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Original Research Article

# FEA-based prediction of self-loosening in unsecured u-bolted joints

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**ABSTRACT**

U-bolted joints are widely used in heavy-duty vehicle suspension systems to ensure structural integrity under dynamic service conditions. The preload applied during assembly is the key factor that keeps the joint secured and prevents premature separation. However, under repeated transverse loading, preload can gradually decrease, leading to self-loosening of the joint and potential safety risks. This study aims to develop a predictive framework for identifying the self-loosening behavior of unsecured U-bolted joints, focusing on preload degradation under realistic transverse loading conditions. A finite element analysis (FEA) model representing a typical U-bolt connection between a tube axle and suspension system was developed. Cyclic transverse displacements, derived from combined vehicle operating loads, were applied to simulate vibration-induced loosening. The preload decay trend was monitored and extrapolated to determine the loosening threshold. The numerical results reveal a clear progressive reduction in preload under cyclic shear loading, allowing estimation of the vibration cycles leading to critical torque loss. The study demonstrates that unsecured U-bolt joints exhibit predictable loosening behavior when subjected to sustained transverse excitation. The novelty of this work lies in introducing a cycle-based predictive FEA framework specifically tailored for unsecured U-bolted joints under realistic multi-directional service conditions. By correlating numerical predictions with representative road test behavior, the approach enables reduction of costly experimental campaigns while improving the reliability of maintenance planning. This engineering-oriented methodology supports early detection of loosening risk, establishment of optimized re-torque intervals, and implementation of proactive predictive maintenance strategies, thereby enhancing the durability and safety of heavy-duty vehicle suspension systems.

**Keywords:** Self-Loosening, U-Bolt, Preload, Shear Load, Torque Loss, FEA

## 1. Introduction

Threaded fasteners are extensively employed in mechanical systems for detachable connections due to their ease of assembly, maintenance, and repair. However, under cyclic or transverse vibration loading, bolted joints may suffer from preload degradation and self-loosening, compromising connection integrity [1]. In heavy-duty vehicle suspension systems, where joints are subjected to complex combinations of shear, vibration and dynamic service loads, such effects become critical. Among these, U-bolted connections are especially vulnerable because their loosening directly affects axle-to-suspension integrity, potentially leading to severe operational and safety consequences. Previous studies have demonstrated loosening mechanisms under transverse or sine-on-random excitation [2] and evaluated countermeasures such as anti-loosening washers and special nuts (Zhao [3], Study on Tightening, Anti-Loosening, and Fatigue Resistance Performances of Bolted Joints with Different Anti-Loosening Washers and Nuts., 2023. Nevertheless, quantitative cycle-based predictions for unsecured U-bolts under realistic multi-directional loading remain lacking, a gap this study aims to fill. Below Figure 1 illustrates the practical relevance of the investigated problem by showing where loosening-critical joints are in real applications.

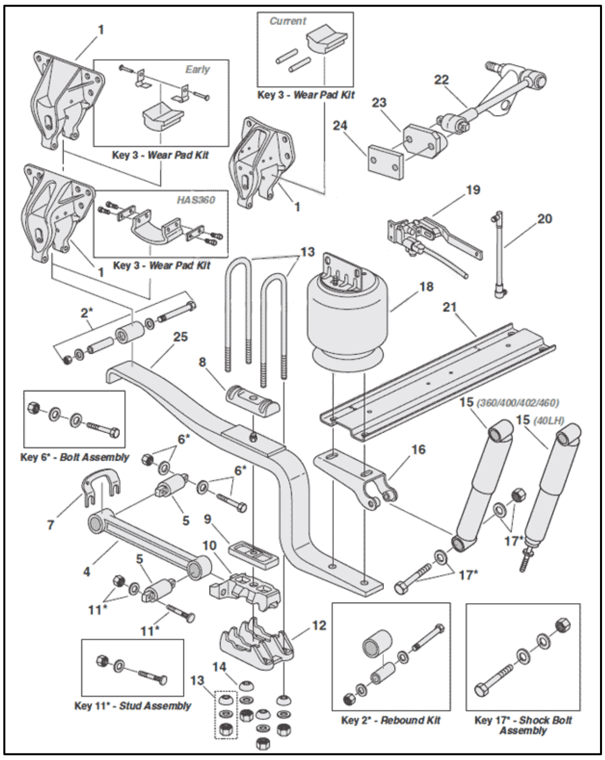


Figure 1: Air suspension system of a heavy-duty vehicle, containing numerous threaded joints including U-bolts-no13.

Extensive research has been conducted on vibration-induced self-loosening of threaded fasteners since Junker's pioneering experiments in the 1960s [4], which demonstrated that transverse vibration is the primary driver of loosening. Subsequent studies have expanded this knowledge by investigating the influence of different load types, configurations, and boundary conditions. Sun et al. [5] examined the effect of transverse load amplitude on loosening behavior, highlighting the sensitivity of preload decay to displacement magnitude. Yang et al. [6] proposed normalized stress-based evaluation methods to predict loosening life across different bolt sizes, primarily focusing on standard bolted joints. Du et al. [2] explored bolt loosening under sine-on-random excitation, emphasizing the role of complex vibration spectra, while Yamamoto et al. [7] investigated offset loading in thin-plate joints, revealing the influence of joint geometry on loosening mechanisms. In parallel, Güler et al. [8] introduced artificial intelligence-based predictive models for bolted joints used in automotive chassis systems, reflecting the shift toward computational approaches.

