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GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

**INVESTIGATION OF FASTENER ELEMENTS
BEHAVIOURS UNDER DYNAMIC LOADINGS**

by

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İZMİR

INVESTIGATION OF FASTENER ELEMENTS BEHAVIOURS UNDER DYNAMIC LOADINGS

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INVESTIGATION OF FASTENER ELEMENTS BEHAVIOURS UNDER DYNAMIC LOADINGS

ABSTRACT

This work aims to improve the understanding of bolt behaviors in dynamic load scenarios, particularly those influenced by transverse loads. In order to forecast fastener outcomes under varying load situations, a comprehensive investigation was conducted encompassing experimental, theoretical, machine learning, and simulation-based study methodologies. Findings indicate that bolts subjected to vibratory loosening are more susceptible to experiencing increased friction in their bearing and thread sections. This is primarily caused by wear in the contacting regions resulting from both macro and micro slips. Therefore, the capacity to reuse fasteners becomes uncertain in certain situations. Experimental results have shown that the type of coating used also has a significant effect on the behaviors observed. A simulation model was developed to address the challenges of conducting repeated experimental testing, specifically focusing on the factors of tightening, friction change, and loosening. The simulation approach enables researchers to predict the effects of fastener loosening risks. The solve time for the loosening simulation might be significantly increased depending on the thread mesh density. A neural network-based deep learning technique and model were created to address this problem. Model was trained using both experimental data and simulated outcomes. To manage the function errors, the Mean Squared Error (MSE), Mean Squared Deviation (MSA), and R-squared values are used. The model loop built in this context is designed to optimize the parameters during the training process and identify the most favorable scenarios. This enables us to input experimental and simulation data into the neural network in the future, hence enhancing the program's dependability without making any modifications to the underlying code. The last part of the study primarily examines the theoretical foundation of bolt stresses and the lifespan of loosening effected bolts. Calculations were conducted using cantilever beam theories, which is a widely used method for analyzing load instances in bolted joints. A methodology has been built to forecast the fatigue life of the bolt based on its transversal fatigue life when subjected to repetitive tightening. This methodology utilizes cumulative damage theories.

Results obtained from the experimental investigation and theoretical assumptions fall within an acceptable margin of error. This theory allowed for the transformation of transversal displacement inputs into lateral shear and force values. These values were then used to make predictions about the lifespan of structures under loosening conditions. This approach also incorporates the computation of bearing friction affects and load transfer capabilities. An increase in the shear load capacity results in an increase in the bending moment while maintaining the same transverse displacement. This increase in bending moment leads to a decrease in the fatigue life of the bolt. This is also consistent with the findings of the friction change test conducted during repeated tightening Junker vibration testing, where samples fail due to bending fatigue. It is crucial to consider the reusability of the fastener when analyzing high transverse loads so as to improve the precision of its estimated lifespan.

Keywords: Loosening, bolt fatigue, life prediction, fastener elements, dynamic loading

DİNAMİK YÜKLER ALTINDA BAĞLANTI ELEMANLARININ DAVRANIŞLARININ İNCELENMESİ

ÖZ

Bu çalışma, enine yüklerden etkilenen dinamik yük senaryolarındaki cıvata davranışlarını incelemeyi ve geliştirmeyi amaçlamaktadır. Değişken yük durumları altında, bağlantı elemanının davranışlarını tahmin etmek amacıyla deneysel, teorik, makine öğrenimi ve benzetim tabanlı çalışma metodolojilerini kapsayan detaylı bir araştırma yürütülmüştür. Bulgular, titreşimle gevşemeye maruz kalan cıvataların kafa altı ve dış bölgelerinde, sürtünme katsayılarında artış yaşandığını göstermektedir. Bunun temel nedeni, temas eden bölgelerdeki makro ve mikro kaymalardan kaynaklanan aşınmadır. Bu sebeple, bağlantı elemanlarının tekrar kullanılması beklenilmeyen sonuçların dogmasına sebebiyet verebilmektedir. Deneysel sonuçlar, kullanılan kaplama tipinin, bahsi geçen sürtünme davranışı üzerinde önemli bir etkiye sahip olduğunu göstermiştir. Bağlantı elemanlarının yanal yükler altındaki sürtünme ve gevşeme davranışlarını incelemek üzere deneysel testler gerçekleştirilmiş ve sonuçları irdelenmiştir. Deneysel test kabiliyetinin son kullanıcıda sınırlı ve test yapmanın maliyetli olmasından dolayı, deneysel yöntem sonucu ile doğrulanmış bir benzetim modeli ihtiyaç duyulduğu ortaya çıkmıştır. Simülasyon yaklaşımı, araştırmacıların bağlantı elemanı gevşeme risklerinin etkilerini test gereksinimi olmadan tahmin etmeyi amaçlamaktadır. Gevşeme analizi her ne kadar güvenilir ve kapsamlı bir model olsa da zaman alması ve modelin zorluğuna bağlı olarak çözüm süresi uzun zaman alabilmektedir. Günümüzün gelişen teknolojisi ile birlikte, karmaşık yaklaşımların ve tahmin metodolojilerinin yapay sinir ağları sayesinde nispeten daha kolay indirgenebileceği bilinmektedir. Bu kapsamda yapay sinir ağı tabanlı bir derin öğrenme tekniği ve modeli geliştirilmiştir. Model hem deneysel veriler hem de benzetim sonucunda elde edilmiş verileri kullanılarak eğitilerek tahmin yapacak düzeye getirilmiştir. Fonksiyon hatalarını yönetmek ve modelin doğruluğunu test etmek amacıyla ortalama karesel hata (MSE), ortalama karesel sapma (MSA) ve R-kare değerleri kullanılmıştır. Bu bağlamda oluşturulan model döngüsü, eğitim sürecinde parametrelerin optimize edilmesi ve en uygun senaryoların belirlenmesi amacıyla yönelik olarak tasarlanmıştır dinamik bir altyapı ile hazırlanmıştır.

Geliştirilen bu alt yapı, gelecekte veri sayısının artırılması için deneysel ve benzetim sonuçlardan elde edilen verilerin modelin iyileştirilmesi için kullanılabilmesine olanak sağlamaktadır. Çalışmanın son bölümünde, öncelikle cıvata gerilmelerinin teorik temelleri ve gevşemeden etkilenen cıvataların ömrü incelenmiştir. Hesaplamalar, cıvatalı bağlantılardaki yük örneklerini analiz etmek için yaygın olarak kullanılan bir yöntem olan ankastre kiriş teorilerini kullanarak yapılmıştır. Akabinde, cıvataların tekrarlayan sıkıya maruz kaldığında, enine yorulma ömrüne dayalı olarak yorulma ömrünü tahmin etmek için bir metodoloji oluşturulmuştur. Bu metodoloji, kümülatif hasar teorisini kullanarak tekrar sıkım altındaki hasar değerlerini hesaplayan bir kümülatif omur tahmin yaklaşımının uygulanabilirliğini ortaya çıkarmaktadır. Bu yöntem ile enine yer değiştirmeye maruz kalan cıvataların, yanal kesme ve oluşan eğilme momentlerini de işin içine katarak, gevşeme verileri ile cıvatanın gerçekçi omur tahminlerinin yapılmasına olanak sağlamaktadır. Elde edilen hesaplamalar ve deneysel sonuçlar, kesme yükünün veya yanal deplasmanın artması ve cıvatanın sürtünme katsayısındaki artışlar ile yanal yük taşıma kabiliyetinin artması sebebi ile cıvatanın yorulma ömründe düşüş olduğunu göstermektedir. Bu nedenle, tahmini ömrünün hassasiyetini artırmak amacıyla yüksek enine yükleri analiz ederken bağlantı elemanının yeniden kullanılabilirliğini dikkate almak çok önemlidir.

Anahtar kelimeler: Gevşeme, cıvata yorulması, ömür tahmini, bağlantı elemanları, dinamik yükleme

CONTENTS

	Page
Ph.D. THESIS EXAMINATION RESULT FORM	ii
ABSTRACT	ii
ÖZ.....	vi
CONTENTS	viii
LIST OF FIGURES.....	x
LIST OF TABLES	xv
 CHAPTER ONE INTRODUCTION.....	 16
1.1 Scope of The Study	16
 CHAPTER TWO - LITERATURE REVIEW	 17
2.1 Literature Review	17
2.1.1 Bolted Joints	17
2.1.2 Bolt Loosening	19
2.1.3 Bolt Fatigue	28
2.1.4 Friction Coefficient	30
2.1.5 FEA Studies For Bolted Joints	34
 CHAPTER THREE - MATERIAL AND METHOD	 38
3.1 Methodology of the Research.....	38
3.2 Experimental Method	39
3.2.1 Friction Coefficient Test.....	39
3.2.2 Junker Vibration Test	43
3.2.3 Axial Fatigue Test	46
3.3 Finite Element Analysis	47
3.4 Neural Network Based Loosening Rate Prediction Method	48
 CHAPTER FOUR - RESULTS AND DISCUSSION.....	 52

4.1 Development of Friction Change Model under Non-Dynamic and Dynamic Loading Cases	52
4.2 Prediction of Loosening Rates Under Transversal Dynamic Loads	60
4.3 Axial Fatigue Test Results.....	66
4.4 Simulation Based Tightening Model Including Friction Changes	70
4.4.1 Model Development with Boundary Conditions.....	72
4.4.2 Friction Model	75
4.4.3 Tightening Simulation Outcomes.....	82
4.5 Simulation Based Vibration Model Including Friction Changes	86
4.6 Loosening Rate Prediction Model with Neural Networks.....	92
4.7 Loosening and Fatigue Effected Prediction Model with Reusability.....	99
4.8 Conclusion	117

LIST OF FIGURES

	Page
Figure 2.1 Axially loaded and shear loaded joint representation (Bickford, 2018)...	17
Figure 2.2 Input torque distribution percentage in order to obtain preload.	18
Figure 2.3 Junker Vibration Test Device (Gong, Ding, et al., 2022).....	21
Figure 2.4 Schematic representation of acting forces due to shear loads (Yamamoto et al., 1977).	24
Figure 2.5 Images of commonly used anti-loosening members (Gong et al., 2022)..	26
Figure 2.6 Comparison of Junker test result of different hardness and tightening strategies with plain and ripped nut (Tanrikulu et al., 2020).	27
Figure 2.7 Junker vibration test results of different nut mechanism (Ince et al., 2017).	27
Figure 2.8 Bolted joint diagram dynamically loaded in axial direction.....	28
Figure 2.9 Effective bearing diameter representation.	32
Figure 2.10 Friction variation in repeated usage of electroplated zinc fasteners without lubricating elements (Eccles et al., 2010).	33
Figure 2.11 Thread strip and bolt shank elongation modeling with suitable material model et plastic region (Hu et al., 2016).	36
Figure 2.12 Finite element model boundary conditions (Pai & Hess, 2002).....	37
Figure 3.1 Friction, fatigue and vibrational loosening model development steps	39
Figure 3.2 BLM friction test device	40
Figure 3.3 M8x1.25 bolts with Zinc Flake coating type on the top and Zinc Cr+3 on the bottom.....	41

Figure 3.4 Bolt installation in Junker vibration test device with Atlas Copco ST Wrench	43
Figure 3.5 Junker vibration test device	44
Figure 3.6 Junker vibration test device, cross section.....	44
Figure 3.7 An example of the preload-cycle graph for loosening rate calculation	45
Figure 3.8 Axial fatigue test device and it's sketch with components.....	46
Figure 3.9 Thread cut theoretical dimensions for M8x1.25 6g thread profile	48
Figure 3.10 Finale thread shape after thread cut and bolt nut assembly.....	48
Figure 3.11 Main structure code of the machine learning model	49
Figure 3.12 Code of retraining the prediction model with the best parameters.....	50
Figure 3.13 Axial fatigue test device and its sketch with components	51
Figure 4.1 Frictional change of Zinc Cr^{+3} under repeated tightening without dynamic effects.....	53
Figure 4.2 Frictional change of Zinc Flake coatings under repeated tightening without dynamic effects.....	54
Figure 4.3 Frictional change comparison of both coating types.	55
Figure 4.4 Frictional change of vibrationally loosened Zn Cr^{+3} under repeated tightening.....	56
Figure 4.5 Frictional change of vibrationally loosened zinc flake under repeated tightening.....	57
Figure 4.6 Frictional change comparison of both coating types under vibrational loosening.....	58
Figure 4.7 Frictional change of ZnCr^{+3} under dynamic loosening and manual loosening	59

Figure 4.8 Frictional change of Zinc flake under dynamic loosening and manual loosening.....	59
Figure 4.9 Worn threads due to multiple reuse	60
Figure 4.10 Clamp load cycle graph of M8 25mm clamp length bolt under 15kN ...	61
Figure 4.11 Clamp load cycle graph of M8 40mm clamp length bolt under 15kN ...	62
Figure 4.12 Clamp load cycle graph of M8 25mm clamp length bolt under 10kN ...	62
Figure 4.13 Clamp load cycle graph of M8 40mm clamp length bolt under 10kN ...	63
Figure 4.14 Clamp load cycle graph of 4 different combination	64
Figure 4.15 Wohler diagram for 15 kN mean load for M8 8.8 fastener	67
Figure 4.16 Wohler curve of M8 fastener experimentally and theoretically.	70
Figure 4.17 Mesh model of the bolt, plate and nut members.....	72
Figure 4.18 Boundary conditions for the tightening model	74
Figure 4.19 Spring model inputs for joint stiffness	75
Figure 4.20 Effective contact diameters and contact areas representation	76
Figure 4.21 Variation of under-head contact surface pressures along the y-axis (a) and representation of effective diameters according to 2 different methods (b)	77
Figure 4.22 Friction changes under repeated tightening.....	79
Figure 4.23 Cumulative sliding distance versus friction coefficient under repeated tightening.....	80
Figure 4.24 Friction coefficient change based on relative sliding distance during tightening, thread (a), under head (b)	81
Figure 4.25 Torque angle values at different spring constants	82
Figure 4.26 Sliding distance-based friction model APDL code.	83

Figure 4.27 Simulation output of axial force-torque change based on the number of repeated tightening	84
Figure 4.28 Von-Mises equivalent stress distribution	84
Figure 4.29 Sliding distance under bolt head and nut thread surface, 2 nd tightening.	85
Figure 4.30 Transvers loading condition in terms of displacement in Y axis.....	86
Figure 4.31 Remote displacement application on the moving plate	87
Figure 4.32 Load case based on main steps and sub steps in ANSYS based on the input values.	88
Figure 4.33 Equivalent von-Mises stress distribution along the bolt under transvers displacement on the bottom and in nominal state at the top.....	90
Figure 4.34 Pressure distribution on the bearing surface with transvers displacement	91
Figure 4.35 Pressure distribution on the thread surface with transvers displacement	91
Figure 4.36 Loosening curve comparison of experimental and simulation result.....	91
Figure 4.37 Loosening curve output of a simulation model.....	91
Figure 4.38 Main optimization function for the best model investigation.....	96
Figure 4.39 Main prediction model for loosening rate outcomes.	96
Figure 4.40 Predicted versus actual plot	97
Figure 4.41 Loosening rate values of predicted versus actual data.....	97
Figure 4.42 Performance evaluation of the trained model	98
Figure 4.43 Input values for loosening curve prediction	98
Figure 4.44 Predicted loosening curve	99
Figure 4.45 Junker Vibration test result for 12.1 kN with repeated tightening	102

Figure 4.46 Junker Vibration test result for 17.1 kN with repeated tightening	103
Figure 4.47 $F_{\max}$ - N_i curve at lateral loading condition with restricted loosening	105
Figure 4.48 Bolt failure due to fatigue during Junker (a), bolt without experiencing any damage at the end of the test (b)	107
Figure 4.49 Accumulated damage per load case versus loosening rate for 17.1 kN	108
Figure 4.50 Accumulated damage per load case versus loosening rate for 15.6 kN	108
Figure 4.51 Load and moment diagram of a bolt loaded in transvers direction	110
Figure 4.52 Schematic representation of bolt elongation (Welch, 2018)	111
Figure 4.53 Length versus cycle graph for theoretical and experimental results.....	116

LIST OF TABLES

	Page
Table 3.1 Coating types.....	41
Table 4.1 Loosening rate data and parameters obtained from Junker vibration test..	64
Table 4.2 Experimental fatigue test Basquin equations and their long-life fatigue limits.	69
Table 4.3 Experimental friction coefficient test results, mean of 5 test.....	78
Table 4.4 Error percentage values.....	85
Table 4.5 Load conditions based on time steps.....	88
Table 4.6 Loosening rates for various load cases and repeated usages.	104
Table 4.7 Cumulative fatigue damage prediction with repeated tightening	105
Table 4.8 Shows the calculated values based on the equation given in the section for bolt stress calculation.	115

CHAPTER ONE

INTRODUCTION

1.1 Scope of the Study

Threaded fasteners are widely recognized as essential components in multiple industries, owing to their extensive utilization as fastening elements. In the areas of engineering and construction, the use of fasteners plays a crucial role in ensuring the proper functioning and integrity of various structures. They also serve as essential components, facilitating the secure attachment of two or more elements, thereby enabling the structure to perform as intended. The utilization of fasteners in different fields is primarily motivated by the convenience and reusability offered by these components. Researches in the field aimed to comprehensively investigate the behavior of fasteners in order to enhance their reliability and predictive capabilities in application conditions. The occurrence of failures associated with bolted fasteners has prompted a research emphasis on investigating these specific failures in order to provide a theoretical foundation for preventing such failures, some of which have had tragic consequences. With the technological development research in this field gained speed and more understanding of the phenomena behind these failures.

Present research aims to focus on the dynamical behaviors of fastener where loosening is present and in which retightening is applicable. In order to have an understanding of the behaviors of fasteners which are reused multiple of times, friction coefficient changes are investigated and modeled for various conditions. Similarly, loosening mechanism of fastener in which are affected by transversal loads are investigated in detail with experimental and FEA methods. A new prediction methodology for loosening rates are introduced with using machine learning algorithms. The study also considered the fatigue response when transverse loads and loosening are present. The investigation resulted in the development of a fatigue life prediction method that incorporates the effects of loosening and reusability. Outcome of this research aims to deliver a comprehensive methodology to engineers in the field to predict the life cycle of their fastener members with including various influencing factors.

CHAPTER TWO

LITERATURE REVIEW

2.1 Literature Review

2.1.1 Bolted Joints

Bolts are fastening elements widely used in numerous application due to their easy nature and cost-effective applications. Nearly all the industries have embraced bolted joints with defining specific guidelines depending on the application criticality. Bolted joints can be classified into two distinct categories based on the orientation of the external loads or forces exerted on the joint. In the event that the forces imposed on the joint align closely with the axes of the bolt, the joint is classified as being subjected to tension and is commonly referred to as a tension or tensile joint. In the event that the load's line of action is approximately perpendicular to the axes of the bolt, it can be inferred that the joint is subjected to shear loading, thereby classifying it as a shear joint, a representation of the acting force directions are given in Figure 2.1 (Bickford, 2018).

