over long periods, with lightly torqued fasteners. This suggests that a larger shaker was required to achieve Stage II loosening at realistic preloads since these experiments were near the limit of the shaker's output force rating. A brief derivation is presented here to explain some of the challenges and limitations with resonance testing with an electrodynamic shaker. Consider the impedance equation

$$\underline{Z}_{mech} = \frac{\underline{F}}{\underline{v}} = i\omega m + c + \frac{k}{i\omega}. \tag{3.1}$$

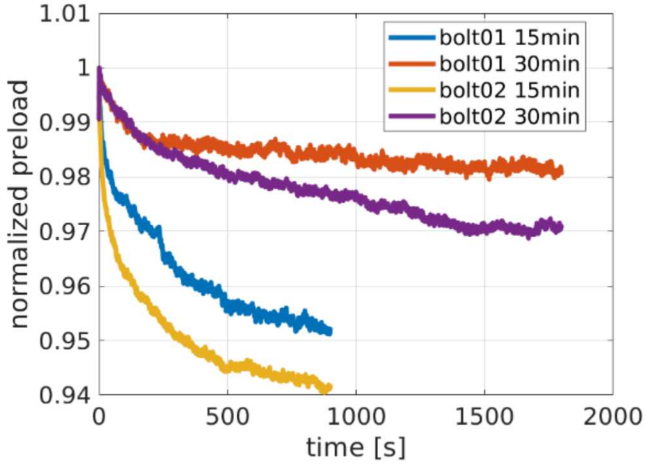


Figure 5: Normalized preload results from longer duration tests.

Here the complex mechanical impedance, $\underline{Z}_{mech}$, of a damped and harmonically driven parallel mechanical oscillator is expressed as the ratio of the driving force, $\underline{F}$, and the resulting velocity, $\underline{v}$. The angular frequency is represented by ω, the mass by m, the damping coefficient by c, the spring constant by k, and the square root of negative one by i. At resonance the angular frequency is

$$\omega_r = \sqrt{\frac{k}{m}}. \tag{3.2}$$

Substitution of $\omega = \omega_r$ into Eq. 3.1 results in the cancellation of the mass and stiffness terms and minimization of the impedance,

$$\underline{Z}_{mech,res} = c\,. \tag{3.3}$$

The power supplied to the structure at resonance is purely resistive, serving only to overcome mechanical damping losses. To maintain the input force at a constant value here requires relatively large shaker output velocities. Large test article input velocities combined with a low mechanical impedance translates to increased power consumption. The amplifier attempts to supply the power by increasing the voltage in current regulation mode or the current in voltage regulation mode. The electromechanical coupling within a modal shaker includes Lenz's law linking velocity with voltage, while Ampère's law couples force and current. Elevated shaker output velocities correspond with equally large armature velocities, and a proportionally large back electromotive force (EMF) opposing supplied amplifier voltage. The amplifier must overcome the back EMF and meet the power requirement of the load. With a sufficiently powerful amplifier in current regulation mode, a voltage increase could potentially avert shaker "force drop off" [13] at resonance. In voltage regulation mode a current increase might do the same. The available equipment (25 lbf (111 N) modal shaker) and the lightly damped c-beam structure invariably resulted in current limiting the shaker's operation well before the shaker's output force rating was reached. Operating in voltage regulation mode, at the current limit of the shaker, was also unsuccessful concerning reaching the specified output force of the shaker. Amplifier performance is suspected in this case but remains an issue for future consideration. In short, shaking the structure at resonance is more demanding of the entire excitation chain when the driven article's resonance behavior is significant relative to the shaker's operating characteristics. The structure-shaker interaction was foreseen by Kerley [9], who recommended sizing the shaker at five times the mass of the device under test.

The complex electromechanical coupling also complicates consideration of shaker placement. During the summer project, the team utilized trial and error to evaluate shaker positioning, quantified by input power measurement. Incidentally, the preceding impedance at resonance discussion was demonstrated by the active input power, the product of the RMS input force and velocity, driving the structure never exceeding 5 W. The highest input power was realized by shaking the structure at one quarter of the length of the assembly. Further modeling of the entire electromechanical system will allow a more precise determination of the optimum excitation point and to better choose the size of the shaker needed to reach the desired force levels.