Despite these advancements, most existing studies predominantly focus on conventional bolted joint geometries and simplified laboratory conditions, often involving uniaxial or idealized transverse loading. The loosening behavior of unsecured U-bolt joints particularly those used in heavy-duty suspension systems — remains insufficiently explored, especially under combined multi-directional displacement inputs that more accurately reflect real-world vehicle operating conditions. Furthermore, quantitative cycle-based predictions of preload loss for such configurations are limited, leaving a notable gap in design-stage evaluation tools for suspension engineers.

In this study, a 12.5-ton GAWR lift-axle air suspension system of a 10×2 configuration Class-8 (Class-8 = GVWR ≥ 33,001 lb (≈ 14,969 kg), Zhao et. al. [9]) heavy-duty truck manufactured for Asian markets was examined, as illustrated in Figure 2.

The vehicle shown in Figure 3 exhibits symptoms of self-loosening, primarily because

Figure 2: Class-8 heavy-duty truck model employed in the analysis; the manufacturer identity is withheld for confidentiality reasons

its bolted joints are assembled without any secondary locking mechanism (i.e. unsecured joints), likely due to cost-driven design decisions. In the absence of such security features, the preload retention capacity of the joint becomes highly sensitive to vibration-induced slip, which accelerates preload decay. Various mechanical and chemical locking strategies, such as thread-locking adhesives, nylon-insert or all-metal prevailing-torque nuts, spring washers, knurled inner–outer washers, and double-nut arrangements are commonly employed to enhance resistance against self-loosening in dynamic environments.

Figure 3: Self-loosening observed in a different joint of the truck during the road test, shown as a representative example of the phenomenon studied.

It is hypothesized that combined ±0.5 mm longitudinal and ±1.0 mm lateral cyclic displacements induce a critical 10% preload loss within a specific vibration cycle in an unsecured M27 U-bolt joint. This hypothesis is evaluated through a finite element analysis framework specifically developed to capture the loosening mechanism under worst-case service displacements derived from vehicle loading scenarios.

The novelty of this study lies in addressing the self-loosening behavior of unsecured U-bolt joints by incorporating multi-axial cyclic displacement inputs in a geometrically realistic U-bolt configuration, moving beyond traditional simplified bolted joint models. By establishing a cycle-based predictive framework capable of estimating torque loss thresholds, the present work provides a conservative and practical approach for early-stage design validation and predictive maintenance planning, while also reducing reliance on costly, limited and time-consuming experimental campaigns.

## 2. FEA Simulation

The commercial heavy duty vehicle model *Figure 4* was used to measure system vibration at U-bolt fasteners and use them as input for analyzing self-loosening behavior. The product was modeled using CATIA (V5 2018), a commercial 3D CAD program.

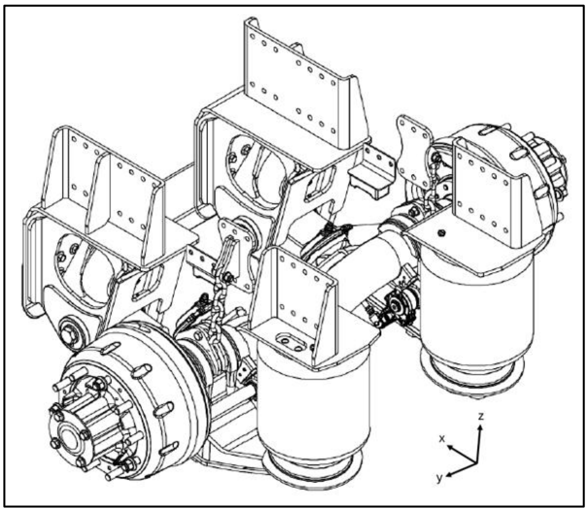

Figure 4: CAD model of the air suspension system used in the simulations.

Below *Figure 5* illustrates the layout of the suspension system as mounted on the truck chassis. In the simulation model, the free-body diagram of the assembly was constructed to accurately represent the actual vehicle configuration.

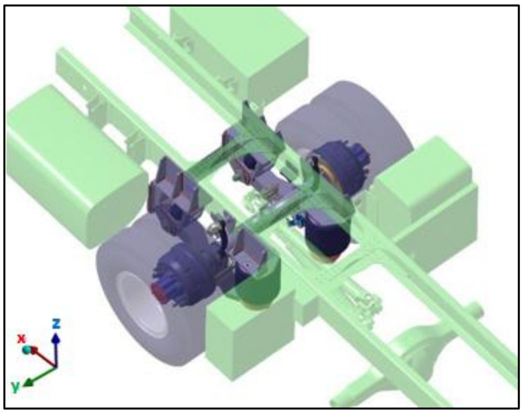

Figure 5: Layout of the suspension system mounted on the truck chassis

### 2.1. Assumptions

The 3D model was imported into FEA software program, ANSYS Mechanical Workbench (2023 R1). During the FEA analyses, the following assumptions were made:

Element size: A quadratic triangular element mesh was employed throughout the model to improve stress resolution in regions with high gradients. A global element size of 4 mm was applied to both the U-bolt and nut, while the contact regions between the bolt threads and nut were refined three times to ensure accurate contact pressure distribution and slip behavior representation. This mesh strategy provided a balance between computational efficiency and solution accuracy.