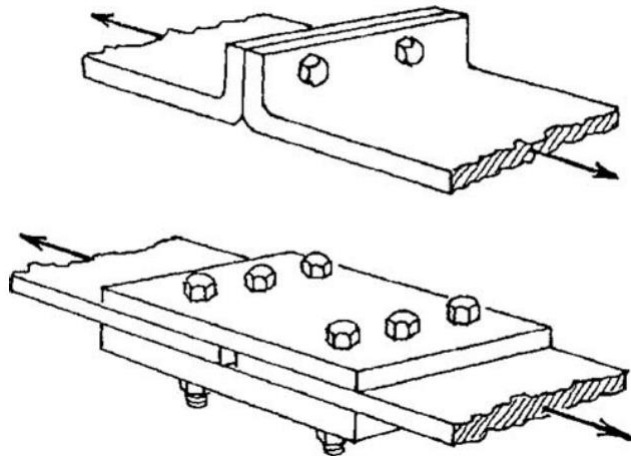


Figure 2.1 Axially loaded and shear loaded joint representation (Bickford, 2018)

The primary objective of using a bolt or an assembly of bolts within both tensile and, in most cases, shear joints is to establish a clamping force between two or more

entities. In certain shear joints, the bolts serve primarily as shear pins with the help of its shank, although with a degree of bolt tension and clamping force that is advantageous for the purpose of nut retention, among other potential benefits. As previously mentioned, the primary objective of the bolt is to securely fasten multiple components together. The application of torque to the nut or bolt induces a torsional deformation in the bolt, resulting in the generation of an axial force that facilitates the establishment of a clamping mechanism. The force applied to secure the connection between two components, commonly known as clamping force, is referred to as pretension or bolt preload in technical literature. Due to the clamping of the members, the resulting clamping force generates tensile stress in the bolt, which in turn leads to compressive stress in the clamped members.

The experimental findings with measuring input and output torques have revealed that less than 10% of the torque applied is effectively converted into clamping force. The remaining portion of energy is taken with the force due to friction energy. Approximately 40% of the total torque is expended in the process of counteracting the friction present in the threaded mating surface of the bolt and nut, while an additional 50% is allocated towards mitigating the friction within the screw bearing surface as shown in Figure 2.2 (Bickford, 2018). Thus, the importance of frictional properties of a threaded member becomes one of the greatest parameters.

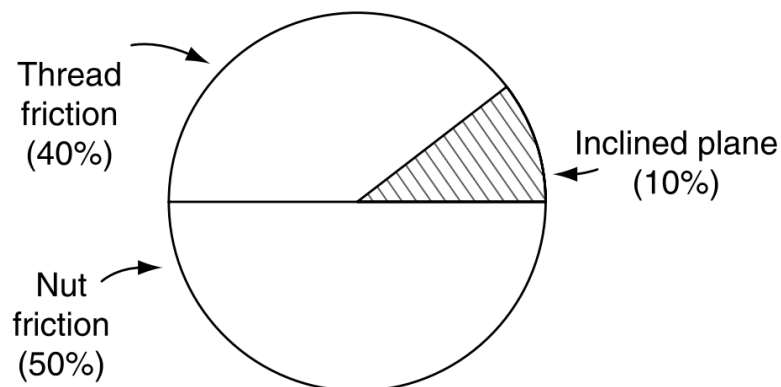


Figure 2.2 Input torque distribution percentage in order to obtain preload.

When bolts are tightened properly, both the bolt and joint members undergo elastic deformation. In essence, they function like to stiff springs. spring-like characteristics significantly effects the dynamics of the joints. Thus, determination of their spring behaviors plays a crucial role in bolted assemblies. In a bolted assembly, stiffness can be separated in two division, these are bolt stiffness K_b and member stiffness K_m . Stiffness of a bolt refers to its ability to resist deformation under the influence of an externally applied force while stiffness of a member is the ability to resist deformation under compressive acting loads. Stiffness constant of a bolt behaves like two springs connected in series, if the bolt is partially threaded. Thus, the equation to calculate bolt stiffness becomes as given in Equation 2.1. Where A_d is the major diameter area of the bolt shank, A_t stress area of the threaded portion, l_t length of the threaded portion and l_d length of the unthreaded shank.

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} \quad (2.1)$$

The grip of the fastener has the potential to include a number of members more than two. In overall, these components function as a series of compressive springs, resulting in the cumulative spring rate of the components. Thus, the general equation of multi stack joint member can be found as given in Equation 2.2 if the bolt frustum angle is taken as 30 °.

$$k = \frac{0.5774\pi E d}{\ln \frac{(1.155t+D-d)(D+d)}{(1.155t+D+d)(D-d)}} \quad (2.2)$$

2.1.2 Bolt Loosening

2.1.2.1 Junker Vibration Theory

First investigation into the phenomenon of loosening in threaded fasteners goes back to the work of Goodier and Sweeney (Goodier & Sweeney, 1945). They observed

that the loosening of threaded fasteners is mostly attributed to the cyclic loading and unloading experienced by the fasteners in lateral direction, leading to a slight rotation in the nut. Later on investigations in order to understand this phenomena gained more attention particularly focusing in lateral cyclic movements. In the 1969, Junker introduced a study that demonstrated the importance of transverse or shear loading as the most critical form of loading in the context of vibration-induced loosening with developing a test bench which was called later Junker vibration test device (Junker, 1969). The study focused on examining the influential factors that contribute to the loss of preload in a bolted joint subjected to a transversely loaded mechanism. Junker vibration theory and test set up, introduced by Gerhard Junker, is now recognized as the present methodology for conducting dynamic evaluations of fasteners and examining their tendency for self-loosening. The device utilized in this work later became as a widely adopted test system and methodology for examining and exploring the phenomenon of loosening in threaded components under lateral loads.

Junker test apparatus consists of two plates, one fixed and one moving, which are positioned perpendicular to the axis of the bolt and have relative motion with respect to each other. The utilization of a cam mechanism or servo hydraulic systems enables an adjustment of displacement or lateral force values, hence facilitating the movement of the top plate along a lateral axis and inducing vibrations in the desired direction. The lateral displacements resulting from this movement induce micro and macro slips in the bearing and thread contact areas of the threaded members, leading to bolt rotation in the direction of loosening. The clamp load acquired in the joint reduces progressively as a result of rotation, in accordance with the cycle time. Representation of the Junker vibration test device used to evaluate clamp load losses based on lateral loads is given in Figure 2.3.

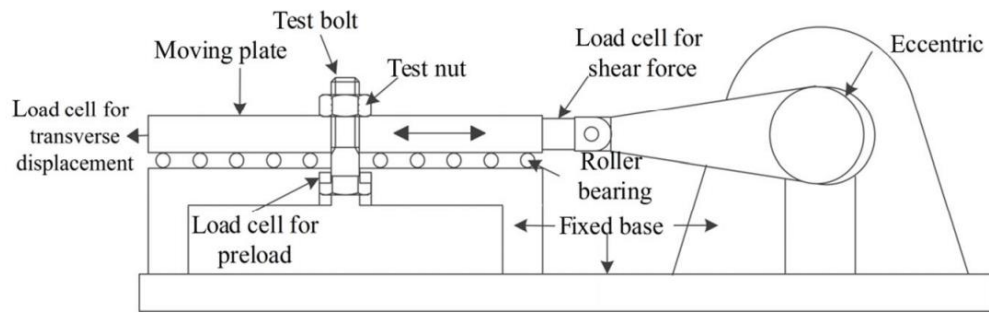


Figure 2.3 Junker Vibration Test Device (Gong, Ding, et al., 2022)

2.1.2.2 Parameters Affecting Loosening

Multiple studies have been carried out in the past decade to gain a better understanding of the behaviors exhibited by bolted joints under vibrating conditions. Investigation in that area, revealed that friction coefficient in the mating surfaces plays a crucial role in terms of loosening speed. One of the investigations examined a range of parameters that could have an effect on loosening behaviors. These factors include preload, frictional coefficients in mating surfaces like thread and bearing and clamp length. Findings obtained from the study revealed that parameters like preload, friction coefficients were the highest effecting factors in terms of loosening effect (Gong et al., 2019). Another study which focused on statistical investigation on the effecting parameters focused on preload, fastener elasticity, hole clearance, major diameter of the bolt and pitch. Findings based on experimental and statistical evaluation showed that preload and bolt elasticity was the two main crucial and effectful parameters. The research also suggests that in order to prevent fastener loosening, it is advisable to employ certain optimal circumstances, including a high level of preload, a low modulus of elasticity, a big diameter (Sanclemente & Hess, 2007). Study carried out by Housari & Nassar (2007) focused on the effects of thread and bearing friction coefficient values. In order to evaluate the outcome a mathematical model was introduced with taking experimental results as a baseline. Obtained results showed that both bearing and thread friction coefficients has a crucial impact on the loosening rate values. Thus, an attention was taken to the importance of using a suitable lubrication on threaded members. A research study was also conducted to examine the impact of clamped length and loading direction on the phenomenon of self-loosening in bolted

joints. A specially designed test rig was used to adjust the load axis. During the experimental studies, clamp length and relative displacement amount of the moving plate were the key parameters which was investigated. The study revealed that increasing the clamped length has the potential to improve the endurance limits of self-loosening (Zhang et al., 2006). It was also found that transvers displacement amplitude has a crucial effect in term of bolt loosening performance. Study in this domain revealed that with the increase of the transvers cyclic amplitude, preload decrease rate also increases (Chen et al., 2020).

2.1.2.3 *Loosening Theories*

The application of external forces in the form of small transversal vibrations to the clamped bolted structures results bending moment on the bolt axis. With the increase of the transvers amplitude of these vibrations, there will be a corresponding increase in the rate of bending, leading to the slipping of the bolt head and threads in the matting surfaces. The phenomenon starts to counteract the frictional processes induced by the moment's influence. Friction is a force that opposes the relative motion or tendency of motion between two surfaces in the phenomenon in which the mechanisms become dysfunctional due to excessive bending over a period of time is referred to as critical slip. Two distinct hypotheses, namely full slippage and local slippage accumulation theories, have been put forward to better understand the rotational loosening phenomenon resulting from transverse vibrations, based on variations in slippage modes. According to Junker theory, the entirety of the internal thread of a nut is considered as the block, whereas the entire exterior thread of the bolt is considered as the inclined surface. Hence, according to the classical theory, rotational loosening is observed when the frictional forces acting on a threaded surface are entirely surpassed, a phenomenon referred to as the complete slippage hypothesis. This theory is widely accepted in the industry and academia due to its applicability and testing methodology. Investigations in this field were mainly focusing on the matting surface contact friction coefficient values based on the Coulomb friction law practices.

Following studies based on Junker theory was to investigate and formulate loosening phenomena. In this context, two step loosening theory was introduced by Yamamoto and Kasei (1977). In the preliminary stage of the experiment, a deliberate

inclination was introduced to the alignment of a bolt thread in relation to its corresponding nut thread, leading to the occurrence of slippage along the surface of the nut thread. The occurrence of slippage resulted in the initiation of elastic torsion in the bolt shank. In the following phase of the experiment, it was observed that the bearing surface of the nut exhibited slippage on the surface of the vibrating plate. The occurrence of slippage was accompanied with the loosening of the nut's rotation, due to a phenomenon that was initiated by the elastic torsion spring back of the bolt shank.

In 1984, Yamamoto and Kasei (1984) conducted a comprehensive analysis on the fundamental of rotational loosening with taking their previous studies. Study encompasses an analysis that focuses on the investigation of two distinct types of slippages: straight slippage occurring on a threaded surface, and arc slippage occurring on a bearing surface. Moreover, a crucial precondition for the phenomenon of rotational loosening was found to be the presence of bearing surface slippage. In their works, they proposed the concept of the critical relative slippage (S_{cr}) as a crucial parameter in assessing the threshold for the onset of loosening through transverse displacement. This parameter serves as a determinant for the upper limit of transverse displacement that can lead to the initiation of loosening. Theoretical critical slippage equation proposed in this study is given in Equation 2.3. Where δ is the critical slip distance, F the clamp force, μ_b Bearing friction coefficient, M_t resultant moment in the threaded portion, E and I are the elastic modulus of the bolt and moment of inertia, l_n clamp length. Representation of the acting force and abbreviations are given in Figure 2.4.

$$S_{cr} = 2\delta = \frac{2l_n F \mu_b (l_n^2 + 3k_w l_n EI)}{3EI} - \frac{3l_n M_t (l_n + 2k_w EI)}{3EI} \quad (2.3)$$

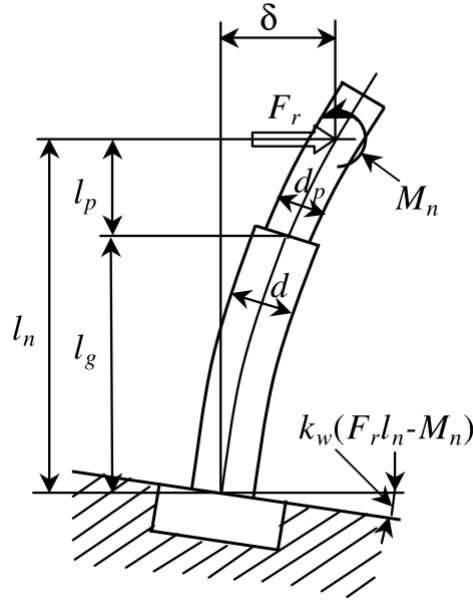


Figure 2.4 Schematic representation of acting forces due to shear loads (Yamamoto et al., 1977)

In the following studies to improve equation proposed, various experimental and theoretical studies followed the lead of Yamamoto and Kasei. The occurrence of relative slip on the nut bearing surface is dependent upon the shear loading F_r reaching the maximum static friction force μF_s . Thus, the equation is later redefined with including thread angle, α in the contacting members and nut height, m (Izumi et al., 2005). Updated equation is given in Equation 2.4.

$$S_{cr} = 2F_s \left\{ \mu_w \left(\frac{l_g^3}{3EI_g} + \frac{l_p^3}{3EI_p} + \frac{l_g l_p l_n}{EI_g} + k_w l_n^2 \right) - \frac{m}{4} \frac{\mu_s}{\cos^2 \alpha} \left(\frac{l_g^2}{2EI_g} + \frac{l_p^2}{2EI_p} + \frac{l_g l_p}{EI_g} + k_w l_n \right) \right\} \quad (2.4)$$

An important theoretically derived criteria was proposed for the purpose of preventing the phenomenon of self-loosening in preloaded threaded fasteners when subjected to cyclic transvers displacements (X. Yang et al., 2010). Proposed criteria aimed to predict the minimum preload limits required for a bolt to prevent loosening under the influence of externally applied cyclic transvers displacements. During development of the model, several key aspects were taken into account. It was

experimentally and theoretically proven that, required clamp load threshold increases with the amplitudes acting in the lateral direction. On the other hand, friction coefficient values in both thread and bearing area has a reverse effect. With the increase of the friction coefficient values, the thresholds start to decrease. Lastly, it was also found that bolt stiffness has a crucial effect on the preload threshold. Thus, a mathematical equation was proposed and is given in Equation 2.5. In which, R_b is transverse bearing friction shear force ratio.

$$R_b = \frac{3EI\delta_0}{\mu_b F_b k L^3} \leq \frac{1}{A_b} f_{BTF}^{-1} \left(\frac{p A_b}{2\pi \mu_b} \right) \quad (2.5)$$

2.1.2.4 Loosening Preventing Designs and Methods

In the years that followed after the introduction of the loosening phenomenon, researchers in this field have directed their attention towards the prevention of loosening through the utilization of various methods and designs. In situations where the adjustment of joint parameters proves ineffective in mitigating the negative consequences of loosening, alternative measures have been implemented. Specifically, the utilization of specially designed threaded members and washer elements has been introduced for preventing this issue. Figure 2.5 shows the most commonly used members in order to slow down loosening effect in structures. It can be seen that there are mainly two categories, these are washer elements and nut members. All of these special elements aim to restrict or limits the rotation of the threaded members in order to prevent vibrational loosening related clamp losses. It can be seen from the literature studies carried out in this field, that elements in the form of a nut, which increase the frictional contact force at the thread surface, are significantly more effective at preventing loosening than those in the form of a washer, which increase the frictional force at the bearing surface (Gong, Liu, et al., 2022; Ince et al., 2017; Izumi et al., 2009; R. Kumar, 2013; Sase et al., 1996).



Figure 2.5 Images of commonly used anti-loosening members (Gong et al., 2022)

It is crucial that each of these specific members has its own special criteria in order to perform as predicted. That is why these features should be selected carefully in order to expect the best outcome from these members. For example, performance of a serrated nut was investigated specifically based on the counter plate hardness and tightening strategy. It was found that, hardness of the member and tightening strategy played a key role in terms of vibrational loosening performance. It can be seen from Figure 2.6 that; loosening rate is not affected in plain nut while the effect of hardness and tightening starts to be more crucial with the serration nut usages. In order to obtain the best performance, it was suggested to tighten the bolt and hold the nut still in a relatively soft material compared to the nut hardness (Tanrikulu et al., 2020).

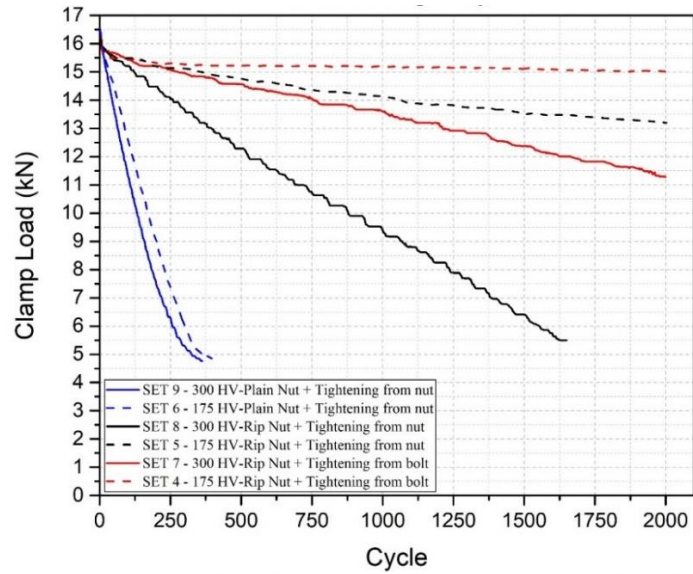


Figure 2.6 Comparison of Junker test result of different hardness and tightening strategies with plain and ripped nut (Tanrikulu et al., 2020)

Additionally, it was discovered that regardless of the utilization of various materials such as stainless steel, the axisymmetric double nut maintained its superior performance compared to the double nut and prevailing nut (Ince et al., 2017). The Junker vibration test result are given in Figure 2.7.

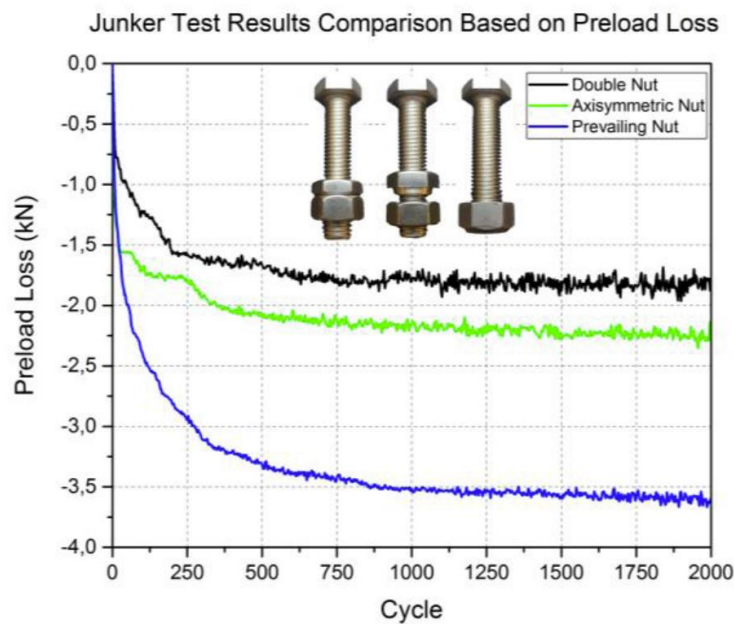


Figure 2.7 Junker vibration test results of different nut mechanism (Ince et al., 2017)

2.1.3 Bolt Fatigue

Bolt fatigue is a form of structural failure that arises when a bolt is exposed to repetitive stress cycles. The potential consequence of this phenomenon is the initiation of a crack and further fracture in the bolt, regardless of the fact that the applied stress remains far lower than the yield strength of the bolt. That is why understanding the effect of dynamically loaded just are crucial in order to maintain structural stability and avoid catastrophic failures due to fatigue. Loading direction of the dynamic cycle load respective to the bolt axis defines the concept of the prediction methodologies. That is why in the further section's details of the axially and transversely loaded dynamical joints will be investigated.