4. Computational Analysis

A three-dimensional finite element model of the bolted c-beam structure was created using Cubit 15.4 [16]. The model was constructed of 8-node first-order hexahedral elements. It consists of two c-beam halves (1,250 hex elements), two nuts (26,880 hex elements), and two fasteners (356,430 hex elements). The fastener and nut geometry were created by sweeping a circular cross-section in a helical fashion similar to previous work by Grimmer et al [12]. This thread shape allowed the usage of hex elements to model thread-on-thread contact. The finite element software, SIERRA – Solid Mechanics [22], was used to apply preloads to the fasteners (step 1) and to apply a harmonic force input (step 2). Preloads were applied using a built-in preload subroutine that allowed a specified preload to be obtained for each joint. The subroutine applied incremental rotations to the bolts until the desired load was reached. An elastic-plastic material model was used for all the components (see Table 2). Parameters for the c-beams were borrowed from previous work using the experimental specimens [20]. The fasteners and nuts used properties from the strain gauge bolt specifications [11]. Additionally, a stick-slip friction model was implemented using a static coefficient of friction of 0.6 and a kinetic coefficient friction of 0.2, taken from literature on steel-on-steel contact [21].

Table 2: Material parameters used for each component in the FEM.

Component	**Young's Modulus [ksi]**	**Density [10^4 lbf-s^2-in^{-4}]**	**Poisson Ratio**	**Hardening Modulus [ksi]**	**Yield Strength [ksi]**
Beam B9A	30,850	7.30	0.283	1.26	275
Beam B9B	30,470	7.40	0.283	1.26	275
Bolts	30,450	7.35	0.283	1.26	150
Nuts	30,450	7.35	0.283	1.26	150

The harmonic force input followed the functional form $f(t) = A \sin \omega t$, where A is the amplitude, ω is the angular frequency, and t is time. The amplitude A varied from 30-2400 lbf (134 N to 10.7 kN), and ω was varied depending on the vibration mode of interest and the preload (higher preloads had higher values of resonant frequency ω). Thus, the linear frequencies predicted from the preloaded model ranged from 241-288 Hz for mode 1 and 321-350 Hz for mode 2. The harmonic loading scheme was applied for up to 80 milliseconds to the c-beam structure, corresponding to approximately 22 cycles of sine loading. This force was applied on a node at the center of beam B9B, similar to the location tested during the experiments. No other loadings or boundary conditions were specified, thus the beam assembly was in a free-free state. Explicit analysis was used for both the preloading and the vibrational loading steps due to the complex contact conditions within the threaded connections. This limited the maximum number of cycles that could be simulated during vibration assuming a maximum of two day run time for each load case using parallel computing with distributed memory.

5. Computational Results and Discussion

The simulations and the results were divided into two steps. The first step was to apply preload to the fasteners by rotating the nut onto the threaded bolt. The second step of each simulation applied an external harmonic force near resonance of the preloaded structure in the first step. The contour plots in Figure 6 show the von Mises stress for the first joint (labeled bolt 1 throughout) for one of the preload forces to a level of 150 lbf (670 N). In this case, the shank of the bolt appears to carry a uniform load as evidenced by the constant contour of stresses. Gradients in the stress state appear near the bolt head and threads due to geometry of the modeled joint. The maximum stress in the static preload analysis is in the first contacting thread in the fastener/nut interaction. At a preload of 150 lbf (670 N), there was no plastic deformation observed in any location of the finite element model. Other preload cases were simulated, and the results showed the plastic deformation occurred at a preload of 2800 lbf (12.5 kN), which occurred in the first contacting thread of the fastener.