A mesh sensitivity analysis was conducted to ensure numerical robustness. Three mesh densities were evaluated: coarse (6 mm global size), medium (4 mm), and refined (3 mm). The peak von Mises stress at the first engaged thread varied by less than 2% between the medium and refined meshes. The medium mesh was therefore selected as the optimal configuration, balancing accuracy and computational cost. This mesh convergence evaluation supports the reliability of the results within the simulated displacement range.

1. Joint configuration: An unsecured M27 U-bolt with nut was modeled, as it represents a widely used fastener in heavy-duty suspension systems.
2. Contact type and solving method: Contact interactions between the bolt threads and the nut were modeled using the Augmented Lagrange formulation, which offers stable convergence while accurately enforcing normal and tangential constraints. The Newton–Raphson iterative method was used with force-based convergence criteria to ensure solution stability during nonlinear contact and large displacement analysis.
3. Material properties: The material behavior of the bolt and nut was defined using isotropic linear elastic-plastic properties. The Young's modulus was set to $E = 2\times10^5$ MPa, Poisson's ratio $\nu = 0.3$, yield strength $\sigma_\gamma = 250$ MPa, and ultimate strength $\sigma_u = 460$ MPa. A bilinear isotropic hardening model was adopted to capture post-yield behavior while maintaining computational stability.
4. Preload definition: The initial preload was introduced via torque control using a tightening torque of 1000 Nm, consistent with industrial tightening practice for M27 U-bolts. The pretension force was calculated using the standard torque–tension relationship. The tightening torque, $F$is the resultant preload, $d$is the nominal bolt diameter, and $K$is the nut factor, a dimensionless constant that accounts for frictional effects and joint geometry in practical assemblies. The nut factor is widely used in engineering practice to relate torque to preload and is influenced by thread and underhead friction as well as surface conditions. Typical nut factors for unlubricated steel fasteners lie in the range of 0.10–0.20 Hadi et al. (10).

$F=T/(D\cdot\kappa)$, where;

T = 1000 Nm (applied torque),

D = 0.027 m (nominal bolt diameter),

κ = 0.16 (nut factor),

resulting in an initial pretension force of 231,481.48 N.

5. Loading conditions: Extreme transverse displacements were considered: ±0.5 mm in the longitudinal (x) and ±1.0 mm in the lateral (y) directions. These values were derived from vehicle dynamics simulations under worst-case vertical, braking, and cornering loads. Since clamping force decreases with larger displacement amplitudes ( (11), these conditions were selected to represent the most severe service scenario. Displacements were applied as fully reversed cyclic (0.25Hz) at the bolt tip while the nut surface was constrained, in accordance with the bolt pretension and self-loosening methodology implemented in ANSYS Mechanical. This approach is consistent with the software's established simulation framework for modeling preload degradation and rotational loosening, where applying displacement at the bolt shank or tip effectively represents the transfer of relative motion across the joint interface under transverse excitation conditions. This configuration ensures stable contact behavior and accurately captures friction-induced slip mechanisms driving self-loosening.
6. Load exposure: Although extreme loads cannot occur continuously during the

operational life of the vehicle, they were conservatively applied throughout the simulations to ensure safety-side estimations.

7. Failure criterion: Loosening was defined as a 10% drop in preload torque, resulting in a limit torque of 900 Nm (equivalent to 208,333 N remaining preload). This criterion is widely accepted in literature as the functional failure threshold of bolted joints.

Friction coefficient (μ): A coefficient of 0.1 was applied between bolt threads and nut. This conservative choice allowed faster convergence in analyses and prevented overestimation of loosening cycles. Iterations with μ = 0.12 confirmed that higher friction delays loosening but does not fundamentally alter the trend. Below *Table 1* summarize the model assumptions, boundary conditions and solver settings.

## 2.2. Limitations

This study has several limitations that should be acknowledged.

1. The friction coefficient at the thread interface was assumed constant, whereas in real applications friction evolves with surface wear, lubrication breakdown, and micro-welding phenomena.
2. Surface roughness and asperity-level contact mechanics were not explicitly modeled, although such effects are known to influence microslip initiation.
3. The applied cyclic displacement is