Bolts are structural components that are subjected to tensile loading with their first installation. The magnitude of the tension force in acting systems is dependent upon several factors, including the input torque, joint stiffness, friction coefficient values, clamp length, and several other characteristics. That is why understanding these effects are crucial to predict bolt fatigue life under tensile loadings. Bolts loaded in tensions carry certain amount of the externally applied load based on their stiffness ratios. Representation of a joint loaded in tension and its resultant dynamic load at bolt member is show in Figure 2.8. Where F_{amp} , is the load amplitude acting to the bolt, F_a externally applied dynamic load, F_{pa} load carried by the member.

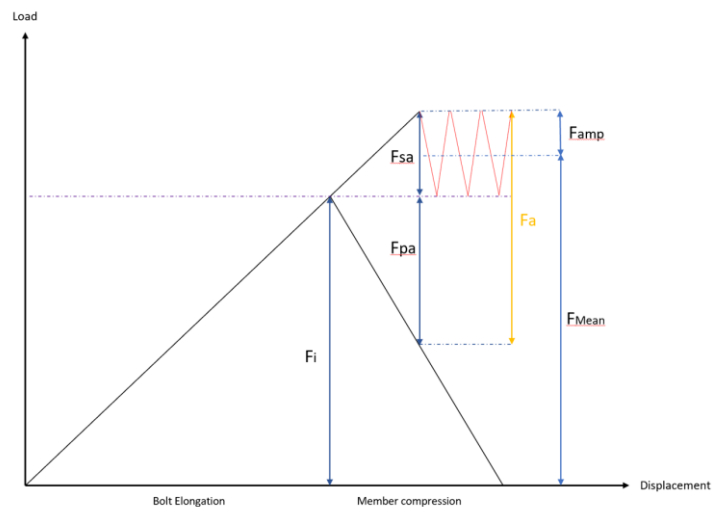


Figure 2.8 Bolted joint diagram dynamically loaded in axial direction

In order to have a baseline in terms of test systematic, ISO (International Organization for Standardization) introduced a standard which then updated to its latest version in 1993 (ISO, 1993). The present standard establishes a systematic approach and a comprehensive testing framework for assessing the fatigue endurance of bolts subjected to axial loading. The primary emphasis of researches in this field has centered on the utilization of a specific testing methodology to assess the variables affecting the fatigue life performance of bolts.

Bolt manufacturing for threading operation is generally happening with thread rolling operation due to its easiness and low cost operations for mass scales. A study focused on the aviation fasteners, investigated the effect of thread operations in terms of bolt fatigue life. For this purpose, bolts threads were obtained by machining, thread rolling before heat treatment and thread rolling after heat treatment. The investigation showed that the mechanical characteristics demonstrated a generally consistent behavior, however the fatigue strength was significantly influenced by the chosen production methods. The study findings indicated that completing the thread rolling operation following to the heat treatment operation is recommended for achieving optimal fatigue performance in a fastener (Ifergane et al., 2001). Research undertaken within this domain has demonstrated that thread rolling has a significant role in determining the fatigue life of bolts. The application of thread rolling after the heat treatment process yields significant effects as a result of the compressive residual stresses that are induced in the thread root. These stresses contribute to an increase in endurance, resulting in an estimated rise of nine percent in load ratio values within a particular setting (Marcelo et al., 2011). Studies in this field also showed that, thread pitch has a critical effect on the fatigue life time of a bolt member. Coarse thread fasteners do have a better endurance limit compared to fine thread fasteners for both Metric and UNF series (Majzoobi et al., 2005; Munn & Li, 2010).

Preload plays a key role for the structural performance of a bolted joint loaded dynamically. A study on a failed wind turbine tower bolt showed that preload loss or inadequate preloads due to install variations are causing a decrease in the overall fatigue life of the joints when subjected to dynamical loads due to winds (Badrkhani Ajaei & Soyoz, 2020). Some of the studies found out that means stress values are

plain a key role in terms of the endurance limits until to a certain point. The observation was made that the endurance limit of bolts exhibits a drop that is related to the rise in mean stress. Nevertheless, when the average stress beyond a specific threshold, the endurance limit also exhibits an increase. It was also provided that most of the fatigue failures happened at the first engaged thread where the stresses are the highest (Shahani & Shakeri, 2015).

2.1.4 Friction Coefficient

The coefficient of friction is among the most critical factors in a fastened joint. By providing dimensional values as input, a dependable friction coefficient determination helps in the prediction and calculation of the resultant clamp load in a bolted joint. For this reason, awareness of the frictional properties is crucial for both design and service life considerations. In the context of applications subjected to purely static loading, it is advantageous to minimize friction in order to reduce torsional stress within the bolt during tightening resulting to a less input torque value. In the case of dynamic applications, a significant consideration is centered around the potential effect of low friction on the phenomenon of self-loosening. The majority of popular automotive companies determine specific friction limits for the fasteners they utilize in order to maintain the torque-tension relationship. The implementation of parametric equations in the industry aimed to provide an accurate basis and approach for regulating friction coefficient values. In attempts to achieve this purpose, equations given in ISO 16047 (ISO 16047: 2013) was formulated and subsequently adopted extensively as a standard control methodology. Formulation consists of three different main formulation given in Equation 2.6. First one is the torque clamp load relationship formula which takes friction coefficient based on thread, μ_{th} and bearing μ_b .

$$T = F \times \left[\frac{1}{2} \times \frac{P + 1.154 \times \pi \times \mu_{th} \times d_2}{\pi - 1.154 \times \mu_{th} \times \frac{P}{d_2}} + \mu_b \times \frac{D_o + d_h}{4} \right] \quad (2.6)$$

Later on, this formulation was then adopted to total friction coefficient as a general metric which is calculated based on the input torque, T , resultant clamp load, F , and

geometrical parameters. Formulation given in Equation 2.7 calculates friction coefficient values. Where P is the pitch of the bolt, D_2 pitch diameter, and D_b is the effective bearing diameter of the bolt. D_b is given more in detail in Equation 2.8 where D_o is the inner diameter of the contacting area and d_h is the outer side of the contacting area. The coefficient of total friction is exclusively valuable when comparing fastened joints under various friction conditions. The assumption underlying the formula for the coefficient of total friction is that the friction coefficients between bearing surfaces and threads are identical.

$$\mu_{tot} = \frac{\frac{T}{F} - \frac{p}{2\pi}}{0.577d_2 + 0.5D_b} \quad (2.7)$$

$$D_b = \frac{D_o + d_h}{2} \quad (2.8)$$

This standard gained more attention in the fastener industry as a quality control systematic in order to monitor frictional properties of a coated bolts. It also ensures a reliable estimation in terms of torque clamp load relationship if the other parameters of the joint are known.

2.1.4.1 *Effective Bearing Diameter Calculation*

Studies in the field of frictional properties of bolted joints showed that contact area especially in the bearing surface plays a key role in torque tensions relationship. Thus a specific effective bearing diameter calculation becomes essential for non-uniform bearing surfaces. Numerous investigations have been conducted within the literature to explore the importance and effects of most effective bearing diameter. ISO 16047 assumes a dimensionally mean value based on the outer and inner contact diameters. This can be visualized as given in Figure 2.9.

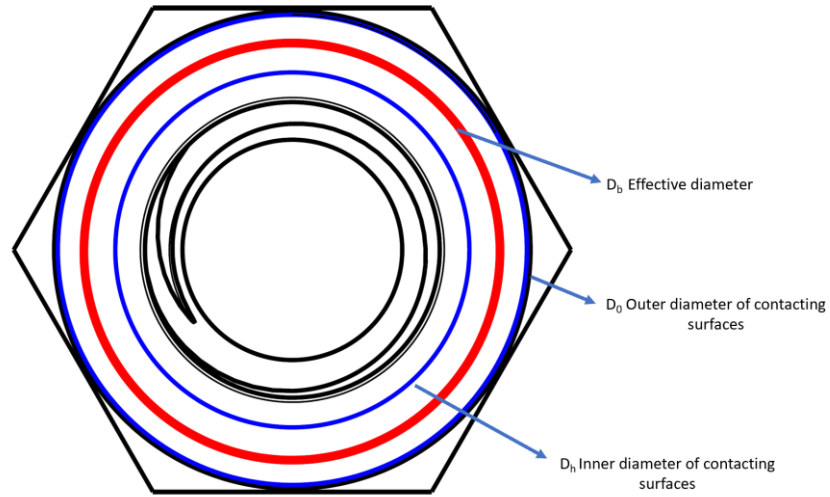


Figure 2.9 Effective bearing diameter representation

One of these studies and the well know, has used a pressure distribution model on the bearing surface. Finding with different pressure distribution showed up to 10% variations in the friction coefficient values (Nassar, Barber, et al., 2005; Nassar, El-Khiamy, et al., 2005). The utilization of a contact mechanics-based model has been a good means to investigate the effect of bearing pressure distribution on various surface conditions and hole diameter ratios. The effective contact value, r_b , obtained independently from rotational speed, is generated by the influence of the inner and outer contact radii on the torque output. The output obtained from the aforementioned study is presented in Equation 2.9. r_{b-max} represents the maximum external radius of contact, r_{b-min} denotes the minimum radius of contact, $P_b(r)r$ signifies the surface pressure at a given radius r , and r represents the distance from the axis in an axial direction. (Zou et al., 2005).

$$r_b = \frac{\int_{r_{b-min}}^{r_{b-max}} P_b(r)r^2 dr}{\int_{r_{b-min}}^{r_{b-max}} P_b(r)r dr} \quad (2.9)$$

2.1.4.2 Friction Change in Repeated Usage

Frictional changes play a key role in variation of the clamp loads. Studies showed that coating on fastener or different lubrication methods has a tendency to change with a repetitive usage. In a study carried out in this scope, used an 8.8 grade M12 electro plated zinc fastener without having an additional lubricating layer on the bolt surface. Results showed an increase in the in the friction coefficient values in both thread and bearing friction values in a drastic amount. Change of the friction based on number of tightening was given in Figure 2.10. A recommendation was made to not re use electro zinc plated fasteners which does not contain any lubricating elements as a conclusion (Eccles et al., 2010).

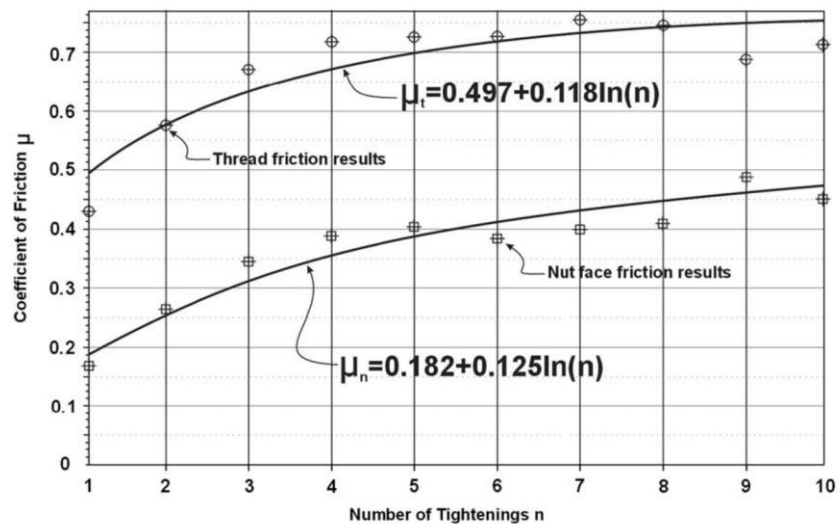


Figure 2.10 Friction variation in repeated usage of electroplated zinc fasteners without lubricating elements (Eccles et al., 2010)

Installation speed was also found to be a parameter which is affecting the friction coefficient of the threaded fasteners. It was found that frictional properties of unlubricated zinc coated fasteners increases drastically at 30 RPM and lower installation speeds while with a higher value effect of frictional variations is reducing. It was also found that the frictional variation range is reaching to a dramatical range when the tightening speeds are very low or repeated usages, where the surface roughness values start to play a key role (Nassar et al., 2007).

2.1.4.3 Coating Types

Application of coatings on fasteners is of significant significance in controlling and regulation of frictional properties. The stabilization of friction variation within a specific range is determined upon the type of lubricating element used, thereby providing engineers with a beneficial basis for their calculations. There is several of coating and lubricating elements depending on the application and friction range request. But coating types can be divided in 2 main sub division. Electroplated coating and non-electroplated flake coating systems. Both types possess their own independent application advantages. The main objective of implementing a coating system in fasteners is to protect the fastener components against the negative effects of the environment, thereby preventing corrosion. The utilization of zinc flake and other coatings containing polytetrafluoroethylene (PTFE) or alternative dry lubricants is a common practice within the automotive sector for the purpose of regulating friction properties. In various industrial sectors, the application of electro zinc plated coatings is prevalent. However, it is worth noting that these coatings generally lack a specific modification to enhance their frictional properties.

Specific studies focused on the performance of zinc flake coating systems revealed that storage condition and durations have an effect on the frictional properties. The effect of storage under cold conditions on tightening torque is found to be particularly significant, resulting in a reduction of approximately 33% during the initial tightening process (M. Kumar et al., 2022).

2.1.5 FEA Studies for Bolted Joints

Numerous engineers and scientists have directed their attention towards the simulation of bolted joints, with the aim of forecasting their behavior under designated load conditions. In order to achieve the objectives of this investigation, many different kinds of studies have been conducted over the course of the previous three decades. With the advancement of finite element analysis techniques and the increasing computational power, researchers have been able to tackle complex challenges within a significantly reduced timeframe. Moreover, these studies have demonstrated the

ability to accurately forecast outcomes, provided that all input parameters and boundary conditions are adequately specified.

Finite element analysis has become known as a valuable tool for researchers in the investigation of the effects of different parameters on bolted joints. By employing FEA, researchers have been able to gain insights into the complex mechanics and behaviors of these joints, thus improving our understanding of their performance characteristics. One of the research projects in order to investigate the effect of process control on the tightening of non-parallel contacting surfaces focused on using FEA methods. The findings of the study exhibited a high level of concordance with the experimental outcomes, therefore supporting its effectiveness as a viable methodology for non-parallel contacting surfaces (Ganeshmurthy & Nassar, 2014).

Some of the critical applications requires a unique tightening strategy which is called as tightening to yield. This method is benefiting from bolts strain hardening zone for a reliable joint performance. Thus, determination of the parameters plays a crucial role. In a study carried out in this scope, focused on using FEA tools to analyze bolt tightening beyond yield levels. For this purpose, an elastic-plastic material model was used and with that they were able to obtain reliable stress distribution and plastic deformation in the area where elongation was expected (Nagata et al., 2009). In order to have a reliable prediction outcome from finite element models, material properties need to be defined in good accordance with the real behavior. In this context, a simulation model to predict, thread strip and shank elongation failures in fire cases was simulated. Simulation results provided confirmation that the failure of thread stripping was primarily attributed to the introduction of various tolerance classes during the thread rolling process, shown as Figure 2.11 (Hu et al., 2016).

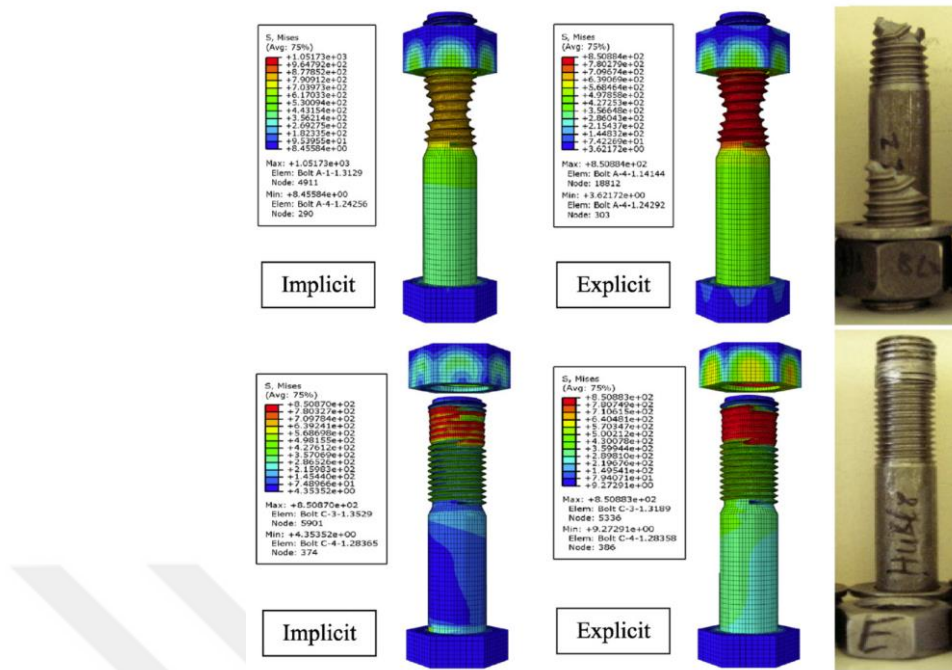


Figure 2.11 Thread strip and bolt shank elongation modeling with suitable material model et plastic region (Hu et al., 2016)

Various studies focused on the modeling of self-loosening phenomena in order to simulate different parametric values. An investigation focusing on simulating loosening effect with different transvers excitations showed various fluctuation in the contact states during loading cycles. The phenomenon of self-loosening in mechanical fasteners can be attributed to various mechanisms, including localized slip and complete screw head slip. It has been observed that self-loosening induced by localized slip occurs at considerably lower transverse displacements compared to self-loosening caused by complete screw head slip (Dinger & Friedrich, 2011; Hou & Liao, 2014). In a similar study, finite element model demonstrates the capability to accurately predict the four distinct loosening processes that have been observed in experimental studies (Figure 2.12). The finite element results effectively capture the fundamental characteristics exhibited by the experimental data. Model incorporates the fundamental factors responsible for the occurrence of loosening and serves as a robust tool for assessing the intricate aspects of fastener loosening (Pai & Hess, 2002).