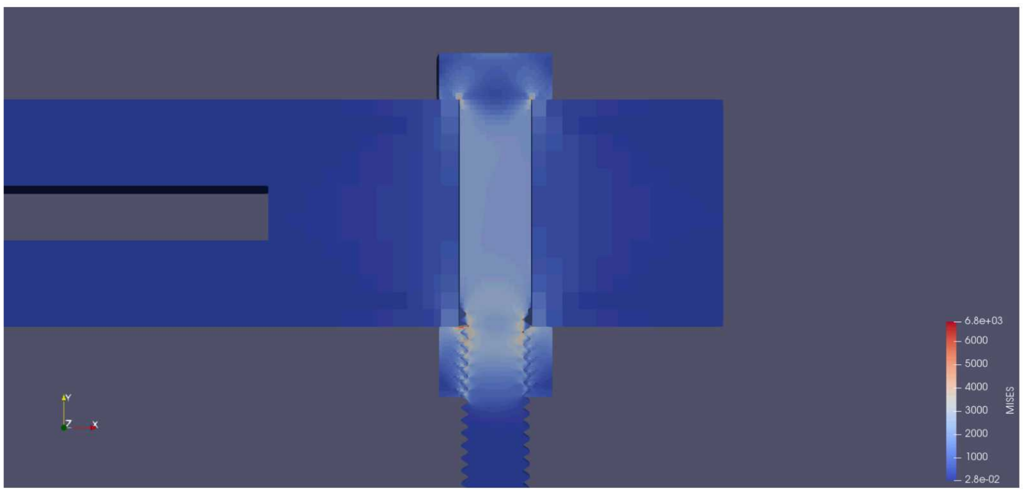


Figure 6: Von Mises stress in joint 1 of the c-beam structure after preloading to 150 lbf.

Prior to applying resonant loads with the externally applied force in the second step of the simulation, a simulation without harmonic loading was ran for the preloaded structure. These simulations are referred as unloaded simulations. The unloaded simulation for a 150 lbf (667 N) preload is shown in Figure 7, where the preload was monitored in each bolt by summing the normal forces acting on the bottom of each bolt head. In the unloaded simulations, preload loss was observed in all simulated preload levels. The decay in preload was linear for all cases, such as the decay shown in Figure 7, with higher preloads having larger magnitude slopes. Rotation was found to follow incremental slip for preloads below 500 lbf (2.2 kN), and cyclic rocking rotation for preloads 500 lbf (2.2 kN) and over. Decay in the preload was likely due to residual stress waves resulting from the preloading step, allowing self-loosening in the fasteners. Movement in the bolt/nut, especially at higher preload levels, allowed the joint to essentially ratchet itself off through incremental slip. This result highlights some of the challenges with using explicit methods for nonlinear contact mechanics problems.

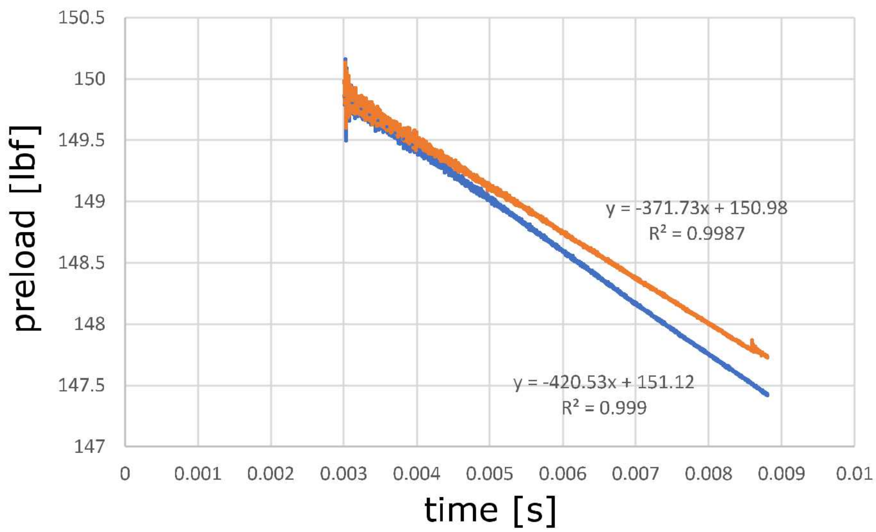


Figure 7: Preload in (blue) bolt 1 and (orange) bolt 2 during an unloaded simulation (no applied boundary conditions/loading) on a 150 lbf preloaded structure.

The step two simulation results in Figure 8 show the bolt preload and nut rotation for bolts preloaded to 150 lbf (667 N) with a 25 lbf (111 N) harmonic force applied at the shaker input location. This simulation covered roughly 20 cycles of sine loading at 275 Hz for mode 1 excitation and 325 Hz for mode 2. Monitoring the preload levels, the harmonic forcing appeared to have caused loosening in the joints over the course of roughly 70 ms. For the mode 1 simulation (top row), the preload in the bolts is spikes up throughout the loading scenario due to the bending of the c-beams during mode 1 excitation, which press up against the bottom of the bolt head. This trend is observed for mode 2 as well but to a lesser degree. Thus, the 'true' preload in the joint is closer to the minimum value of each oscillation, where the beam is nominally flat with the bolt head surface.