Table 1: Summary of Model Assumptions, Boundary Conditions and Solver Settings

| Category | Parameter | Value / Description |
|---|---|---|
| **Joint Configuration** | Joint type | Unsecured M27 U-bolt with nut |
| **Element & Mesh** | Element type | Quadratic triangular elements |
| | Global element size | 4 mm (bolt and nut) |
| | Local refinement | 3× refinement at bolt–nut thread contacts regions |
| | Mesh strategy | Optimized for accurate stress with computational efficiency |
| **Material Properties** | Young's modulus (E) | $2 \times 10^5$ MPa |
| | Poisson's ratio (ν) | 0.3 |
| | Yield strength ($\sigma_Y$) | 250 MPa |
| | Ultimate strength ($\sigma_u$) | 460 MPa |
| | Hardening model | Bilinear isotropic hardening |
| **Contact Definition** | Contact formulation | Augmented Lagrange |
| | Contact interfaces | Frictional contact between bolt and nut threads |
| | Friction coefficient (μ) | 0.1 |
| **Boundary Conditions** | Constrained region | Nut surface fixed |
| | Displacement application | Bolt tip |
| | Displacement type | Fully reversed cyclic |
| | Longitudinal displacement (x) | ±0.5 mm |
| | Lateral displacement (y) | ±1.0 mm |
| **Load Definition** | Load source | Worst-case vehicle dynamics conditions (combined vertical, braking, cornering) |
| | Load exposure | Continuously applied (conservative assumption) |
| | Purpose | Safety-side estimation of loosening severity |
| **Preload Definition** | Tightening torque | 1000 Nm |
| | Torque–tension relation | $F = T / (D \cdot \kappa)$ |
| | Nominal diameter (D) | 0.027 m |
| | Nut factor (κ) | 0.16 |
| | Initial pretension force | 231,481.48 N |
| **Failure Criterion** | Loosening definition | 10% preload torque loss |
| | Critical torque | 900 Nm |
| | Remaining preload | 208,333 N |
| **Solver Settings** | Analysis type | Nonlinear static with cyclical loading (0.25Hz each cycle) |
| | Solution scheme | Newton–Raphson iterative method |
| | Convergence criteria | Force-based convergence |
| | Nonlinearity sources | Large deformation + contact nonlinearity |
| **Modeling Assumptions** | Load scenario | Extreme service condition assumption |
| | Objective | Conservative life prediction & safety evaluation |
| **Validation Strategy** | Intended correlation | Road test trend comparison should be performed |

simplified a fully reversed triangular waveform, whereas real service conditions may involve mixed-frequency and multi-harmonic excitation.

4. The FE model represents a single U-bolt configuration and does not account for multi-bolt interactions or flange deformation.

5. Thermal effects and preload loss due to temperature cycling are not included.

6. The absence of experimental validation limits the quantitative accuracy of the prediction, although the observed trends align well with established experimental results.

These limitations will be addressed in future work through combined numerical–experimental studies.

### 2.3. Analyses

The FEA study was conducted using ANSYS Mechanical. The workflow consisted of:

1. <u>Loading cases</u>: Extreme service loads (vertical, braking, cornering, and longitudinal) were applied individually and in combinations to the suspension system. Below, *Figure 6* shows sample loading of the system with vertical direction. This figure demonstrates how service loads were translated into input conditions for the FEA model.

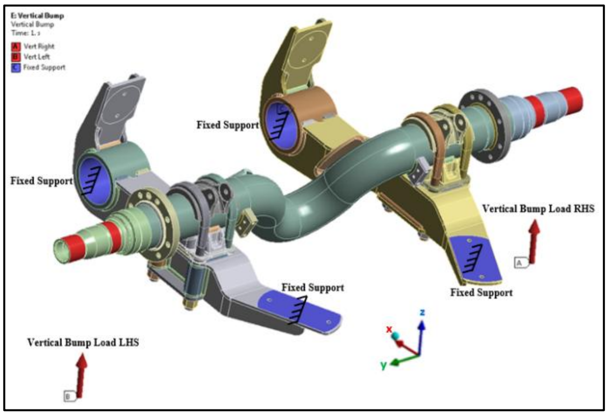


Figure 6: Example of vertical loading applied to the suspension system in ANSYS®.

2. <u>Displacements in U-Bolts</u>: The displacements correspond to extreme service loads of the vehicle were obtained at the tip of the U-bolts. These displacements will represent transverse loading in the loosening analysis. Below, *Figure 7* shows sample displacements of the U-bolts corresponding to vertical loading. The values are representative of the shear-induced motions that contribute to bolt loosening. Displacement of U-bolt was extracted for both transverse directions; longitudinal (x) and lateral (y), please refer to Table 2 for details. Table 2 summarize the resulting displacements at the U-bolt tips.

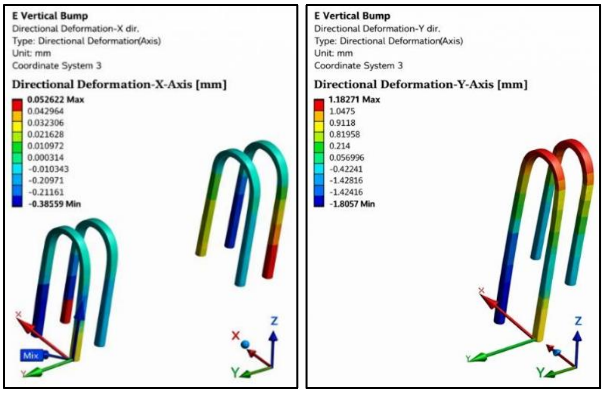


Figure 7: Resulting displacements (both longitudinal (x) and lateral (y) directions) at U-bolt tips under vertical loading in ANSYS.