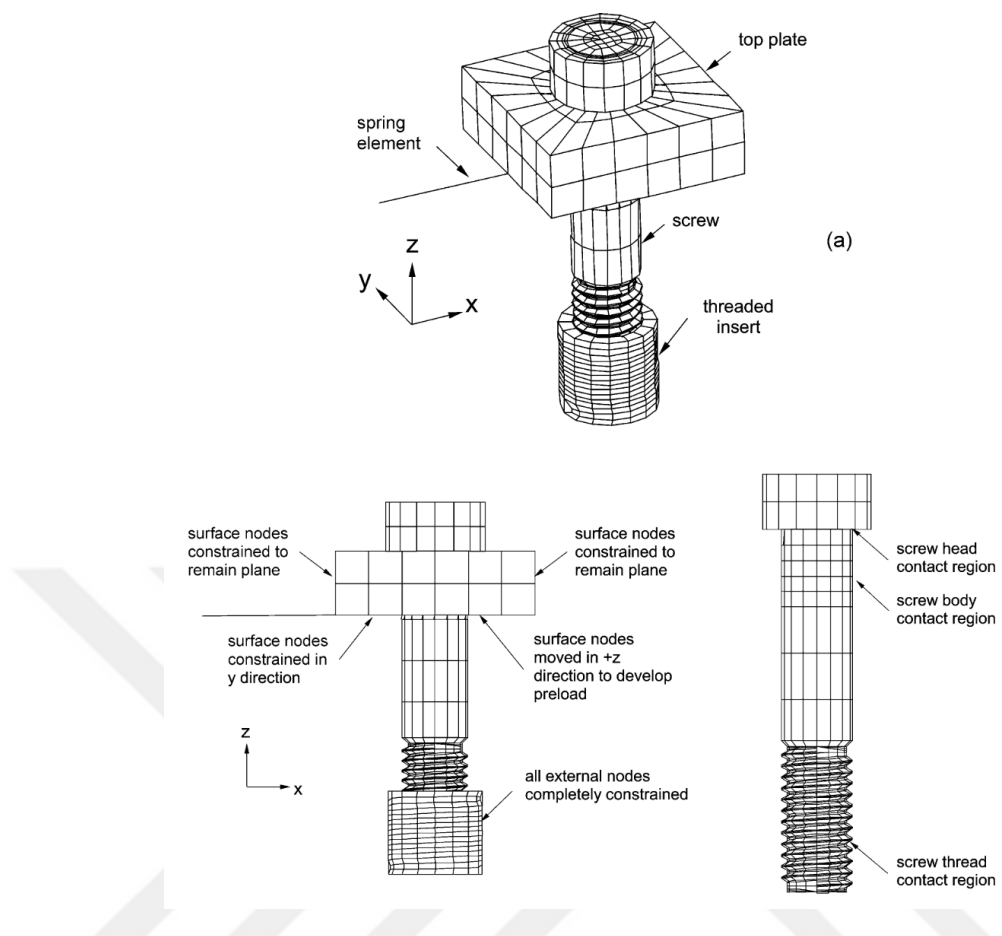


Figure 2.12 Finite element model boundary conditions (Pai & Hess, 2002)

CHAPTER THREE

MATERIAL AND METHOD

3.1 Methodology of the Research

In order to establish a robust predicting technique for bolts subjected to dynamic loads, it is essential to develop a comprehensive integrated system, particularly when transverse loads are present. Within the limitations of this particular field, it is vital to use and enhance empirical findings. Experimental studies play a crucial role in establishing a foundational framework for a comprehensive investigation, enabling the development and following evaluation of outcome results. Throughout the duration of this research, an extensive variety of experimental methodologies were utilized. The empirical observations were afterwards employed as input parameters to forecast and establish the phenomena exhibited by bolted connections across a range of dimensions, covering both the micro and macroscopic scales. In order to fulfill the objectives of this study, a series of experimental procedures were conducted, such as friction coefficient testing, Junker vibration testing and fatigue testing. Figure 3.1 presents a comprehensive overview of the various methods utilized to develop a model for the development of friction, fatigue, and vibration loosening. A development of a machine learning model which is capable of predicting loosening rates based on experimental and simulation inputs is also introduced. This model also enables the addition of future inputs, so serving as a database for predicting loosening rates. In which it automatically adjusts its performance by iteratively evaluating multiple key parameters to find the best performing model parameters. The development of a comprehensive prediction model requires an in-depth knowledge of the specific impacts linked to conducting experimental studies. This understanding serves as essential for the establishment of a detailed and accurate model. Each sub steps for mode development and details about the experimental studies will be given in the upcoming sections in detail.

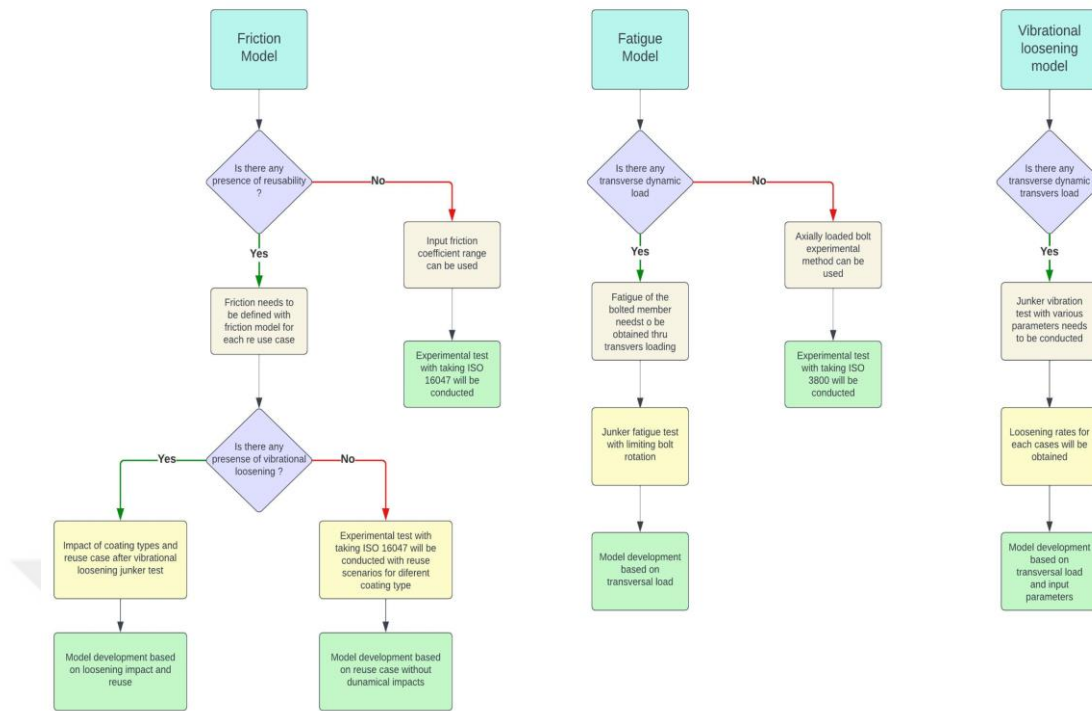


Figure 3.1 Friction, fatigue and vibrational loosening model development steps

3.2 Experimental Method

3.2.1 Friction Coefficient Test

Experimental tests are used to determine friction coefficient outcomes of various conditions in order to have a baseline for the prediction method. For this purpose, a widely accepted industry standard to determine friction coefficient values are used with a standard test bench for this purpose. For this purpose, ISO 16047 was used as a main calculation and evaluation methodology, where it defines the requirements of the test bench and formulations for friction coefficient calculation. Test setup given in Figure 3.2 shows the sub components of the overall system. Device are made of multiple sections. Torque and clamp load values are measured with the load transducer. Nut adapter aim to keep nut without rotation while tightening the bolt onto the test plate which has a hardened washer in the surface.

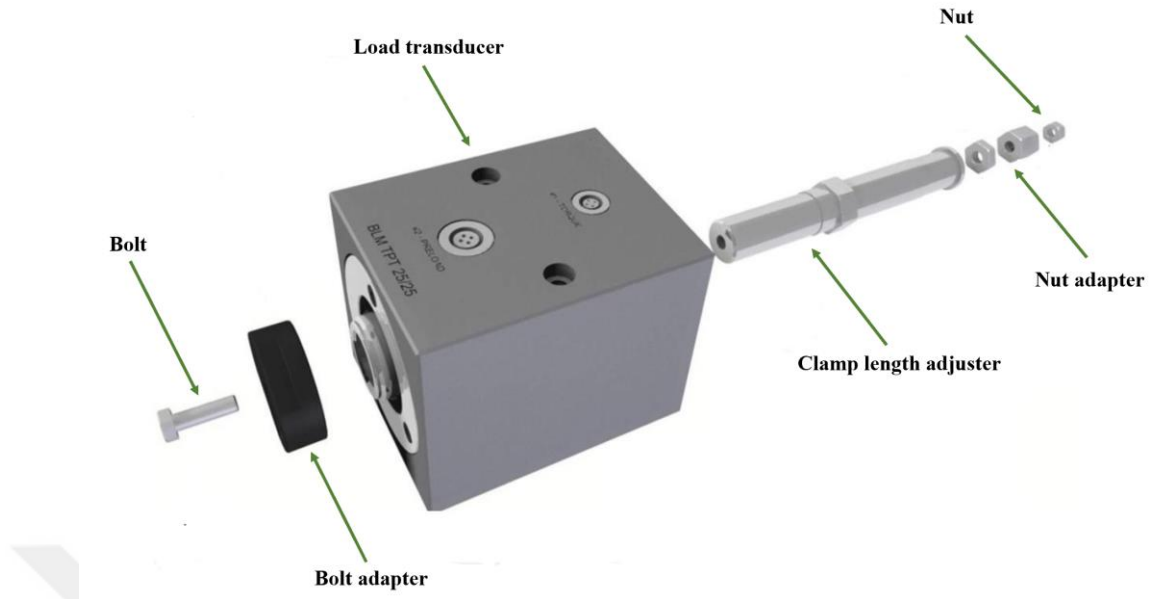


Figure 3.2. BLM friction test device

In order to calculate the friction coefficient values, Equation 3.1 and 3.2 are used based on the geometrical inputs in combination with the measured preload and torque values.

$$\mu_{tot} = \frac{\frac{T}{F} - \frac{p}{2\pi}}{0.577d_2 + 0.5D_b} \quad (3.1)$$

$$D_b = \frac{D_o + d_h}{2} \quad (3.2)$$

For the under head washer, a special hardened, grinded, uncoated test washer was used based on the requirements of ISO 16047 requirements. This is guaranteeing the consistency of outcomes without getting affecting by the other variations. During the experimental study a various bolt condition has been tested for both friction and Junker vibration test scope. For the bolts, a standard ISO 4017 fully threaded fasteners were used with a grade 8.8 and 10.9 with various metric sizes and coatings which all are

coarse thread. For the female threaded member, a nut according to ISO 4032 was again used with a grade 10 including various coating.

In order to evaluate the effect of frictional properties and coating type 2 different base coat type was used during the experimental study. These are Zinc Cr^{+3} electroplated coating series with TNT12 top coat for friction modifier and Zinc aluminum flake Delta Protect KL100 series with a top coat of VH301GZ, pictures of the bolts are given in Figure 3.3. Details regarding the coating types and their frictional properties are given in Table 3.1.

Table 3.1 Coating types

Coating Type	Base Coat	Top Coat	Friction Coefficient Range
Electroplated	Zinc Cr^{+3}	TNT12	0.09-0.14
Non electroplated	Delta Proteckt KL100	VH302GZ	0.09-0.14

In order to evaluate frictional changes during repeated tightening a series of test were carried out with reusing the same bolt multiple times. Their resultant clamp and torque values are monitored, and their frictional values are obtained for each cases.



Figure 3.3 M8x1.25 bolts with Zinc Flake coating type on the top and Zinc Cr^{+3} on the bottom

3.2.1.1 *Friction Coefficient Test for initial install*

Initial friction coefficient range of a fasteners are controlled by the topcoat friction coefficient range. That is why, bolt manufacturer is obliged to keep the friction variations in the specifications given by the topcoat value or by the OEM specifications. This test methodology consists of calculating frictional properties based on the input torque and outcome clamp load values with using dimensional values of the tested bolt. In this scope, we also tested our test samples for validating if the sample batch which we are going to use during experimental studies falls in the desired range or not. 10 repeated tests at the same conditions were carried out and the mean friction coefficient value is calculated based on the outcomes.

3.2.1.2 *Friction coefficient test in reuse application without dynamic effects*

In this scope a similar methodology has been followed as explained in the methodology section. In addition, each bolt has been used for at least 5 times and their clamp load torque values are recorder. At the end of each tests frictional values are calculated and the change of the frictional properties are obtained for this condition.

3.2.1.3 *Friction coefficient test in reuse application with loosening effects*

This test methodology is unique to this thesis in order to evaluate the loosening effect on the frictional properties of the fastener in each re use case. That is why this test has been carried in Junker vibration test setup and the clamp loads are read from its transducer. Input torque values are obtained with the Atlas Copco ST Wrench series. Later on, frictional coefficients are calculated for each condition. In order to evaluate dynamic loosening effect on the friction coefficient value, at least 10 repeated usages have been carried out. Bolts are not replaced with a new test bolt, but instead they were reused after each Junker vibration test and retightened again to their clamp load targets. Figure showing the tightening process of a fastener is given in Figure 3.4.



Figure 3.4 Bolt installation in Junker vibration test device with Atlas Copco ST Wrench

3.2.2 *Junker Vibration Test*

The Junker test equipment is comprised of two plates, with one plate being stationary and the other plate being movable. These plates are positioned in a manner that is perpendicular to the axis of the bolt, and they exhibit relative motion in relation to each other. The implementation of cam mechanisms allows for the manipulation of displacement or lateral force values, hence enabling the lateral movement of the top plate and creating vibrations in the intended direction. During the entire study the total displacement of the moving plate was kept at 0.9 mm which corresponds to ± 0.45 relative to the bolt axis. This value was measured with a comparator which is installed at the edge of the moving plate. Test duration was set to 2000 cycle, which is an accepted range for low-mid cycle loosening, if bolt clamp load is not loss by %80 percent. A picture of the test setup is given in Figure 3.5 and a sketch of the cross section for inner visibility is given in Figure 3.6.

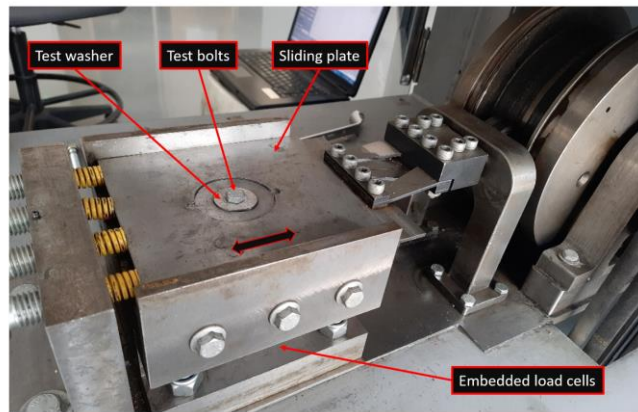


Figure 3.5 Junker vibration test device

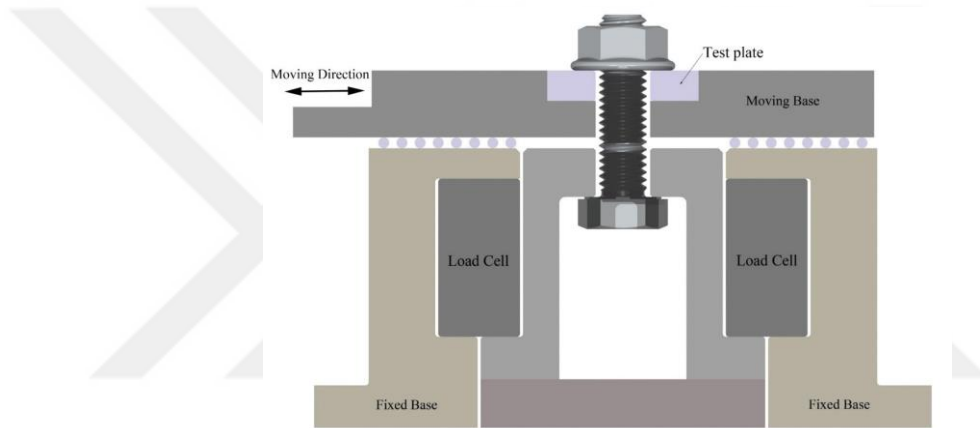


Figure 3.6 Junker vibration test device, cross section

3.2.2.1 *Junker Vibration for Loosening Rate Investigation*

This test method aims to test virgin bolts and nuts for obtaining their loosening rate responses at the end of each test competition. Aim is to obtain clamp load cycle graphs and evaluate their clamp load reductions per cycle. Test bench is capable of reading clamp load values in each transvers amplitude, allowing us to create clamp load cycle graphs. Lateral load frequency is fixed to 5 Hertz and a lateral displacement of ± 0.45 mm. Clamp load, clamp lengths and metric sizes are adjustable in the test bench. In the upcoming sections conducted test series and their input values will be given in details. Due to the nature of the loosening behavior at harsh lateral loads, trend of loosening generally becomes linear. That is why a linear fit has been carried out for the mean of the tested combinations in order to obtain loosening rates for that test

inputs. A representation of how clamp load cycle graphs are obtained and how the loosening rates are calculated is given in Figure 3.7.

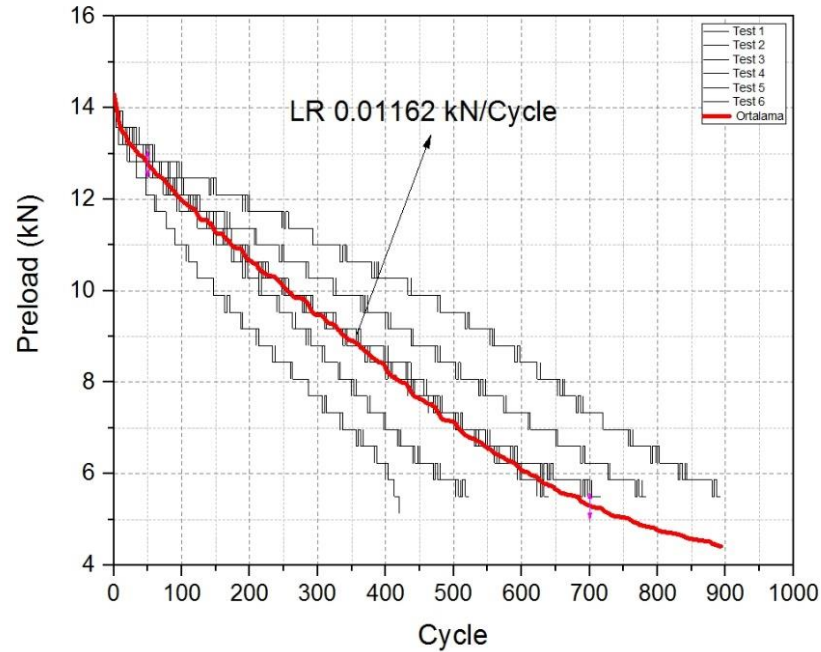


Figure 3.7 An example of the preload-cycle graph for loosening rate calculation

3.2.2.2 Junker Vibration for Loosening Rate Investigation of Re-tightening Condition

A test technique was created to evaluate the reusability of fasteners when subjected to dynamic effects that may cause loosening. The objective was to determine if the same bolt could be reused for a minimum of 10 times. The bolt, nut, and test washer remained consistent throughout the whole re-use cycle in order to assess the rate of loosening. Within the parameters of this study, the initial installation clamp load and starting torque value were documented, and subsequently, the Junker test was initiated. After the completion of each Junker test, the bolt has been retightened to its designated clamp load by progressively increasing the input torque during installation. This approach enabled the examination of the influence of friction coefficient values on the behavior of loosening rate in a repeated usage scenario.

3.2.2.3 Junker Vibration Test for Fatigue Life under Transvers Load

In the scope of the study, bolts are subjected to lateral loads which causes bending loads in the first contact thread of the bolt as well as under the bolt head radius. Experimental studies in the Junker vibration test setup showed that bolts are getting fractured due to excessive bending loads. In order to evaluate this condition, bolt rotation was restricted with a special test washer. This ensures that bolt clamp load change will not affect the fatigue life test results and thus make possible to obtain fatigue responses of the bolts under bending loads. A various clamp load needs to be tested in order to use these fatigue data for a prediction methodology. A detailed explanation of the test methodology will be given in the upcoming sections more in details with the results obtained.

3.2.3 Axial Fatigue Test

Axial fatigue tests has been carried out for obtaining the Wohler curves of each load cases and in order to develop a Haigh diagram for axial fatigue prediction methodology. A classical method described in ISO 969 is used to obtain the curves under various amplitudes and mean stresses. Test device used for the experimental test is Zwick Roell Vibraphone 250 kN. Test device used during the axial fatigue test and its representation is given in Figure 3.8.