In the mode 1 simulation, the preload drops sharply at around 50 milliseconds. In the mode 2 simulation, the preload loss is linear, which could be largely due to the loosening behavior seen in the 'unloaded' simulation. The maximum force amplitude is much smaller compared to the mode 1 simulation. Preload also drops less, reaching 110 lbf (489 N) at the end of the simulation. Rotation of each bolt was calculated using node positions of a node at the center of the bolt head and a node at a corner of the head. The rotation follows a cyclic pattern, with decreases in rotation due to the displacement in the structure from the harmonic loading. True rotation in the joint follows a stick-slip pattern, with only positive increases in the rotation. The maximum rotation in the bolt increases with each cycle, indicating that the joint is loosening through ratcheting. At 50 ms, the periodic behavior is briefly suppressed, suggesting the possibility of complete slip between the bolt and the nut for this brief period. Rotation in the nut (not shown) was roughly 1.6-1.9 times the value of the bolt rotation for all simulations.

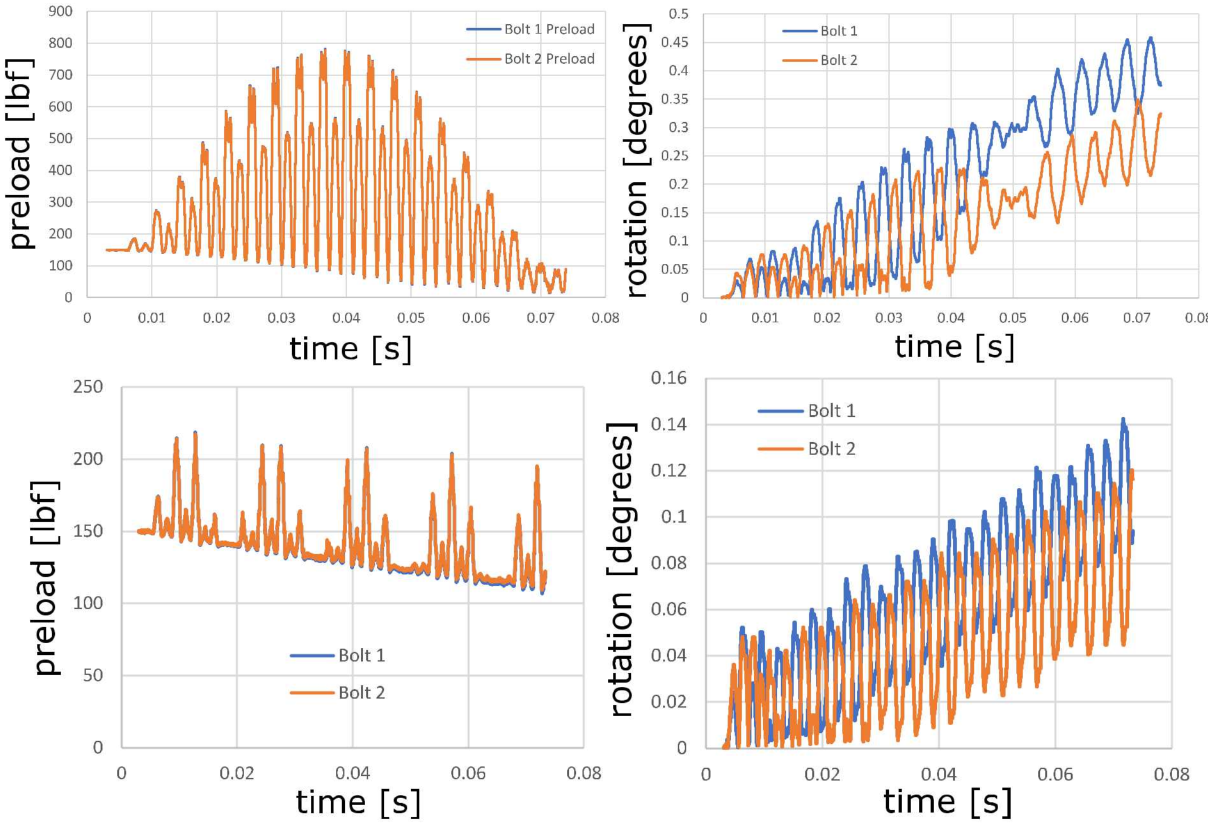


Figure 8: Preload and relative rotation in each joint during mode 1 (top row) and mode 2 (bottom row) excitation at 25 lbf (111 N) for a structure preloaded to 150 lbf (667 N).