Table 2: Vehicle Loading Types and Corresponding Displacement Values

| Vehicle Load Types | M27 U-Bolt Ends | |
|---|---|---|
| | Max./Min. Displacements in Longitudinal (x) Direction [mm] | Max./Min. Displacements in Lateral (y) Direction [mm] |
| Vertical Load | 0.52/-0.38 | 1.04/-1.02 |
| Braking Forward Load | 0.15/-0.15 | 0.37/-0.37 |
| Braking Backward Load | 0.26/-0.2 | 0.53/-0.52 |
| Cornering Right Load | 0.17/-0.11 | -0.11/-0.3 |
| Cornering Left Load | 0.46/-0.31 | 0.96/-0.5 |
| Longitudinal Load | 0.34/-0.58 | 1.09/-1.07 |

The displacement on the bolt ends varies in between above values and during the operation conditions; the bolts will encounter those shaking situations in various combinations.

3. <u>Worst-case selection</u>: To ensure a conservative approach, the highest displacements observed (±0.5 mm in x and ±1.0 mm in y) were selected and rounded in *Table 3*. These were applied cyclically to maximize shear-induced loosening.

4. <u>Simulation of vibration</u>: Longitudinal (x) and lateral (y) displacements were combined to maximize the movement of the bolt in the analytic coordinate and applied to the system as cyclic load which will lead to self-loosening of the bolt. The displacement cycle was

applied to the individual U-bolts in a closed repetitive vibration of the bolt-nut connection (*Figure 8*).

Table 3: Rounded Displacements for Worst Loading Scenario

| Bolt Type | M27 |
|---|---|
| Max./Min. Displacements in Longitudinal (x) Direction [mm] | 0.5/-0.5 |
| Max./Min. Displacements in Lateral (y) Direction [mm] | 1.0/-1.0 |

Each full loop corresponded to one cycle (4 steps) of transverse vibration. In the below *Figure 8*, at left hand side, the xy plane can be seen in bolt-nut 3D model drawn by using a commercial CAD program CATIA (V5 2018) and at right hand side, the selected vibration starts and end positions can be seen:

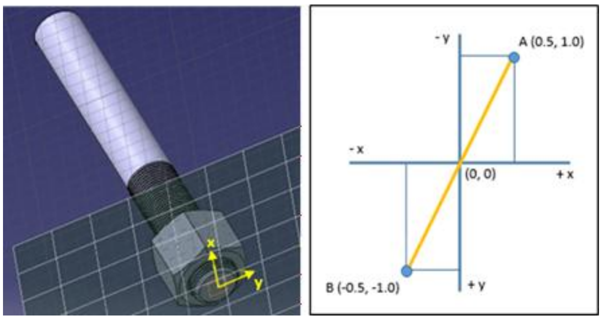


Figure 8: CAD model of the bolt-nut pair (left) and extreme displacement positions on the xy-plane (right). This figure shows how cyclic transverse displacements were defined to simulate vibration-induced loosening

5. <u>Preloading</u>: The U-bolts were preloaded to the defined torque value (1000 Nm which equals 231,481N for M27 bolt). *Figure 9* and *Figure 10* illustrate the preload application on the bolt and tabulated stepwise, respectively. This provides verification that the target initial pretension (231.481 N) was correctly implemented and locked in the ANSYS® model.

After that, the loosening mechanism applied on the system by adding displacements simulating the vehicle on road conditions. First, constraints were given as frictional contact between bolt and nut, and as fixation at outer faces of nut, please refer to B displacement at below *Figure 11*:

Then, the transverse displacements were applied as a continuous vibration that will lead self-loosening on the bolt-nut connection in the FEA model, please see below C displacement constraints in *Figure 12*:

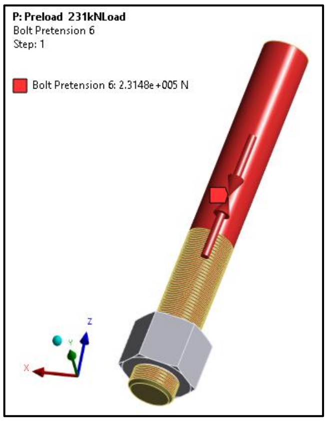


Figure 9: Application of bolt pretension to 1000 Nm torque prior to applying vibration displacements

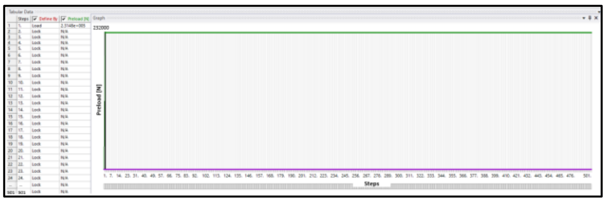

Figure 10: Preload application at 1st step and lock it for the rest of the steps.