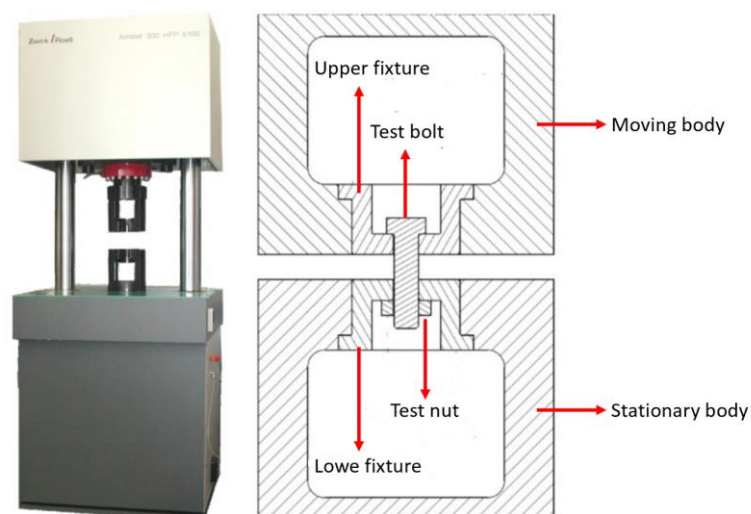


Figure 3.8 Axial fatigue test device and its sketch with components

Following to the Wohler curves a Haigh diagram was developed by merging various Wohler curves, resulting in the creation of a fatigue map for the M8 8.8 quality bolt. The fatigue maps are dependent upon the quality of the bolts. Furthermore, consistent outcomes are achieved across various head geometries. Two distinct Wohler curves have been obtained within the limits of this investigation. The provided data includes the Wohler curve corresponding to an average load value of 15 kN.

3.3 Finite Element Analysis

Finite Element Methods are widely used in the industry to deliver a reliable outcome depending on the boundary conditions. Within this study, ANSYS R23 workbench and transient solvers are used to simulate tightening process and also Junker vibration loosening phenomena. In order to have a proper outcome from the FEA, there is a need to define material, boundary, and load conditions properly. Due to the nature of the bolt tightening region and loosening phenomena being in the elastic region, there is no need to define plastic behaviors of the materials in detail. Both tightening and loosening simulations are modeled with transient analysis with a time dependent inputs.

Joint CAD is designed with using CATIA V5 and exported as step file. Step file is then imported to ANSYS workbench as a model input. Fasteners threads are created with high precision with using the theoretical and measured values from the actual bolts in order to obtain the best outcome from the simulations. Theoretical thread form is given in Figure 3.9 for a M8x1.25 6g thread. This thread for is then sweep cut from a helix profile and our threaded models are obtained as Figure 3.10.

FEA results are used to have a solid foundation for the loosening prediction method as well as a robust friction change model based on sliding distances. Both tightening and loosening models are validated and calibrated with experimental results. Later on these models are used to predict variable amplitude behaviors of a bolted joint where experimental tests are not applicable. Details in terms of model and results are shared in the results and discussion section more in detail.

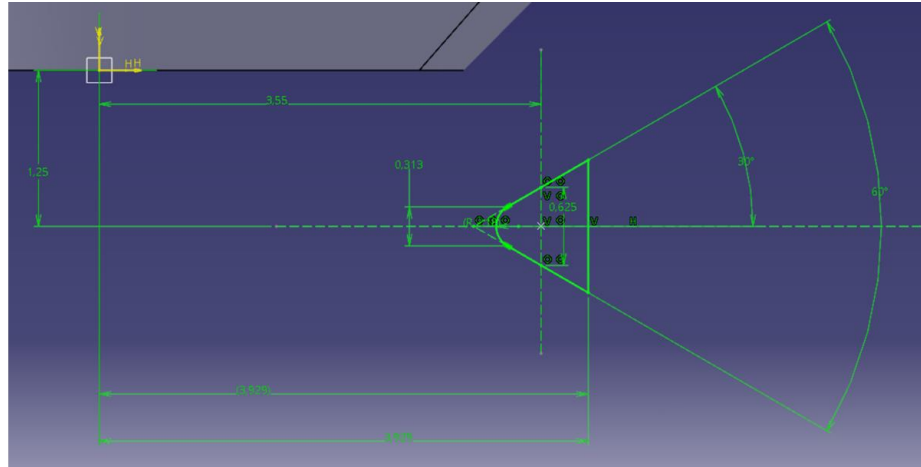


Figure 3.9 Thread cut theoretical dimensions for M8x1.25 6g thread profile

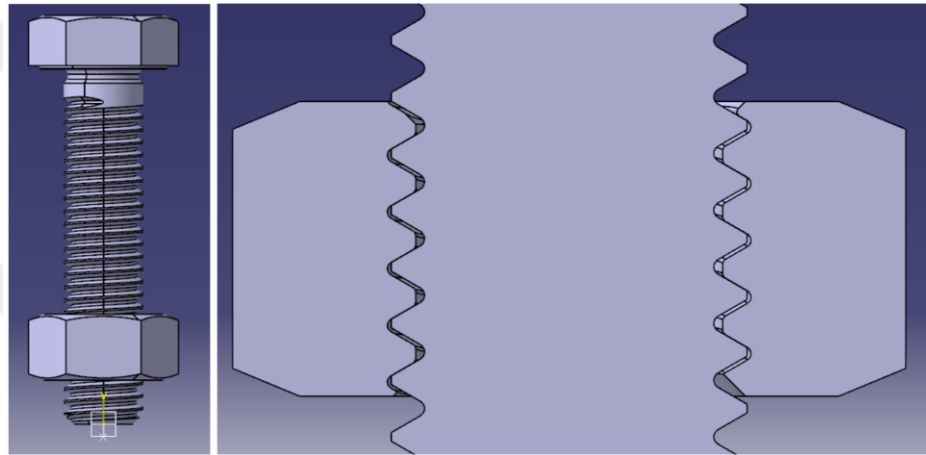


Figure 3.10 Final thread shape after thread cut and bolt nut assembly

3.4 Neural Network Based Loosening Rate Prediction Method

Neural networks are created to have a database for loosening rate predictions with obtaining a general model based on experimental and simulation results. Main inputs used to train machine learning model is clamp load, clamp length, friction coefficient, transvers displacement. Model is trained to provide loosening rates for any given inputs. To develop and train the model, Python codes are used with various libraries for training, plotting and predicting the outputs. The Python code is built with a feedforward artificial neural network architecture which is a suitable method for regression problems. General model was implemented with the commonly used deep

learning platform TensorFlow/Keras. Model is based on a live systematic which can be fed with input data in time and can generate a new updated model with iterating various activation and dense functions. This enables the structure of the code to be flexible and able to adjust itself based on new input data. Main structure of the machine learning model can be used as a database for loosening rate prediction. Model can be trained with non-limited experimental and simulation inputs which will make its predictive capabilities more robust in time without affecting the main outcome methodology of the structure. Initial model is trained with 120 Junker vibration test results and 62 simulation results. During the training %20 percent of the inputs data is used as a test and %80 as a train. A section of the general training model is given in Figure 3.11

```

8 # Input datas are loaded from the excel files
9 file_path = '/content/fulldata.xlsx'
10 data = pd.read_excel(file_path)
11
12 # Split data into input (X) and output (y)
13 X = data[['clamp load', 'clamp length', 'friction coefficient', 'lateral force', 'metric size']]
14 y = data['loosening rate']
15
16 best_r_squared = -float('inf')
17 best_config = None
18 tried_configs = set()
19 max_trials = 1000 # Define max_trials here
20
21 # Define a function to generate all configurations
22 def generate_all_configs():
23     layer_counts = list(range(3, 9))
24     layer_dense = list(range(5, 25))
25     activations = ['relu'] # Randomly select activation function
26     return list(itertools.product(layer_counts, layer_dense, activations))
27
28 # Get all possible configurations
29 all_configs = generate_all_configs()
30
31 # Define a function for model training and evaluation
32 def train_evaluate_model(layer_count, layer_dense, activation, X_train, X_test, y_train, y_test):
33     model = tf.keras.Sequential()
34     model.add(tf.keras.layers.Dense(layer_dense, activation=activation, input_shape=(X_train.shape[1],)))
35     for _ in range(layer_count - 1):
36         model.add(tf.keras.layers.Dense(layer_dense, activation=activation))
37     model.add(tf.keras.layers.Dense(1)) # Output layer (1 neuron for regression)
38
39     model.compile(optimizer='adam', loss='mean_squared_error')
40
41     model.fit(X_train, y_train, epochs=100, batch_size=32, verbose=0)
42
43     predictions = model.predict(X_test)
44     r_squared = r2_score(y_test, predictions)

```

Figure 3.11 Main structure code of the machine learning model

In order to obtain the best predictive performance, an iteration loop which adjust layer count, layer density and activation functions is used and adjusted r square outcomes for each iteration are measured. Once all iterations are completed, model re trains the base model with the highest r square obtained iteration and use as a main predictive model. Section of the best model re-training is given in Figure 3.12. Epoch for each iteration are set to 100. Various visualization tools and codes are used to monitor MSE, R square and predictions. For this purpose, matplotlib library is used in general. A general logic of the developed code is given in Figure 3.13.

```

76 # After finding the best configuration, retrain the model with the best configuration
77 best_layer_count, best_layer_dense, best_activation = best_config
78 X_train, X_test, y_train, y_test = train_test_split(X, y, test_size=0.2, random_state=42)
79
80 # Retrain the model with the best configuration
81 model = tf.keras.Sequential()
82 model.add(tf.keras.layers.Dense(best_layer_dense, activation=best_activation, input_shape=(X_train.shape[1],)))
83 for _ in range(best_layer_count - 1):
84     model.add(tf.keras.layers.Dense(best_layer_dense, activation=best_activation))
85 model.add(tf.keras.layers.Dense(1)) # Output layer (1 neuron for regression)
86
87 model.compile(optimizer='adam', loss='mean_squared_error')
88 model.fit(X_train, y_train, epochs=100, batch_size=32, verbose=0)
89
90 # Predict on test data using the trained model
91 predictions = model.predict(X_test)
92
93 import matplotlib.pyplot as plt
94
95 # Visualize actual vs predicted values
96 plt.figure(figsize=(8, 6))
97 plt.scatter(y_test, predictions)
98 plt.xlabel('Actual Values')
99 plt.ylabel('Predicted Values')
100 plt.title('Actual vs Predicted Values')
101

```

Figure 3.12 Code of retraining the prediction model with the best parameters

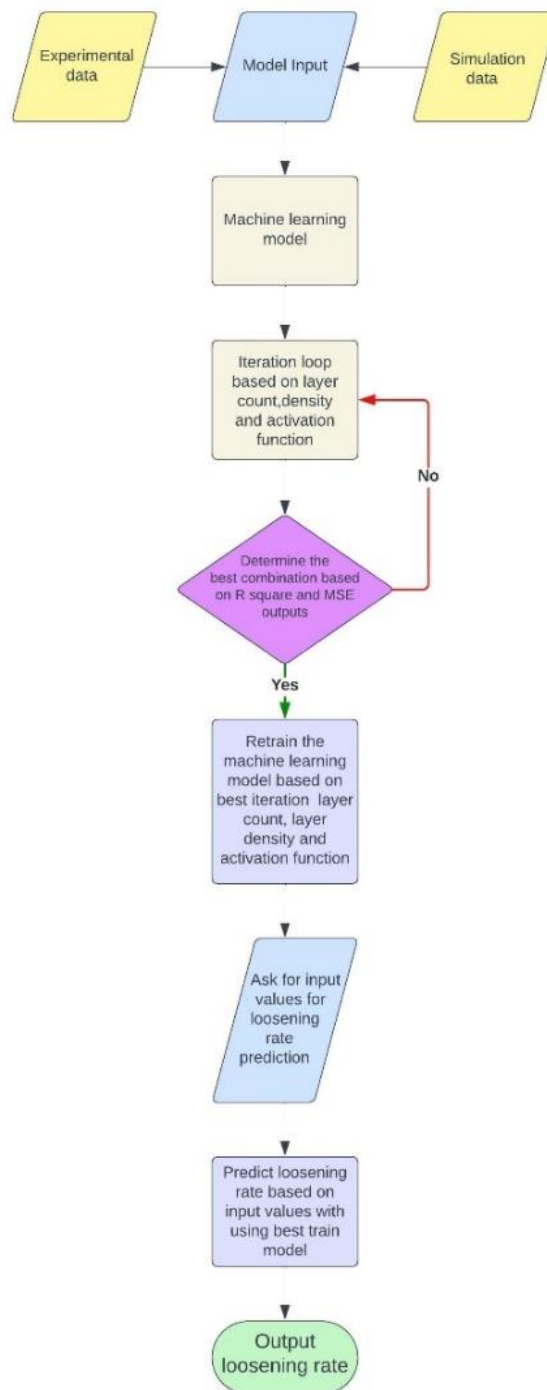


Figure 3.13 Axial fatigue test device and its sketch with components

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Development of Friction Change Model under Non-Dynamic and Dynamic Loading Cases

Frictional properties of a bolt play a crucial role in determining the clamp load outcome of a fastener after install. That is why change in friction coefficient is a key affecting parameter which can lead to failures and as unexpected outcomes. Empirical equations are capable of determining the frictional properties of a bolt with taking various parameters into account, but even in these outcomes the frictional change models have not been investigated in detail. Fasteners are elements which can be used multiple times if the clamp load range stays in the elastic region. But one effect is always underestimated while reusing fasteners. Depending on the coating type of the fasteners, frictional responses of these surfaces react in a different manner in re-use application. In order to stabilize frictional variations during multi-use, a lubricating element was suggested especially in construction environment. But in other industries or in non-controlled service conditions bolts are kept as they are. That is why some of the international standards do a re-use case at least 5 times in order to see the coating degradation and frictional change responses depended on the coating type.

During testing, two principal process methods were used. For loosening effect analysis, following are the procedure steps for each installation cycle: "Method 1": Tightening > Installation torque reading > Clamp Load reading > Vibration test > Completely loosening the fastener dynamically > Retightening bolt > friction measurements, "Method 2" is utilized; Tightening > Installation torque reading > Clamp Load reading > Bolt completely loosened manually > Retightening. Table 2 provides a detailed Test matrix.

During the scope of this section, an investigation was made for the most used two different coating types and their frictional responses in multi-use application in test bench was monitored. Figure 4.1 illustrates the variations in frictional properties of fasteners coated with Zinc Cr^{+3} in several instances of recurrent usage, specifically in the context of manual loosening. Manual loosening is deferred as a loosening case with

manual force, without any acting vibrational loads to cause this phenomenon. A trend of frictional change in both M thread and M bearing as well as in M tot, has been seen clearly. With the bolt frictional design range being in the range of 0.09-0.14 it is obvious that with the third installation there is a high chance that bolt friction coefficient will stay out of design specification defined on the coating. With the following tightening on the same bolt, it was seen that after fourth tightening friction change trend starts to decrease and normalize itself and stay in the same range. This indicates that after the third cycle the effect of the lubricating layer, aka wax, starts to loose its effect to stabilize friction coefficient values. In order to have a broad understanding of the change behavior of the total friction coefficient, a curve fit is implemented on the overall data.

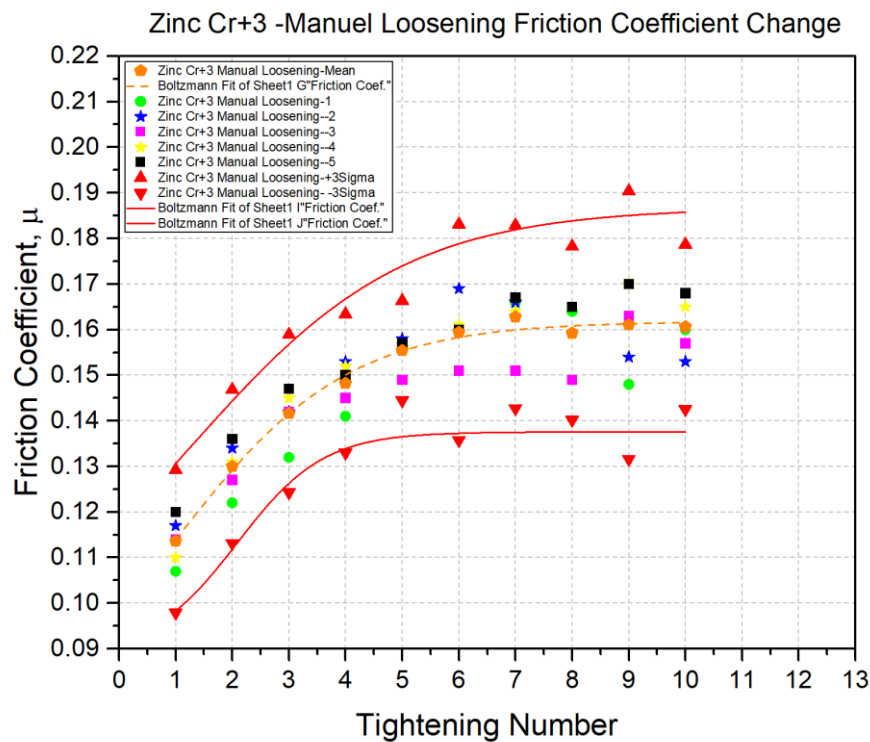


Figure 4.1 Frictional change of Zinc Cr⁺³ under repeated tightening without dynamic effects

A similar test has been conducted to the zinc flake (non-electroplated) coated fasteners as well. A similar behavior has been seen but with a less aggressive frictional changes. Frictional changes of 10 repeated assembly were given in Figure 4.2. From

the results obtained, after fourth cycle friction coefficient starts to increase and stabilize itself after 8th tightening. In full repeated tightening sequence, resultant friction coefficient is still on the desired target design friction coefficient range. ± 3 Sigma represent the desired normal distribution which a joint can experience with the variation in frictional ranges. Even in that range max friction coefficient is not increasing above the maximum design limits.

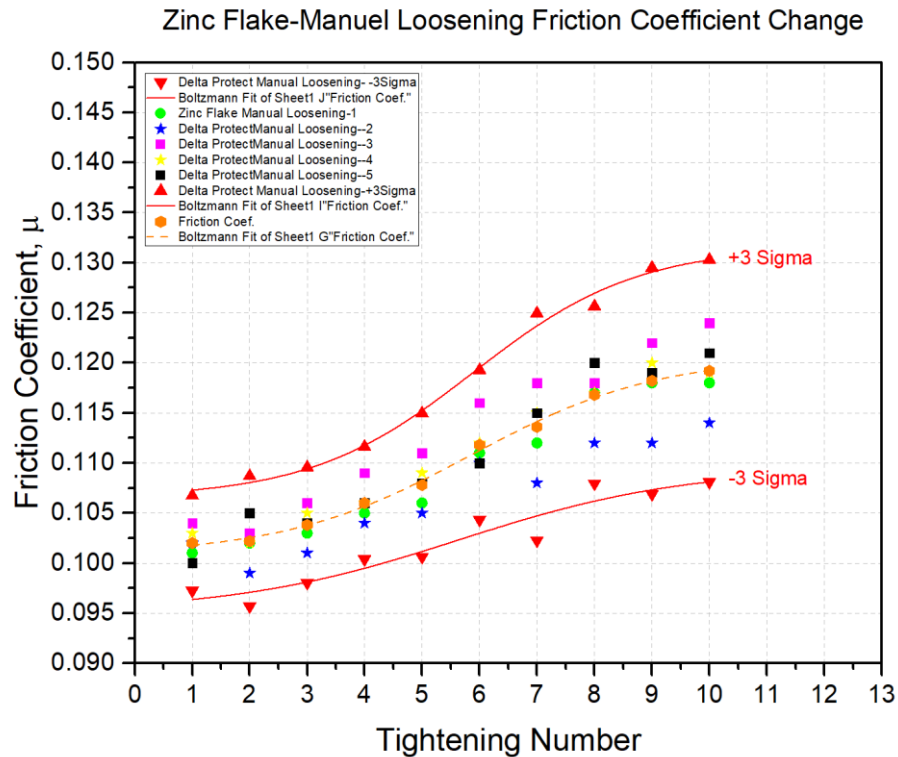


Figure 4.2 Frictional change of Zinc Flake coatings under repeated tightening without dynamic effects

A comparison was made clear in bar chart given in Figure 4.3 between these two coating types. A change in frictional properties of both types where there are not any dynamical loads which causes bolt to loosen or rotate shows a clear outcome that Zinc Flake coating systems will be a better option. But in the other hand it should be noted that these variations and change behavior may vary with various surface properties acting on the contacting surfaces. In both instances, however, the trend of frictional changes will remain constant, and the layer performance will be the primary input and control.