The results in Figure 8 are difficult to interpret knowing that the bolt loses preload under a 'no load' condition. Subtracting off the data from the unloaded simulation loosening provides a better idea of what degree of the loosening is due to the harmonic force input. While it is not quantifiably correct to apply superposition to bolt self-loosening, it can provide some insight into the degree of loosening caused by the resonance. The results with the subtraction of unloaded response is shown in Figure 9. In the mode 1 simulation, the sudden decrease in preload becomes clearer at the 40 to 50 ms mark, along with a break in the periodic rotation behavior, resulting in a final preload of roughly 50 lbf. (222 N). The mode 2 simulation shows less preload loss, with a final preload level of 140 lbf (622 N) or about 6.7% of the initial preload. Higher preloads up to 2400 lbf (10.7 kN) were simulated under harmonic loading conditions, but the preload loss was difficult to distinguish between losses due to the unloaded simulation loosening versus loss due to the forcing input. The plot in Figure 10 shows that at a higher shaker force input of 100 lbf (445 N), loosening could be observed in 2400 lbf (10.7 kN) preloaded joints under mode 1 excitation, where about 200 lbf (889 N) of preload, or 8.3%, was lost due to a rotation in the bolts/nuts of roughly 0.2-0.6 degrees. In this figure, the unloaded simulation data has been subtracted from the raw preload and rotation data.

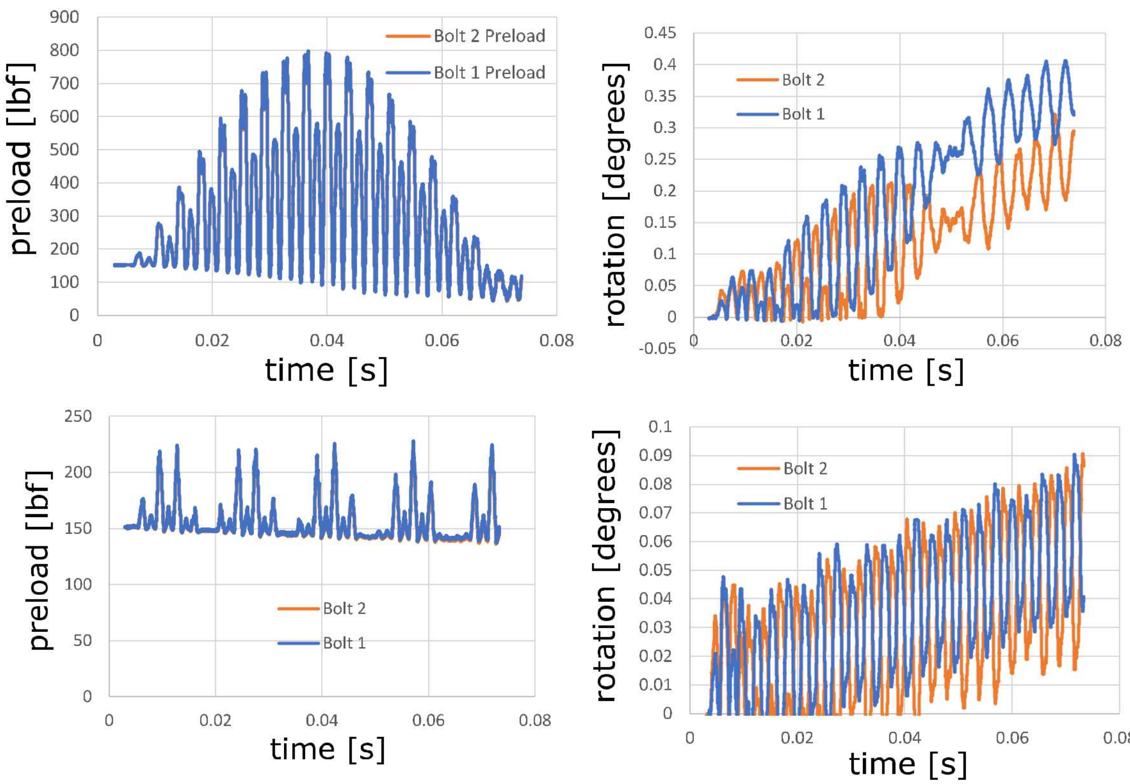


Figure 9: Preload and relative rotation in each joint during mode 1 (top row) and mode 2 (bottom row) excitation at 25 lbf (111 N) for a structure preloaded to 150 lbf (667 N) after subtracting off the unloaded empty simulation data.