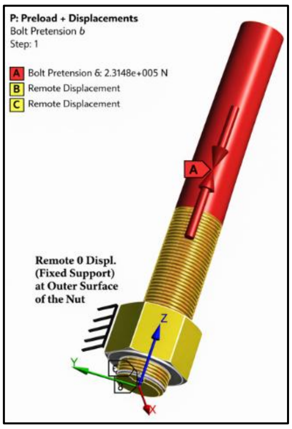


Figure 11: Fixation of the nut outer faces to prevent rigid-body motion during vibration simulations

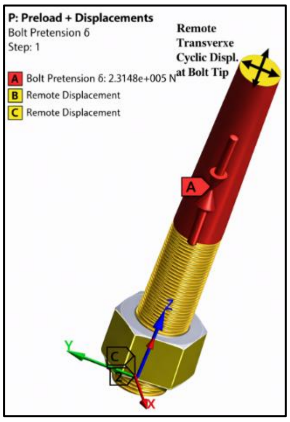


Figure 12: Application of transverse displacement cycle, repeated shear motion leading to self-loosening

The route starts from 0 position then goes to point A, follow with position 0, then point B and finally returns to 0 and this equals to 1 cycle (4 steps) and takes 4 seconds (0.25Hz). This route continues all over again until the specified cycle reaches. The preloaded bolt, tabular sample input of the displacement for 12 steps (*Table 4*) and vibration graph of the joint can be seen in the below *Figure 13*. This figure illustrates how the vibration loading sequence was applied to the joint in the simulation. Cyclic displacement load in transverse directions is expected to drop pretension on the bolt and consequently lose the U-bolt.

Table 4: Tabular Preload Input Data for 3 Cycles (12 Steps) in ANSYS®

Tabular Data

| | Steps | Time [s] | X [mm] | Y [mm] |
|---|---|---|---|---|
| 0 | 1 | 1. | 0. | 0. |
| A | 2 | 2. | 0.5 | 1. |
| 0 | 3 | 3. | 0. | 0. |
| B | 4 | 4. | -0.5 | -1. |
| 0 | 5 | 5. | 0. | 0. |
| A | 6 | 6. | 0.5 | 1. |
| 0 | 7 | 7. | 0. | 0. |
| B | 8 | 8. | -0.5 | -1. |
| 0 | 9 | 9. | 0. | 0. |
| A | 10 | 10. | 0.5 | 1. |
| 0 | 11 | 11. | 0. | 0. |
| B | 12 | 12. | -0.5 | -1. |

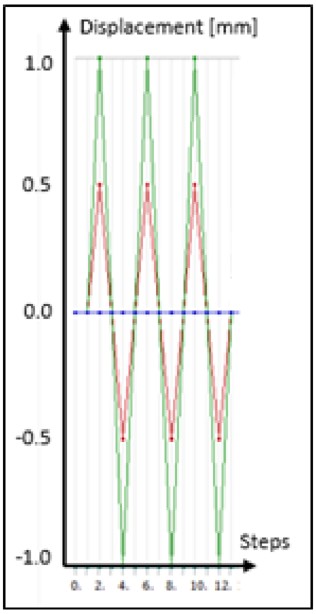


Figure 13: Graphical Illustration of 12-steps cyclic displacement input for 3 vibration cycles in ANSYS®.

6. <u>Analysis limits</u>: Due to computational constraints, the model remained stable up to 500 steps. Beyond this point, divergence and abnormal preload fluctuations occurred. Therefore, results were recorded up to 380 steps, which provided reliable convergence (*Figure 14*). The data reveal a progressive decrease in preload, which was later extrapolated to estimate the loosening life.

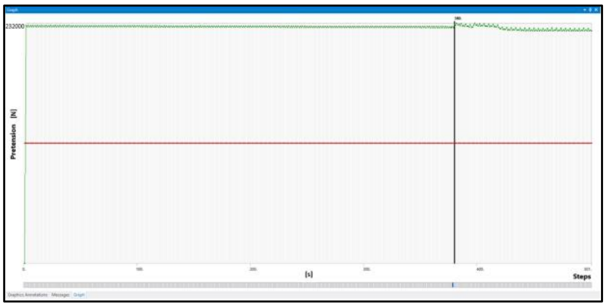

Figure 14: Pretension drop curve of the bolt throughout 380 vibration steps in ANSYS®.

To optimize computational efficiency, the simulation was performed for the first 380 load steps, and the resulting preload trend was extrapolated to predict when the pretension dropped below 90% of the initial torque level. Also, the shear stress induced by the transverse vibration on the bolt can be seen in below cross section *Figure 15*.

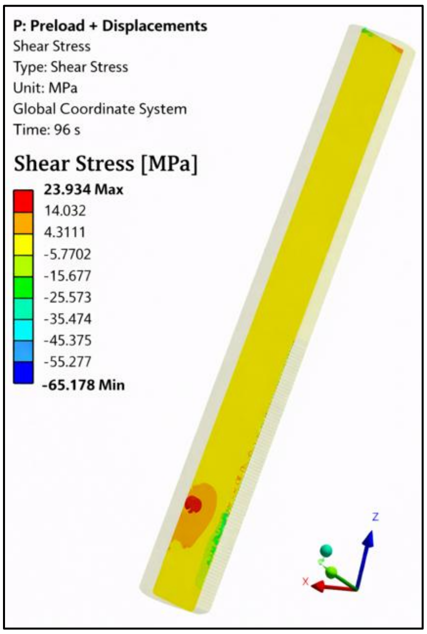


Figure 15: Shear stress distribution on the bolt cross-section under transverse vibration at 96$^{th}$ step/second.