In order to investigate vibrational loosening effect on frictional properties a different test methodology was followed. Junker vibration test device was used and repeated tightening before each install has been carried out. In order to establish a comparable baseline for loosened and non-loosened fasteners, clamp load and install torque measurements were taken immediately from the device. Test sequence of consist of previously mentioned ‘Method 1’ systematic where after each cycle, re tightening of the same bolt is being made and Junker vibration test device started again. With this combination, the effect of micro and macro slips due to lateral vibrations has been investigated. This methodology has been followed in both coating types.

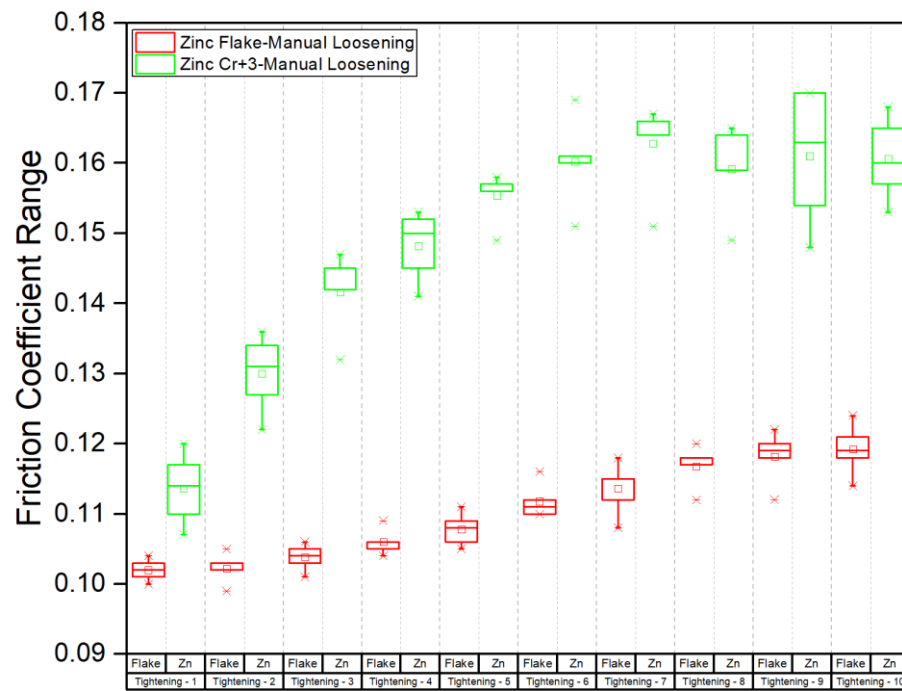


Figure 4.3 Frictional change comparison of both coating types

In order to investigate vibrational loosening effect on frictional properties a different test methodology was followed. Junker vibration test device was used and repeated tightening before each install has been carried out. In order to establish a comparable baseline for loosened and non-loosened fasteners, clamp load and install torque measurements were taken immediately from the device. Test sequence of consist of previously mentioned ‘Method 1’ systematic where after each cycle, re tightening of the same bolt is being made and Junker vibration test device started again. With this combination, the effect of micro and macro slips due to lateral vibrations has been investigated. This methodology has been followed in both coating types.

tightening of the same bolt is being made and Junker vibration test device started again. With this combination, the effect of micro and macro slips due to lateral vibrations has been investigated. This methodology has been followed in both coating types.

Based on the aforementioned test methods involving Zinc Cr⁺³ coated fasteners, it is evident from Figure 4.4 that the friction coefficient value exhibits a significant rise after each cycle. During the third cycle, an observation of the friction coefficient value reveals a deviation from the expected range as specified by the design parameters.

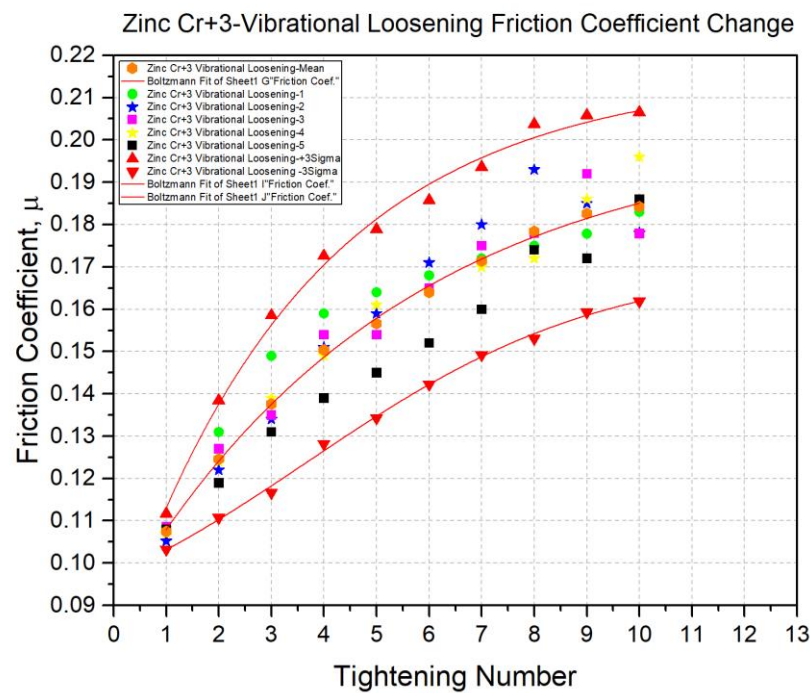


Figure 4.4 Frictional change of vibrational loosened Zinc Cr⁺³ under repeated tightening

A similar experiment was conducted on the series of fasteners coated with zinc flake, and the findings are presented in Figure 4.5. A similar trend compared to Zinc Cr⁺³ fasteners has been observed. The friction coefficient exhibits a quick rise following to the first use of tightening and subsequent loosening due to vibrations. The data demonstrates a noticeable and quick increase in the friction coefficient value beginning from the fourth cycle, followed by a stabilization of the value after the sixth cycle. One notable observation is that the biggest range of friction variation between

the fourth and sixth cycles is comparatively higher in this particular phenomenon when compared to other cycles.

Similar box chart comparison was made for Zinc Flake versus Zinc Cr+3 coating series as well as given in Figure 4.6. It was clear that the trend of friction increase is similar to Zinc Cr+3 when there is an exposure of dynamical loadings which causes bolt to loosen in time.

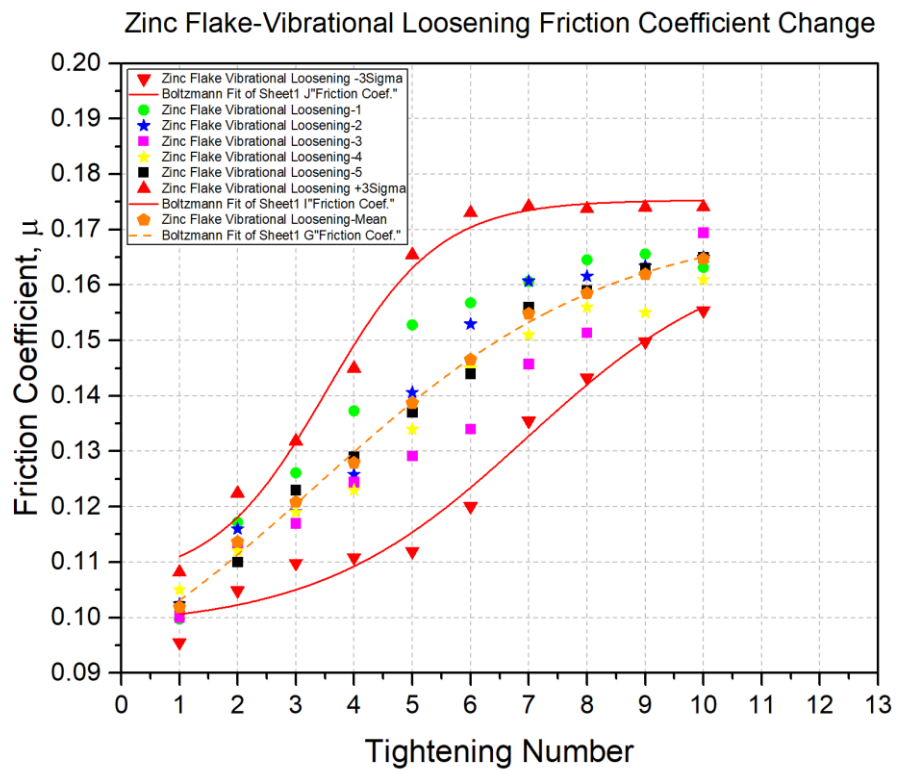


Figure 4.5 Frictional change of vibrational loosened zinc flake under repeated tightening

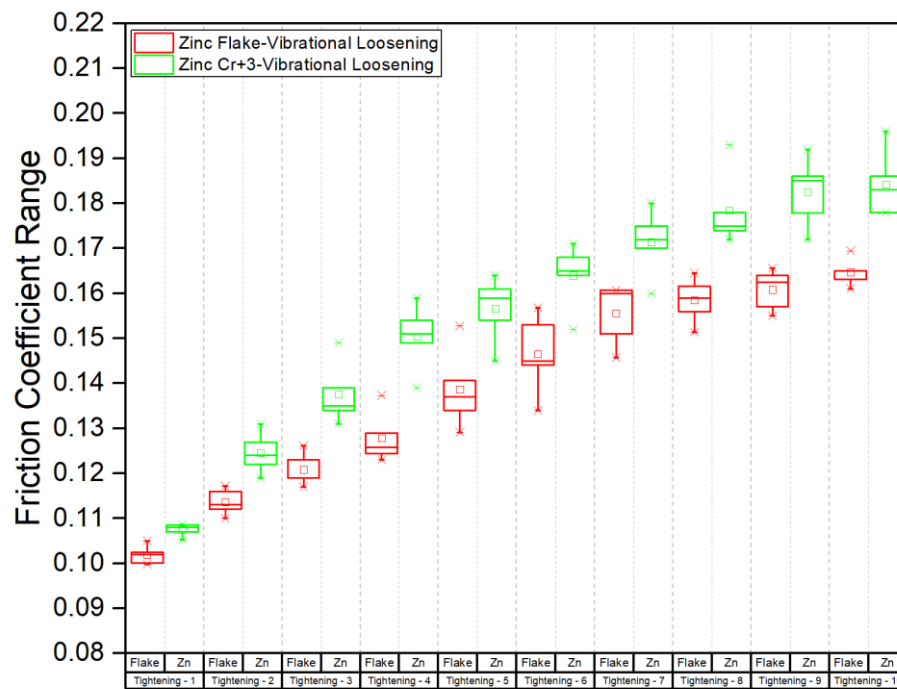


Figure 4.6 Frictional change comparison of both coating types under vibrational loosening

Like previously stated, friction change of zinc flake coating series are more prone to increase when there is dynamical loosening effects. It was concluded that if there is a concern of a vibrational loosening in fastener and a target to reuse the same bolt, it was clear that friction coefficient value will surpass the max design range starting from 6th tightening.

In order to have a better visual understanding and a comparison of the dynamical effect on the friction variations, various scatter plots were created. Figure 4.7 shows the response of dynamically loosened and manually loosed Zinc Cr⁺³ fastener behaviors. In both loading condition, it was observed that friction change trend followed a similar behavior and range. This can be attributed of the surface hardness of the coating type and the effect of the additional wax layer on the bolts. According to the experimental results, Zinc Cr⁺³ fasteners friction coefficient values are not affected by the dynamical effects where loosening's are effective. Starting with 6th tightening friction values for both cases start to surpass the design friction limitation values.

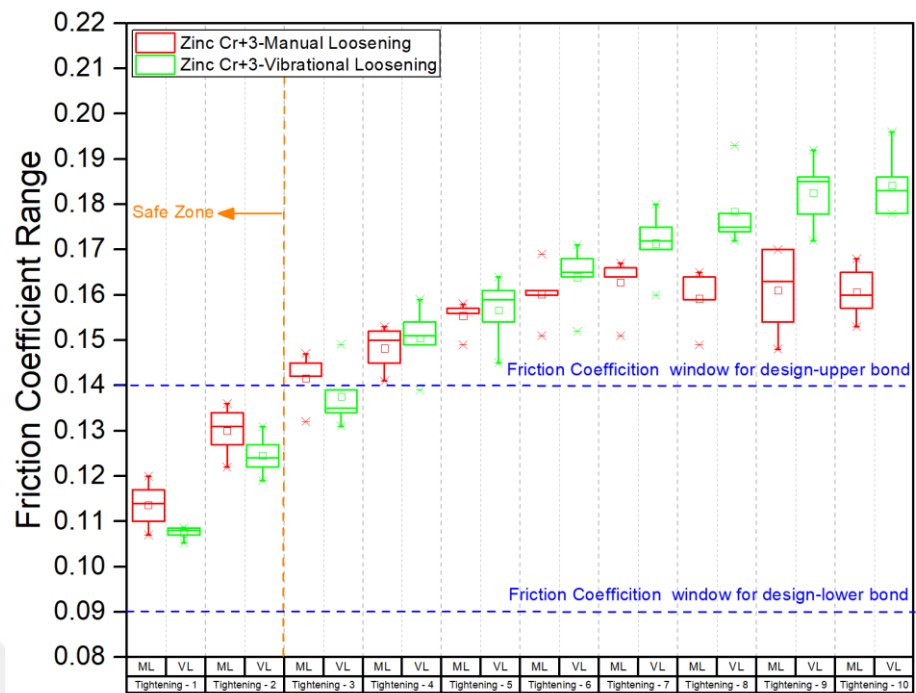


Figure 4.7 Frictional change of Zinc Cr³ under dynamic loosening and manual loosening

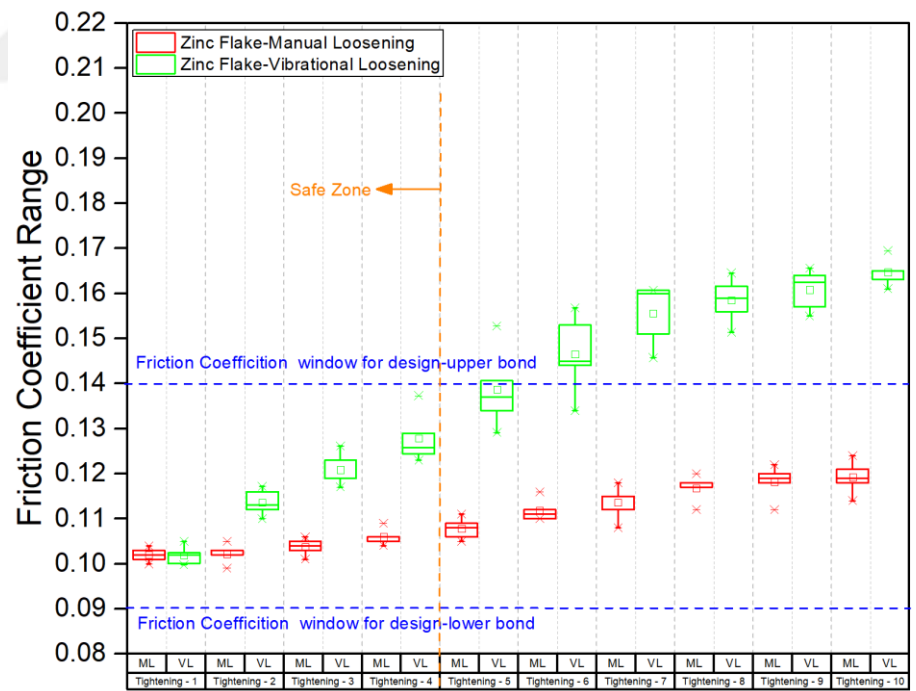


Figure 4.8 Frictional change of Zinc flake under dynamic loosening and manual loosening

A similar representation was also made for Zinc flake coating series under both loading condition and shared in Figure 4.8. It was clear that the dynamical loosening effects frictional responses and cause to surpass design limitations starting with 6th tightening. It was concluded that loosening has a key effect in terms of frictional behavior and needs to be considered where there is a possibility of reusing the same bolt multiple times. Picture of the thread surface of a worn out fastener due to multiple use.



Figure 4.9 Worn threads due to multiple re use

4.2 Prediction of Loosening Rates under Transversal Dynamic Loads

Loosening rates are one of the key outputs in order to evaluate the bolt clamp load losses. This expression can be described as an amount of clamp load loss in each transvers cycle. It is generally expressed as kN/cycle. In order to determine that loss rate there is a need to run Junker vibration tests and extract the curve of clamp load reduction per cycle. Curve responses may vary depending on the input parameters like bearing diameter, clamp length, clamp load, transvers displacement and friction coefficient values. Clamp load decrease in cycle follows a linear trend generally if the transvers loads are severe. In some instances where the loss rates are small enough, sudden loss of clamp load may happen after certain cycle and cause the curve to be

polynomial. In the concept of the study, we are focusing in the area where the most critical loosening states happens in which the outcome is linear. Multiple Junker tests were carried out in a displacement range of ± 0.45 with different parametric set. Each combination contains 5 distinct test which are then averaged to obtain a mean loosening curve and to obtain a linear curve fit to determine their loosening rate ratios. This value will later be used as the key input parameter in the relaxation estimation model. As already know from the literature, clamp load has an effect on the loosening curve responses. Plotted graphical test data obtained from the Junker vibration test device is given in 2 different clamp load values in which the effect can be seen clearly. Figure 4.10 shows the clamp load cycle values of a 25mm clamp length value at 15 kN. On the other hand, Figure 4.11 shows the 40mm clamp length version of the same bolt similarly for Figure 4.12 and 4.13.