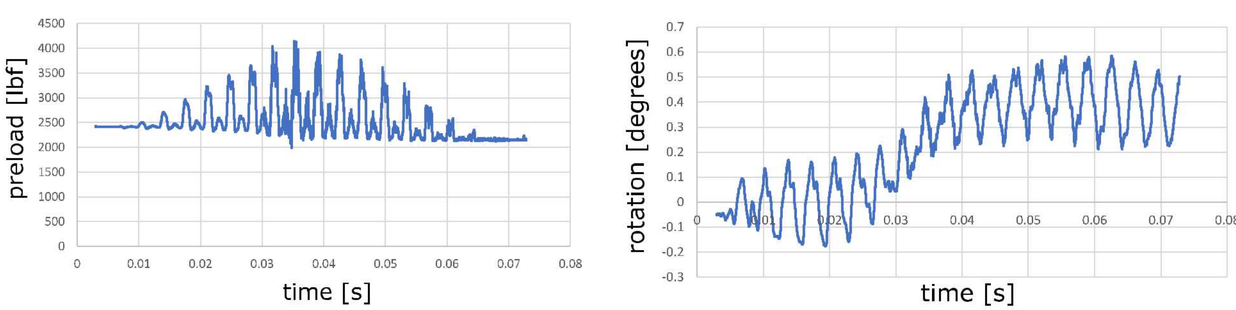


Figure 10: Preload (left) and rotation (right) in bolt 1 for a mode 1 excitation at 288 Hz, 100 lbf input force for a structure preloaded to 2400 lbf [10.7 kN].

Thus, the model demonstrates the ability to simulate bolt self-loosening under resonance using a fully-threaded, high-fidelity FEM. During each harmonic loading simulation, the contacting threads in each joint experience oscillating contact stress. A contour plot of the stress state during oscillation is shown in Figure 11 shows how the contact in the threads is intermittent and not uniform around the circumference of the joint. The oscillation in the thread contact is not consistent across the whole joint, causing some fraction of the threads to be in contact while others are not. This intermittent contact allows the bolt and the nut to rotate with respect to each other through partial slip. This is similar to behavior investigated previously by Pai and Hess [6].

Figure 11: Snapshot of the contact stress in joint 1 during mode 1 excitation at 25 lbf (111 N)for a structure preloaded to 150 lbf (667 N).

6. Conclusions

A direct comparison of the experimental and computational results is not appropriate given the different stages of self-loosening observed in the studies. Experimentally, Stage I preload losses were observed with a suspected cause of plastic deformation of bolt thread tips. The classification of Stage I follows from preload loss decay trends, specifically initial exponential drops followed by linear decreases plateauing with time. The initial exponential drop is not present when repeat modally excited loosening tests are performed without intermediately torqueing fasteners. This observation suggests the initial exponential preload drop follows from conditions set by fastener installation. For vibratory environments, torqueing fasteners slightly beyond the desired preload may compensate the predictable effect. Relative fastener rotation was not observed during these studies due to the limited structure input power. Mode 1 excitation was used for the majority of tests to maximize c-beam input power.

Computationally, self-loosening followed that of Stage II behavior with gross slip between the bolt and nut. The Stage II classification stems from the demonstration of relative fastener rotation using nodal displacements, as well as large preload losses observed in many tests. A clear benefit of an accurate computational model is the ability to specify a desired input force. Stage I loosening in the computational model may be realized by artificially lowering the elastic moduli of the threads or increasing the fidelity of the thread mesh. Doing so may lower the preload value at which plastic deformation is observed in the modeled threads, more closely matching experimental observations. Modeling the threads as a separate body bonded to the bolt shank may help preserve other mechanics of the joint. Such an approach may help more realistically model complexities in both thread geometry imperfections and material properties. Sufficient plastic deformation in the thread tips may help better match the experimental Stage I results. To this end, imperfectly matched internal and external threads, more in line with reality, may see preload losses as the threads settle in under dynamic excitation. Fretting wear, also implicated in Stage I loosening, is not currently computationally feasible concerning finite element analysis. Here thread mesh fidelity may help better resolve contact surfaces.