The shear stress distribution within the U-bolt threads exhibits a distinct concentration pattern characteristic of shear-dominated microslip. Peak stresses occur at the first engaged thread on the nut side, reaching approximately 20-25 MPa under combined pretension and transverse displacement. The stress gradient decreases along subsequent threads, with the third and fourth engaged threads carrying roughly 40–55% of the peak value in agreement with the literature (ref). These values remain below the material's yield strength but are within ranges where cyclic plasticity and progressive preload degradation are feasible, which supports the observed early-stage loosening trends. Stress concentration highlights the mechanism responsible for gradual loosening (*Figure 16*).

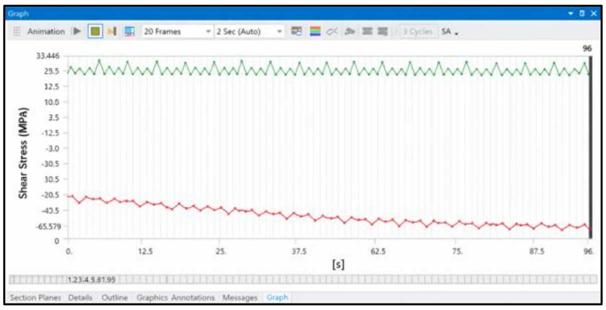


Figure 16: Graphical Illustration of shear stress gradual decay in 96 steps/seconds.

## 3. Results

### 3.1. FEA results

The bolt and nut connection were modelled in ANSYS program, and several iterations had been performed to obtain characteristics of the bolt loosening and to get as many steps as to get dropped preload on the bolt under constant vibration of the shear load. The largest interval of the analysis was obtained for 95 cycles of extreme displacements (from 0.5, 1.0 to -0.5, -1.0) without distortion of the mesh elements or non-convergence of the analysis model. In the literature, smaller bolts were used, and FEA programs are more capable of dealing with such less loading on the mesh elements [12]. In this study, a relatively larger bolt was used as M27. Below *Figure 17* shows that preload decreased progressively with each vibration cycle. A linear curve fit (with $R^2$=0.9979*)* was applied to the drop curve, providing a robust basis for extrapolation to higher cycle counts.

### 3.2. Extrapolation of the results

Based on the fitted curve, the number of cycles required to reach the 10% torque loss threshold was estimated. *Figure 18* illustrates this extrapolation, $R^2 = 0.9979$. The predicted loosening life corresponds to 2170 vibration cycles under worst-case loading.

As summarized in *Table 5*, the unsecured M27 U-bolt is expected to reach 10% torque loss after approximately 2170 cycles of extreme shear vibration.

Although no quantitative back-calculation was executed within the current scope, the displacement amplitudes were mapped from worst-case load cases generated through vehicle dynamics models. Future work will correlate in-service measured accelerations with preload loss for direct calibration.

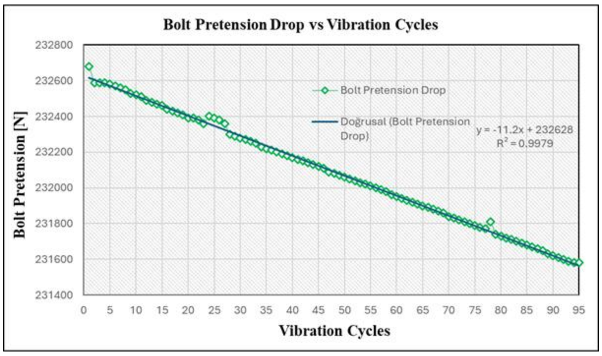


Figure 17: Pretension drop vs. vibration cycles for 95 cycles of shear loading.

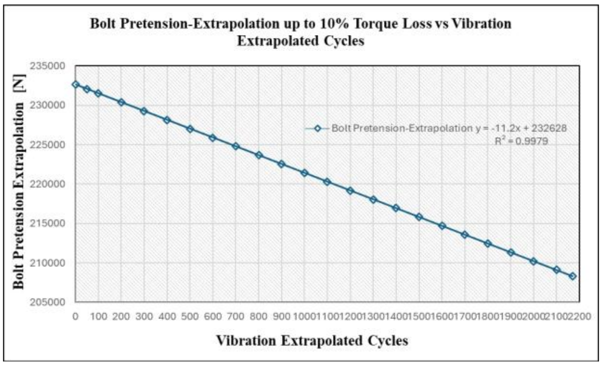


Figure 18: Extrapolated pretension drop up to the 10% torque loss threshold.