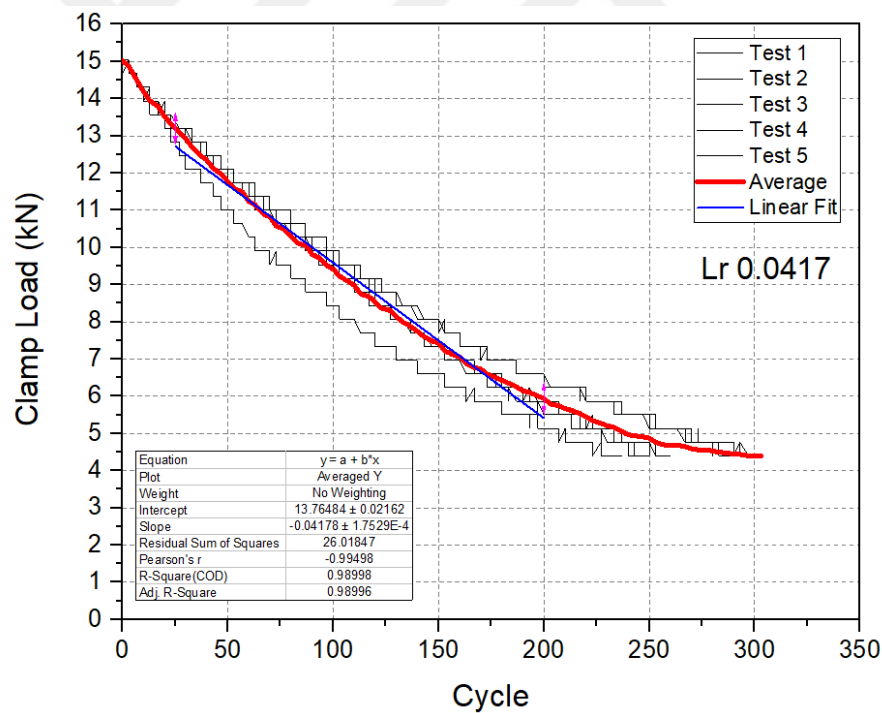


Figure 4.10 Clamp load cycle graph of M8 25mm clamp length bolt under 15kN

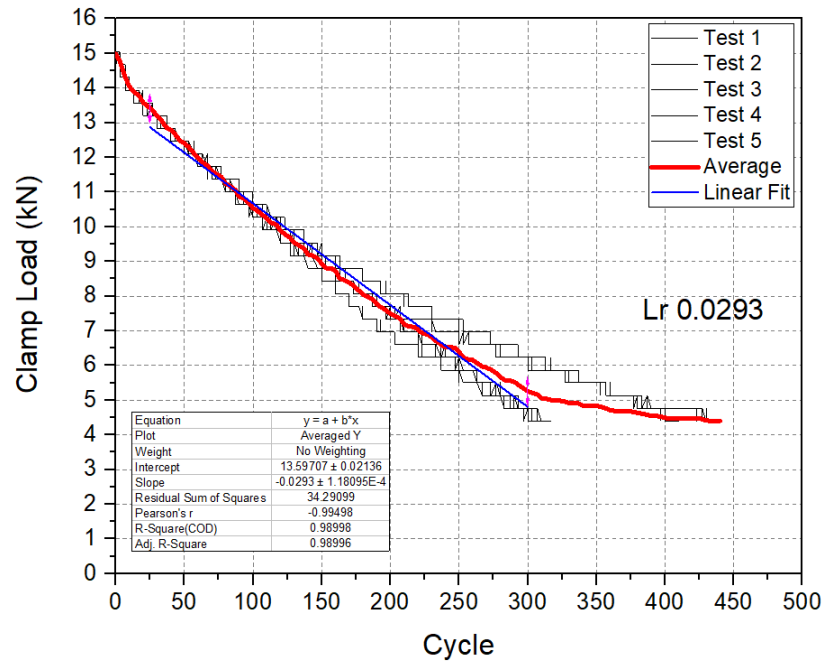


Figure 4.11 Clamp load cycle graph of M8 40mm clamp length bolt under 15kN

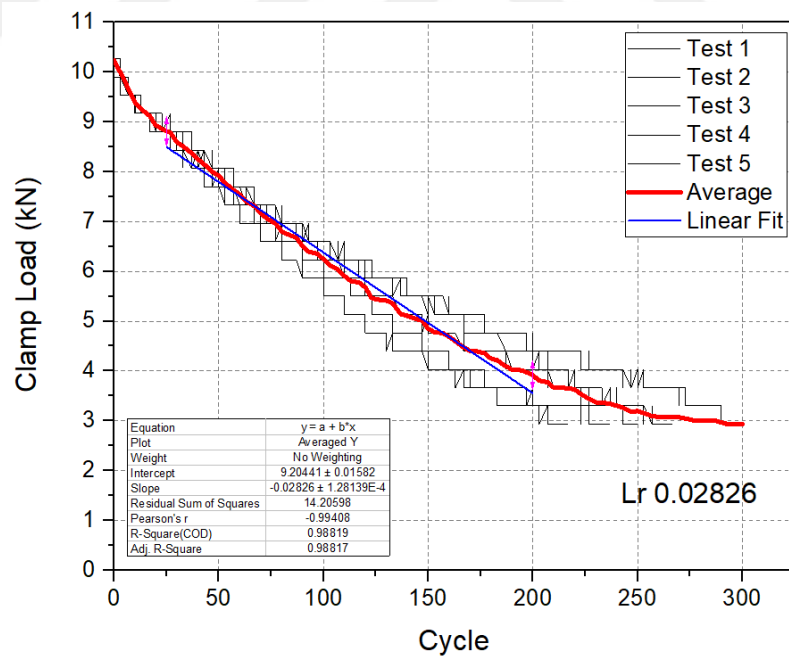


Figure 4.12 Clamp load cycle graph of M8 25mm clamp length bolt under 10kN

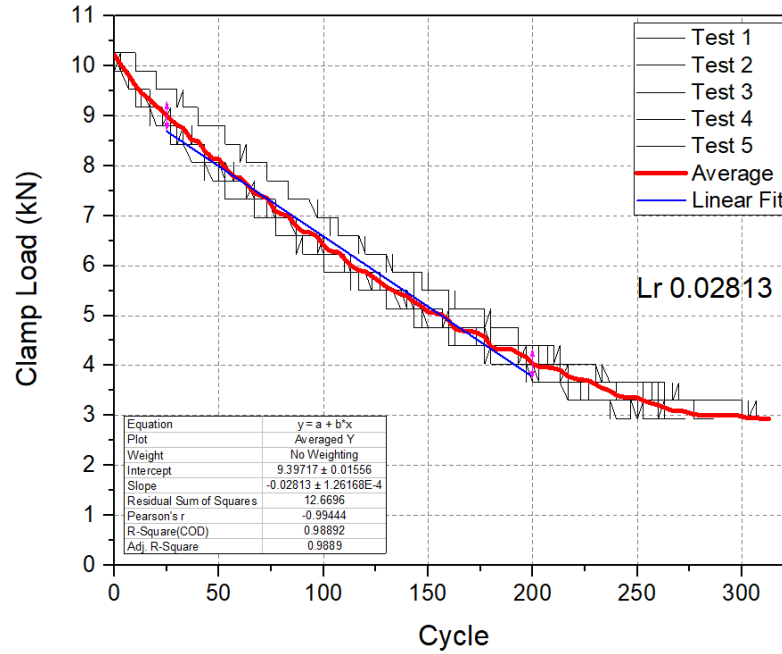


Figure 4.13 Clamp load cycle graph of M8 40mm clamp length bolt under 10kN

The data obtained for the comparison of 4 different test averages with each other were placed on a single graph and their behaviors were compared in Figure 4.14. Accordingly, although there is no obvious difference in terms of relaxation slopes, the clamp length factor has emerged as a more significant parameter at the 15 kN clamp load value. The reason for this was determined to be due to the fact that the dominant parameter is different at low clamp load values due to the lateral displacement value being too high.

Combined test approach was employed, considering all relevant elements. The primary objective of our testing is to ascertain the value of the 'loosening rate' (LR). The coefficient of friction parameter includes four distinct values. The values provided are 0.1, 0.12, 0.14, 0.16, and 0.18. The reference values for the clamp load are 10-15 kN and 20 kN. Tests were conducted at various values of 25, 30, 40, 50, and 60 mm to determine the clamp length. A total of 80 sets, including 400 tests, were conducted.

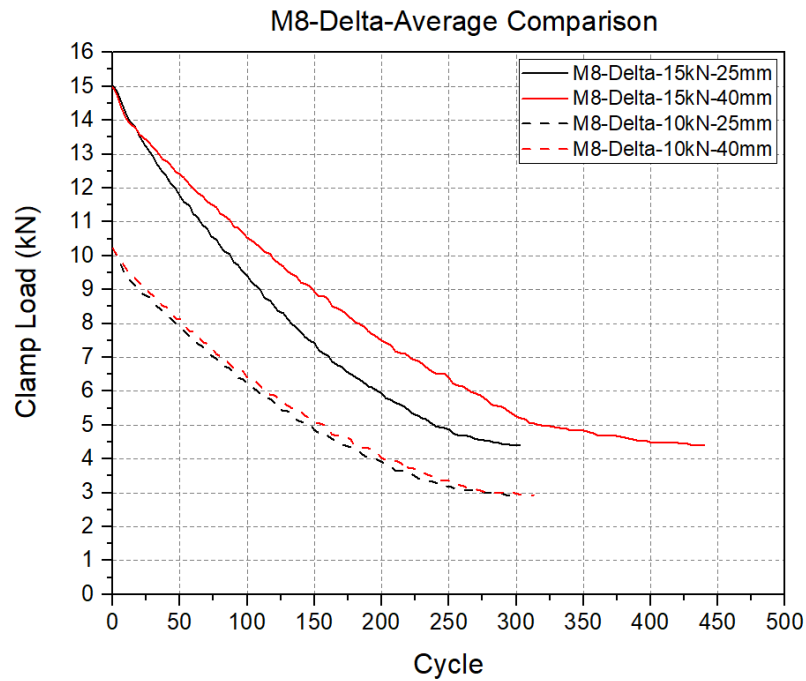


Figure 4.14 Clamp load cycle graph of 4 different combination

The parameters and LR values of the tests are shown in Table 4.1. The data that has been acquired will serve as an input for the predictive model throughout the next stage of the project. Hence, the loosening estimation model may be investigated by the use of LR values.

Table 4.1 Loosening rate data and parameters obtained from Junker vibration test.

Set	Clamp load [kN]	Clamp length [mm]	Friction coefficient	Lateral displacement [mm]	Loosening rate [kN/cycle]
1	20	25	0.12	0.45	0.01020
2	15	25	0.12	0.45	0.04120
3	10	25	0.12	0.45	0.02826
4	20	30	0.12	0.45	0.00774
5	15	30	0.12	0.45	0.01825
6	10	30	0.12	0.45	0.03032
7	20	40	0.12	0.45	0.00513
8	15	40	0.12	0.45	0.02931
9	10	40	0.12	0.45	0.02813
10	20	50	0.12	0.45	0.00392
11	15	50	0.12	0.45	0.00505
12	10	50	0.12	0.45	0.01258

Table 4.1 Continued

13	20	60	0.12	0.45	0.00015
14	15	60	0.12	0.45	0.00235
15	10	60	0.12	0.45	0.00478
16	20	25	0.14	0.45	0.00297
17	15	25	0.14	0.45	0.01529
18	10	25	0.14	0.45	0.03034
19	20	30	0.14	0.45	0.00176
20	15	30	0.14	0.45	0.01059
21	10	30	0.14	0.45	0.02138
22	20	40	0.14	0.45	0.00020
23	15	40	0.14	0.45	0.00511
24	10	40	0.14	0.45	0.01109
25	20	50	0.14	0.45	0.00002
26	15	50	0.14	0.45	0.00276
27	10	50	0.14	0.45	0.00677
28	20	60	0.14	0.45	0.00002
29	15	60	0.14	0.45	0.00043
30	10	60	0.14	0.45	0.00256
31	20	25	0.16	0.45	0.00180
32	15	25	0.16	0.45	0.01095
33	10	25	0.16	0.45	0.02584
34	20	30	0.16	0.45	0.00126
35	15	30	0.16	0.45	0.00792
36	10	30	0.16	0.45	0.01859
37	20	40	0.16	0.45	0.00200
38	15	40	0.16	0.45	0.00403
39	10	40	0.16	0.45	0.00995
40	20	50	0.16	0.45	0.00189
41	15	50	0.16	0.45	0.00219
42	10	50	0.16	0.45	0.00617
43	20	60	0.16	0.45	0.00101
44	15	60	0.16	0.45	0.00022
45	10	60	0.16	0.45	0.00232
46	20	25	0.18	0.45	0.00017
47	15	25	0.18	0.45	0.00668
48	10	25	0.18	0.45	0.02132
49	20	30	0.18	0.45	0.00016
50	15	30	0.18	0.45	0.00528
51	10	30	0.18	0.45	0.01577
52	20	40	0.18	0.45	0.00088
53	15	40	0.18	0.45	0.00297
54	10	40	0.18	0.45	0.00879

Table 4.1 Continued

55	20	50	0.18	0.45	0.00075
56	15	50	0.18	0.45	0.00164
57	10	50	0.18	0.45	0.00555
58	20	60	0.18	0.45	0.00013
59	15	60	0.18	0.45	0.00026
60	10	60	0.18	0.45	0.00208
61	10	55	0.1	0.45	0.00230
62	10	45	0.1	0.45	0.03307
63	10	30	0.1	0.45	0.03776
64	10	25	0.1	0.45	0.04423
65	10	15	0.1	0.45	0.06520
66	12.5	55	0.1	0.45	0.00615
67	12.5	45	0.1	0.45	0.03286
68	12.5	30	0.1	0.45	0.03609
69	12.5	25	0.1	0.45	0.04173
70	12.5	15	0.1	0.45	0.06009
71	15	55	0.1	0.45	0.00073
72	15	45	0.1	0.45	0.02723
73	15	30	0.1	0.45	0.03589
74	15	25	0.1	0.45	0.03735
75	15	15	0.1	0.45	0.05758
76	15.5	55	0.1	0.45	0.00062
77	15.5	45	0.1	0.45	0.02274
78	15.5	30	0.1	0.45	0.03370
79	15.5	25	0.1	0.45	0.01857
80	15.5	15	0.1	0.45	0.04058

4.3 Axial Fatigue Test Results

In the framework of fatigue testing, Wohler curves were initially derived for fasteners exhibiting varying average amplitudes. The fatigue curves are some input data that are specific to the load condition due to their dependence on the quality of the bolts. Moreover, the outcomes remain consistent across various head geometries. As a consequence, the test was predominantly conducted on bolts of M8 8.8 quality. Two distinct Wohler curves were acquired as part of the investigation. The Wohler curves presented here pertain to data in which the average load value, which is the clamp load, is 15 kN which is the roughly %70 percent equivalent of the proof load

value of a M8 8.8 coarse thread bolt. Wohler curve is illustrated in Figure 4.15. For a Wohler curve, an average of 25 to 30 fatigue testing needs to be conducted for a reliable outcome. Curves obtained are given in 3 different failure probability ranges which are %10, %50 and %90 respectively.

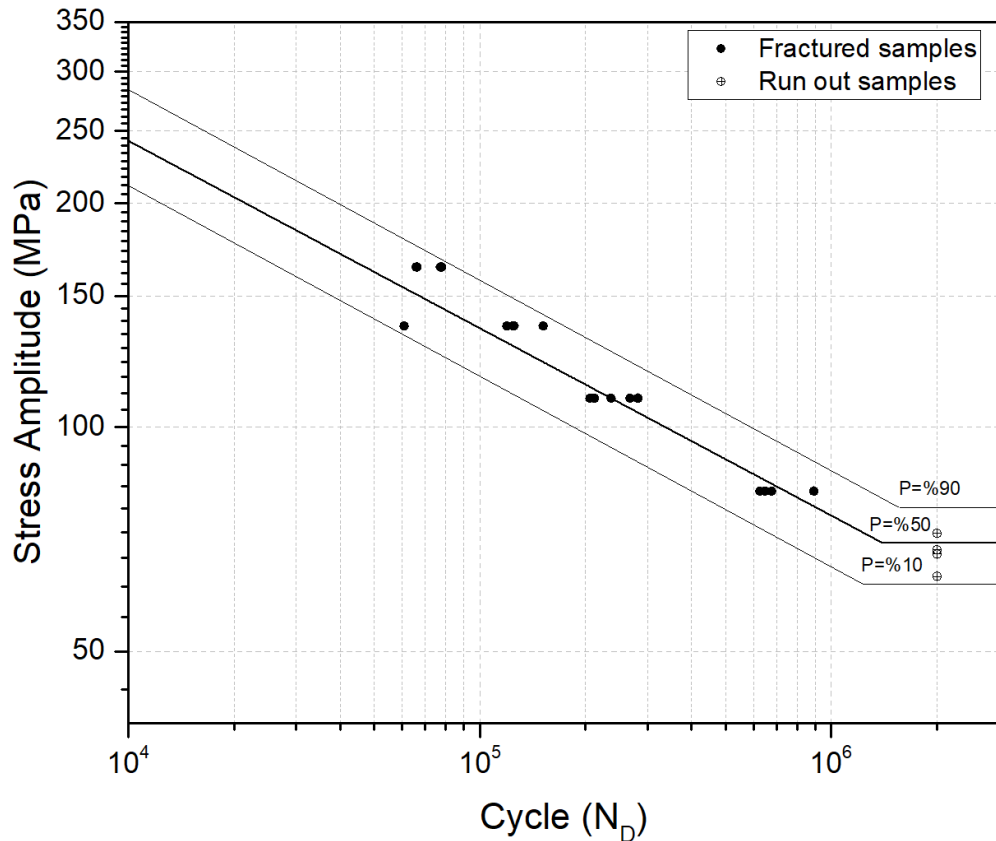


Figure 4.15. Wohler diagram for 15 kN mean load for M8 8.8 fastener taken from Zwick Roell test device

VDI 2230 (VDI, 2003) states that fatigue endurance limits for fasteners ranging from certain metric sizes and criteria follow a general equation. Distinction between thread rolled before (SV) and after (SG) heat treatments are made clear and included to the equation. Validity of both equations are valid unless the load ratio stays in a range of $0.3 < F_{sm}/F_{0.2} < 1$. A main equation to calculate endurance limits which the alternating life cycles are above $N_D = 2 \cdot 10^6$ is given in Equation 4.1 for thread rolled before heat treat and in Equation 4.2 for thread rolled after heat treat. Where F_{sm} is the

average bolt load which can be calculated based on Equation 4.3 in which F_{SAo} is the maximum bolt additional load, F_{SAu} is minimum bolt additional load and F_{Mzul} is the permissible assembly preload. Experimental tests conducted in this study does not includes thread rolling after heat treat. That is why the equation for thread rolling before heat treatment will be the one which we are going to use for calculation.

$$\sigma_{ASV} = 0,85 \left(\frac{150}{d} + 45 \right) \quad 4.1$$

$$\sigma_{ASG} = \left(2 - \frac{F_{Sm}}{F_{0,2min}} \right) \cdot \sigma_{ASV} \quad 4.2$$

$$F_{Sm} = \frac{F_{SAo} + F_{SAu}}{2} + F_{Mzul} \quad 4.3$$

For alternating load cycles which sits in the finite range, above equations are not applicable. For this purpose, an equation to predict stress amplitudes were given as below in Equation 4.4 for thread roll before and Equation 4.5 for thread roll after heat treat. Where σ_{AZSV} is the stress amplitude for fatigue strength, σ_{ASV} stress amplitude at endurance limit, N_z number of alternating cycles during loading within the fatigue strength range and N_D number of alternating stress during continues loading. In order to have a reliable and precise fatigue life prediction, an experimental test is advised based on the samples which will be used according to VDI 2230. Experimentally derived Basquin relationship is given in Equation 4.6 in terms of amplitude load. This was later derived into a stress value with multiplying with the cross-sectional stress area of the threaded section (ISO 898-1, 2004), which is 36.6 mm² for M8 coarse thread.

$$\sigma_{AZSV} = \sigma_{ASV} \left(\frac{N_D}{N_Z} \right)^{\frac{1}{3}} \quad 4.4$$

$$\sigma_{AZSG} = \sigma_{ASG} \left(\frac{N_D}{N_Z} \right)^{\frac{1}{6}} \quad 4.5$$

$$F(P) = F_A \left(\frac{N_K}{N} \right)^{-\frac{1}{k}} \quad 4.6$$

Experimentally obtained Basquin equations in 3 different probability range are given in Table 4.2. In order to evaluate the outcome of the tested specimens in terms of their fatigue responses, Wohler curves obtained experimentally and theoretically were plotted in the same log-log plot and shared in Figure 4.16.

Table 4.2 Experimental fatigue test Basquin equations and their long-life fatigue limits

Type of the curve	Failure probability P	HCF equation	Nk	LLF: FA
Experimental	%10	LogN=7.06626-3.976038 x LogF	1038804	1.84
Experimental	%50	LogN=7.25806-3.976038 x LogF	1396839	1.91
Experimental	%90	LogN=7.44986-3.976038 x LogF	1887948	1.97
Theoretical	%50	LogN=7.44986-3.976038 x LogF	2000000	1.49

From the obtained results it can be seen that there is a slight variation on the outcomes of the fatigue test. This is mainly because of the test conditions and parameters which effects the fatigue outcomes. While VDI stays on the more conservative side, actual experimental results show samples real behaviors to the load conditions. In such cases where the fatigue predictions play a key role, equations obtained from experimental outcomes deliver a more reliable prediction.