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References

[1] G. H. Junker, "New criteria for self-loosening faster under vibration," vol. 78, pp. 314–335, 1969.
[2] Y. Jiang, M. Zhang, and C.-H. Lee, "A Study of Early Stage Self-Loosening of Bolted Joints," *J. Mech. Des.*, vol. 125, no. 3, p. 518, 2003.
[3] J. Liu *et al.*, "Experimental and numerical studies of bolted joints subjected to axial excitation," *Wear*, vol. 346–347, pp. 66–77, 2016.
[4] M. Zhang, L. Lu, W. Wang, and D. Zeng, "The roles of thread wear on self-loosening behavior of bolted joints under transverse cyclic loading," *Wear*, vol. 394–395, no. July 2017, pp. 30–39, 2018.
[5] R. I. Zadoks and X. Yu, "An Investigation of the Self-loosening Behavior of Bolts under Transverse Vibration," vol. 208, pp. 189–209, 1997.
[6] N. G. Pai and D. P. Hess, "Three-dimensional finite element analysis of threaded fastener loosening due to dynamic shear load," *Eng. Fail. Anal.*, vol. 9, no. 4, pp. 383–402, 2002.
[7] M. Zhang, Y. Jiang, and C.-H. Lee, "Finite Element Modeling of Self-Loosening of Bolted Joints," *J. Mech. Des.*, vol. 129, no. 2, p. 218, 2007.
[8] S. Fernando, "Mechanisms and prevention of vibration loosening in bolted joints," *Aust. J. Mech. Eng.*, vol. 2, no. 2, pp. 73–92, 2017.
[9] J. J. Kerley, "An application of retroduction to analyzing and testing the backing off of nuts and bolts during dynamic loading," NASA Technical Memorandum 4001, NASA Technical Reports Server (NTRS). [Washington, D.C.]:[U.S. National Aeronautics and Space Administration]. 1987.
[11] Strainsert, "Standard Internally Gaged Hex Head Cap Screws, SXS Series," 2019.
[12] P. W. Grimmer, J. P. Mersch, J. A. Smith, Y. B. Veytskin, and D. F. Susan, "Modeling empirical size relationships on load-displacement behavior and failure in threaded fasteners," *AIAA Scitech 2019 Forum*, no. January, pp. 1–10, 2019.
[13] P. S. Varoto and L. P. Rodrigues de Oliveira, "On the force drop off phenomenon in shaker testing in experimental modal analysis," *Shock and Vib.*, vol. 9, no. 4, pp. 165–175, 2002.
[14] C. DellaCorte, S. A. Howard, D. P. Hess, "Preload Loss in a Spacecraft Fastener via Vibration-Induced Unwinding," NASA/TP—2018-219787, NASA Technical Reports Server (NTRS). [Washington, D.C.]: [U.S. National Aeronautics and Space Administration]. 2018.
[15] "Simcenter SCADAS." https://www.plm.automation.siemens.com/global/en/ products/simcenter/simcenter-physical-testing.html. Siemens AG, 2019. Last accessed, Sept 2019.
[16] Geometry and Mesh Generation Toolkit. Ver 15.4 2019. https://cubit.sandia.gov/. Last accessed, Sept 2019.
[17] "Force Sensors." https://www.pcb.com/sensors-for-test-measurement/force-sensors. PCB Piezotronics, 2019. Last accessed, Sept 2019.
[18] Bickford, J.H., "An Introduction to the design and behavior of bolted joints", 3rd Edition, Taylor & Francis Group, New York, 1995.
[19] Mersch, J. P., Smith, J. A., Johnson, E. P., "A Case Study for the Low Fidelity Modeling of Threaded Fasteners Subject to Tensile Loadings at Low and High Strain Rates," ASME Pressure Vessels and Piping Conference, PVP2017-65518, ASME, Waikoloa, HI, 2017.
[20] Fronk, M., Guerra, G., Southwick, M., R.J. Kuether, Brink, A.R., Tiso, P., Quinn, D.D., "Predictive Modeling of Bolted Assemblies with Surface Irregularities," in *37th International Modal Analysis Conference (IMAC XXXVII)*, Orlando, Florida, January 2019.

[21] CRC Handbook of Physical Quantities. Boca Raton, FL: CRC Press, 1997: 145-156.
[22] Sierra Solid Mechanics Team, "Sierra/Solid Mechanics 4.52 User's Guide," Sandia National Laboratories, Albuquerque, NM, 2018. SAND2019-2715.