Table 5: Extrapolated Displacement Cycles until 10% Pretension Drop on the Bolt

| Parameter | Value |
|---|---|
| Bolt | M27 |
| Solving Steps | 380 |
| Solving Cycle | 95 |
| Displacements in Transverse Directions | from (0.5, 1.0) to (-0.5, -1.0) |
| Friction Coefficient | 0.1 |
| Initial Preload [N] | 231,481.5 (corr. to 1000 Nm torque) |
| Limit Preload (10% drop) [N] | 208,333.3 (corr. to 900 Nm torque) |
| Extrapolated (predicted) Cycle | 2170 |

### 3.3. Validation of numerical loosening trend

Although the present study focuses solely on numerical prediction, the obtained preload-loss trend was qualitatively compared with well-established experimental observations reported in classical loosening studies. Junker-type experiments consistently demonstrate an approximately linear preload decay during the early stages of microslip-driven loosening, followed by a nonlinear acceleration as the residual clamping force decreases. The numerical model employed in this work reproduces this characteristic early-stage behavior: within the simulated cycle range, the preload reduction exhibits a nearly linear trend, which is consistent with experimentally observed microslip-dominated loosening mechanisms. While direct quantitative validation is not possible in the absence of dedicated experimental testing, the agreement in trend with published experimental findings [4, 13] supports the physical relevance of the proposed FEA-based methodology. Experimental correlation is therefore identified as a key objective for future work to further assess and refine the predictive capability of the model.

This demonstrates that finite element analysis can be used to assess early-stage self-loosening trends of bolted joints under defined and controlled loading conditions. Although the numerical results are intentionally conservative, since real vehicles are subjected to variable rather than continuous maximum excitations, the proposed methodology offers a consistent framework for comparing loosening tendencies prior to experimental validation.

The preload decay presented reflects a deterministic trajectory. Variability in friction, seating, and load cycles introduce scatter that broadens the loosening window. Sensitivity runs ($\mu$ = 0.10–0.12) indicate variation in the onset of 10% preload loss, to be considered for re-torque intervals.

## 4. Conclusions

This study investigated the early-stage self-loosening behavior of a pre-tensioned, unsecured M27 U-bolted joint subjected to conservative cyclic transverse displacements using a finite element–based approach. A three-dimensional numerical model incorporating realistic pretension, thread engagement, frictional contact, and nonlinear contact mechanics was developed to capture preload degradation under multi-directional excitation representative of suspension-induced loading.

The results demonstrate that unsecured U-bolted joints are susceptible to progressive preload loss under lateral displacement amplitudes. Within the simulated cycle range, the preload reduction exhibits an approximately linear trend, which is consistent with the early phase microslip mechanism widely reported in experimental studies. Differences in loosening rate between the examined displacement amplitudes further indicate the strong influence of transverse excitation severity on preload degradation behavior. Although this study focuses on a 12.5-ton GAWR lift axle, the analysis process is platform-agnostic. By replacing displacement amplitudes with those derived from tandem, tridem, or tractor rear axles, all steps can be replicated without modification.

Unlike many previous investigations focusing on standard bolts or simplified plate joints, the present work specifically addresses unsecured U-bolt connections commonly used in heavy-duty suspension systems, thereby filling an important gap in existing literature. The proposed FEA-based framework enables a cycle-dependent comparison of loosening tendencies under controlled displacement conditions and provides mechanistic insight into the initiation of self-loosening at the joint interface.

The findings emphasize the value of secondary locking solutions for unsecured joints, including prevailing-torque nuts, double-nut systems, and serrated or conical washers. The modelling approach can be extended to compare these options numerically.

It should be noted that the cycle-based values obtained in this study are intended as indicative measures derived from short-range numerical trends rather than definitive service-life predictions. Several modeling assumptions, including simplified sinusoidal displacement waveforms, a constant friction coefficient, and the absence of experimental correlation, limit the direct extrapolation of the results to real operational conditions. These limitations are explicitly acknowledged and do not detract from the primary objective of the study, which is to establish a consistent numerical methodology for assessing early-stage self-loosening behavior.

Future work will focus on correlating the numerical predictions with experimental data, such as standardized loosening tests or vehicle-based measurements, and on extending the model to incorporate variable-amplitude loading, surface roughness effects, and more realistic vibration spectra. Such developments will further improve the applicability of the proposed approach for evaluating the durability and safety of bolted joints in automotive and heavy-duty engineering applications. The methodology will be extended to multi-bolt configurations and late-stage nonlinear preload decay, enabling comparative assessment of different locking strategies under identical loading conditions. Once correlated with road measurements, this approach has the potential to reduce the need for costly experimental setups and serve as a predictive guideline for re-torque intervals. Reported loosening cycles must not be interpreted as allowable service limits. Failure to re-torque may lead to critical loss of axle-to-suspension integrity, especially under sustained dynamic excitation.

### CRediT authorship contribution statement

The author confirms sole responsibility for the following: study conception and design, methodology development, finite element modelling, analysis and interpretation of results, and manuscript preparation.

### Declaration of competing interest

The author declares that there are no known competing financial interests, personal relationships, or other relationships that could have appeared to influence the work reported in this paper. The author confirms that the study was conducted with full academic independence and that the results and interpretations presented are based solely on scientific and engineering analysis.

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### Symbols

| | | |
|---|---|---|
| E | : | Young's modulus |
| $\nu$ | : | Poisson's ratio |
| $\sigma_\gamma$ | : | Yield strength |
| $\sigma_u$ | : | Ultimate strength |
| $\mu$ | : | Friction coefficient |
| GAWR | : | Gross Axle Weight Ratings |
| GVWR | : | Gross Vehicle Weight Ratings |
| T | : | Applied Torque |
| D | : | Nominal Bolt Diameter |
| $\kappa$ | : | Nut Factor |

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