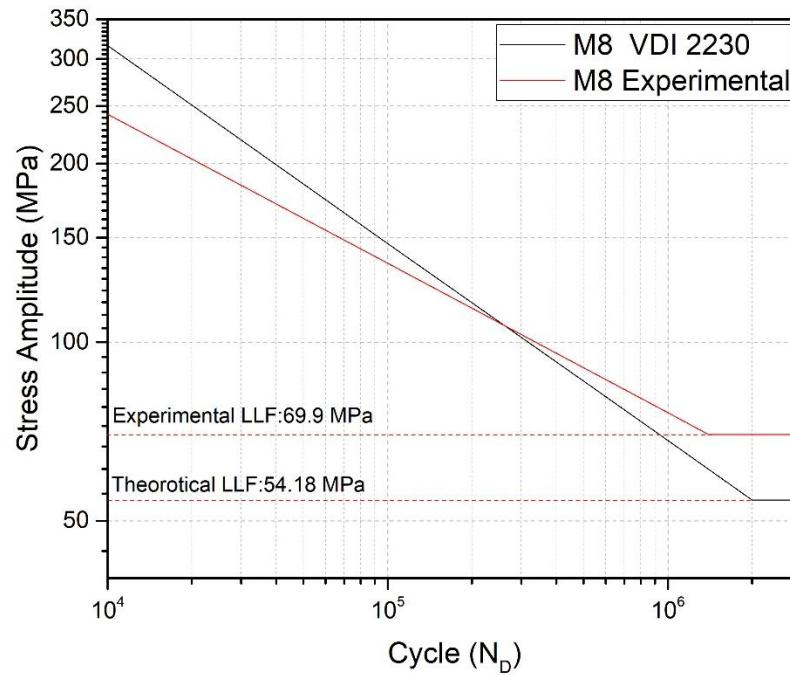


Figure 4.16 Wohler curve of M8 fastener experimentally and theoretically

4.4 Simulation Based Tightening Model Including Friction Changes

In order to establish a simulation model for loosening prediction, there is a need to have a reliable tightening simulation model. For this purpose, a tightening simulation model was developed with using experimental tightening results as a baseline and for the verification of the model. Despite the existence of several software and models for fastener simulation, the current approaches in the industry primarily prioritize maintaining of a consistent clamping load at a macroscopic level. However, most models lack a comprehensive consideration of friction effects, especially within the thread region. This circumstance gives rise to an inadequate, particularly in instances where the coefficient of friction through the tread and bearing region has significance. Despite extensive research has been conducted so far, there is a notable absence of studies focused on the comprehensive modeling of fastener behavior in response to repetitive usage. In the context of this research, a variation in the friction coefficient due to re-tightening will be employed as an input parameter in the numerical simulation model. Consequently, the numerical modeling will enable the

representation of the torque-clamp load connection that arises during repetitive tightening.

To investigate the changes of the friction coefficient caused by repeated use of the fastener, a first experimental study was conducted to assess the change of the friction coefficient. In the context of the experimental study, the friction coefficient was determined by using the ISO 16047 (ISO 16047, 2013) standard often employed in the industry. The friction coefficients provided in the standard were determined using the formulation presented in Equation 4.7. In this context, the symbol μ_{th} represents the friction coefficient in the threaded region, while μ_b represents the friction coefficient in the underhead bearing region. T denotes the torque, F represents the clamp load, P represents the pitch, d_2 corresponds to the section circle diameter, while D_o and d_h denote the inner and outer contact diameter values in the underhead region.

$$T = F \times \left[\frac{1}{2} \times \frac{P + 1.154 \times \pi \times \mu_{th} \times d_2}{\pi - 1.154 \times \mu_{th} \times \frac{P}{d_2}} + \mu_b \times \frac{D_o + d_h}{4} \right] \quad 4.7$$

The experiment involved conducting 25 tests in 5 sets to examine the variations in friction coefficients of bolts during re-tightening. The friction coefficient estimation method outlined in the ISO 16047 specification was employed for this purpose. During the experimental trials, a tightening speed of 50 revolutions per minute (RPM) was employed. The load sensors were utilized to measure and record the locking load and torque data. During the experimental trials, the load value required for target locking was determined to be 18.24 kN. This value was obtained by using 60% of the "proof load" value specified for M8 coarse thread bolts with a 10.9 grade according to the ISO 898/1 (ISO 898-1, 2004) standard. Subsequently, tests were carried out by adjusting the obtained rotation angles to 60° starting from a settlement value of 1 Nm. The ANSYS Transient analysis module was utilized in the context of the numerical simulation study. The 3D design of bolt, nut thread profiles, and under-head contact plate was constructed using CATIA V5 within the framework of the model.

Afterwards, the resulting models were exported as a STEP file and imported into the ANSYS module.

4.4.1 Model Development with Boundary Conditions

Several refinements were implemented during the process of meshing, with particular focus on the contact regions. A decision was made to utilize a tetrahedral mesh structure throughout the meshing process. In the first model, a 3mm mesh was chosen, however, in order to improve the surface quality in the contact regions, a reduction to 2mm was implemented on the contact surfaces. The area covering the interface between the bolt head and plate was divided into 60 identical segments, followed by the use of surface meshing techniques. The overall quantity of meshes in the system is recorded as 57482, whilst the number of nodes is documented as 95582. The visual representation of the mesh model is depicted in Figure 4.17.

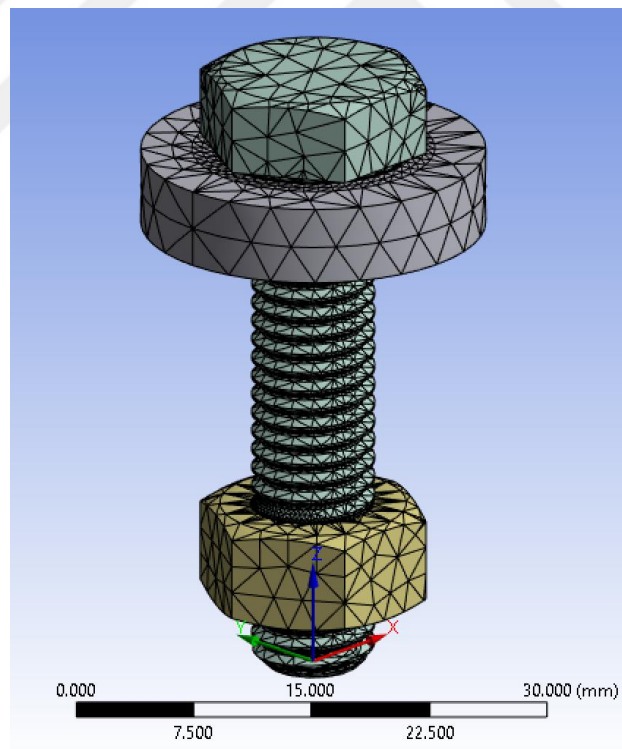


Figure 4.17 Mesh model of the bolt, plate and nut members

Friction coefficient value is a crucial element for ensuring accurate clamping load results in the simulation model. The friction coefficient in fasteners is often categorized into two components: thread friction and underhead friction. Accurately inputting these numbers is crucial for obtaining reliable simulation results. Hence, the contact regions between the thread and bearing surface were designated as "frictional," and the corresponding friction coefficients were determined based on the initial experimental results. The solver formulation selected for this study was the "Augmented Lagrange" model.

In the transient analysis settings, the main model was entered as consisting of 2 time intervals. It was aimed to increase the accuracy of the simulation by entering the sub-step numbers on a time basis. While the first step is composed of 5 seconds and 40 sub-steps, the second 5-second period is similarly composed of 40 steps. In total, the model contains 80 sub-steps.

The angular rotation of the fastener with regard to the Z axis was inputted as tabular data in the simulation model. The angular displacement from the head of the bolt was intended to achieve a 60° rotation at the conclusion of the tenth second, in order to maintain a constant clamping load output irrespective of the output torque. The upper plate was defined utilizing a frictionless support model with a single degree of freedom. In pursuit of this objective, lateral surfaces were chosen and it was ensured that the plate possessed degrees of freedom exclusively along the Z axis. The upper surface of the nut was represented as a fixed support in the nut region, and any rotation of the nut was prevented. Subsequently, the total torque value was obtained by using the bolt head as a reference when measuring the moment value. The clamping load data were obtained in Newton using the fixed support point on the bottom surface of the nut as the reference point during the measurement process. Figure 4.18 presents the visual representation of the boundary constraints of the model.

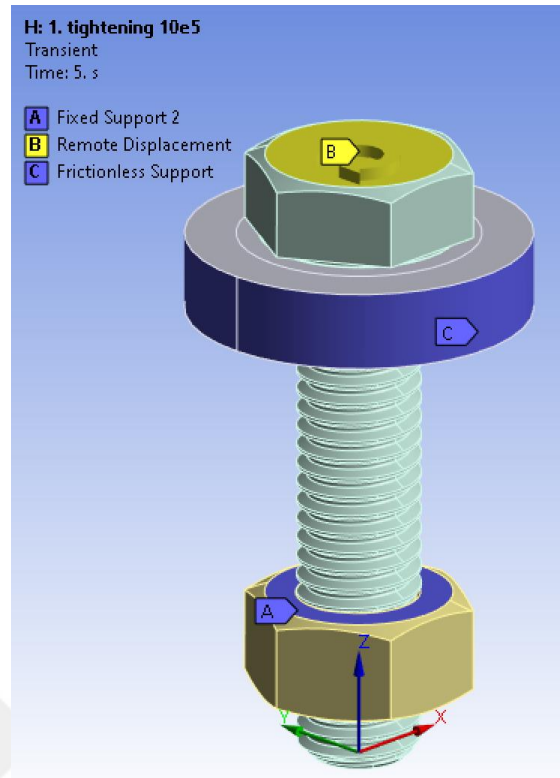


Figure 4.18 Boundary conditions for the tightening model

An additional critical factor that can influence the behavior of the torque clamping load is the rigidity value of the fastened component. The clamp force output is generated by axial elongation in the direction of the bolt axis; therefore, the axial elongation is directly influenced by the rigidity of the tightened components. The experimental and numerical simulations have highlighted the criticality of accurate contact modeling and the spring constant effect, as they have a direct effect on the elongation of the bolt (Ganeshmurthy & Nassar, 2014; Zadoks & Kokatam, 2001). In order to determine this value, simulation output with the torque-angle-clamp load graph gathered during the initial tightening trials and validated it with experimental data. The assembly rigidity was calculated theoretically and the simulation model was calibrated using experimental results during the development of the reference model. In the simulation, this value represents the axial spring along the clamping length region. Figure 4.19 illustrates the spring input values and the visual representation of the model.

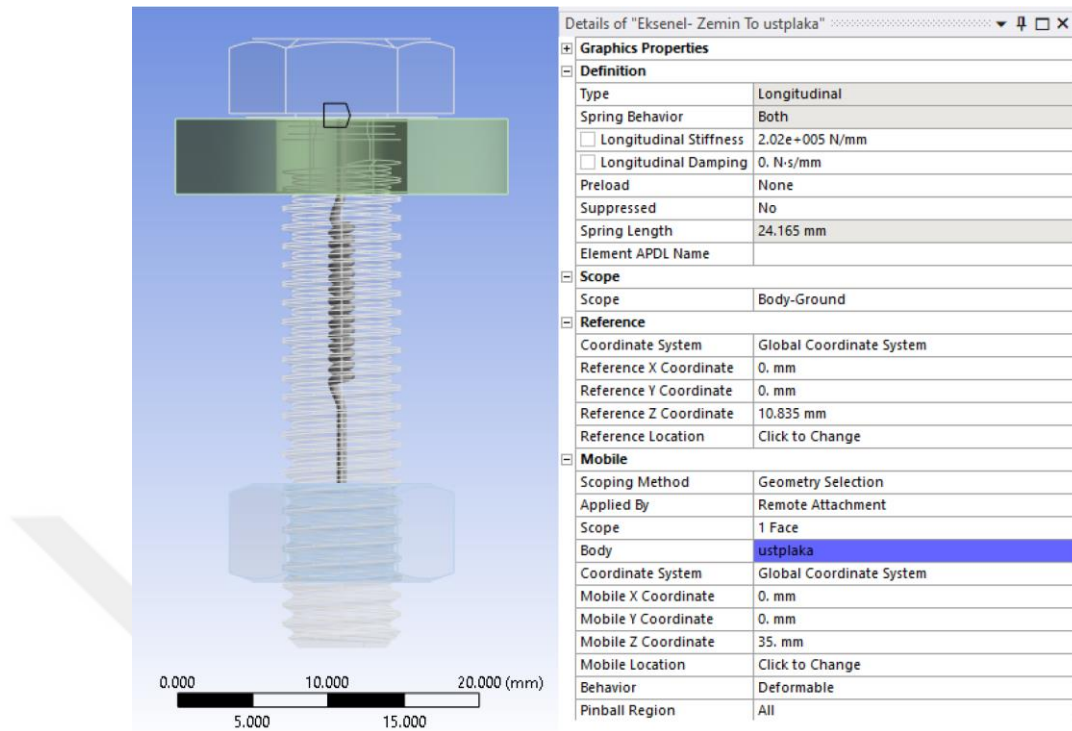


Figure 4.19 spring model inputs for joint stiffness

4.4.2 Friction Model

A sliding distance-based friction model was developed utilizing experimental data in order to provide clamp load-torque relationship during repeated tightening, which was the primary objective of this section. In the context of the model, unique friction parameters were formulated for the thread and underhead bearing regions due to the different sliding distance values observed in these areas. The sliding distance was calculated during the development of the underhead friction model using the effective underhead contact diameter at 60 degrees of rotation. This approach was chosen due to the fact that the linear slip distance varies with distance from or proximity to the bolt axis. Many studies have been carried out in the literature on the importance and effect of effective bearing contact diameter. A study was carried out to assess the effect of bearing pressure distribution on various surface conditions and hole diameter ratios using a contact mechanics-based model. In addition to this, the torque output influences the effective contact value r_b , which is determined regardless of the rotational speed. This value is affected by the inner and outer contact radii. The

resultant output from the previously mentioned investigation is represented by Equation 4.8. In this context, r_{b-max} represents the outer radius of maximum contact, r_{b-min} denotes the inner radius of the minimum contact, $P_{b(r)}$ represents the surface pressure at radius r , and r signifies the axial distance to the main axis. Similar research were carried out by deriving equations from the distribution of surface pressure under various case conditions.

$$r_b = \frac{\int_{r_{b-min}}^{r_{b-max}} P_b(r) r^2 dr}{\int_{r_{b-min}}^{r_{b-max}} P_b(r) r dr} \quad 4.8$$

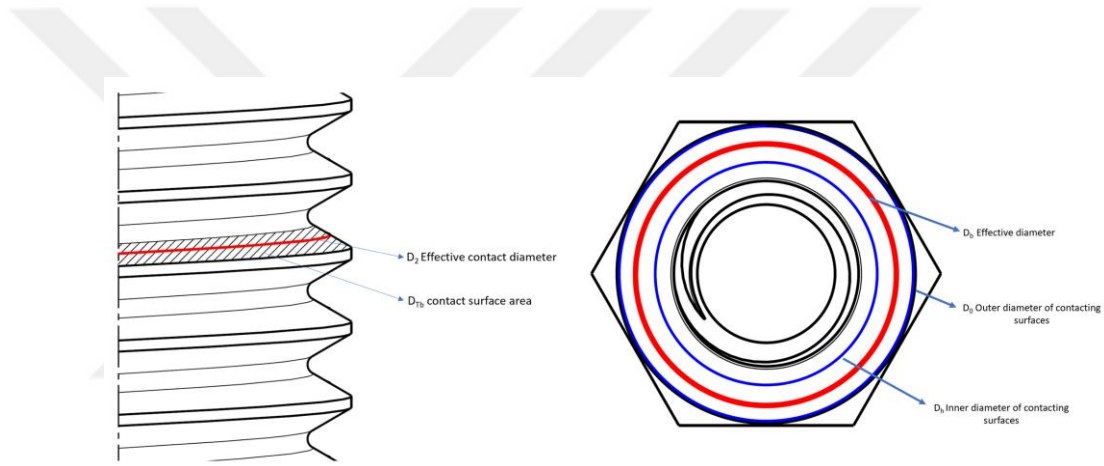
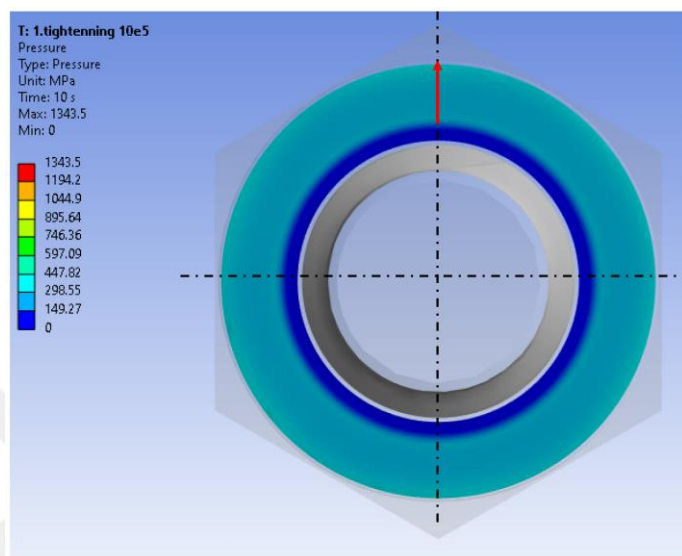


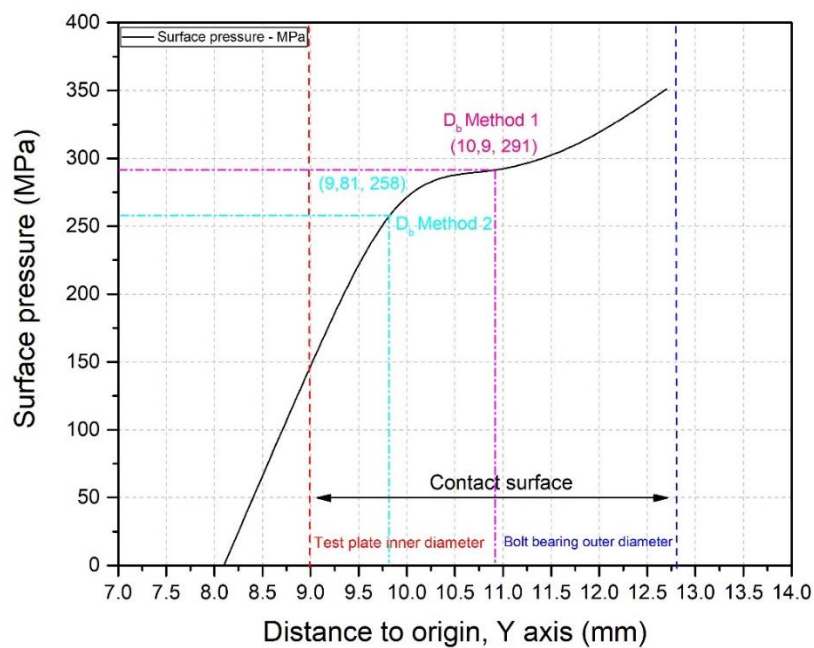
Figure 4.20 Effective contact diameters and contact areas representation

Studies on the effect of the effective diameter of the thread contact area have determined that the diameter change will remain limited due to the helix angle and the contact surface, and therefore its effect on the friction coefficient will be lower than the effect of the effective diameter under the head. Accordingly, the effect of the effective thread radius and the section effective diameter on the friction coefficient was determined to be less than 1% (Nassar, Matin, et al., 2005). In the light of the aforementioned findings, sliding distances were determined by taking the effective diameter of the thread surface as the D_2 effective circle diameter. Based on the findings of the investigation that calculated the effective radius by comparing two distinct methods and the effective bearing pressure distribution, it was concluded that the suggested geometric center corresponded suitably to both the obtained output results

and the pressure distribution. The surface pressure distribution is illustrated in Figure 4.21 (a) for this purpose, while the x-axis surface pressure change is given in Figure 4.21 (b).



(a)



(b)

Figure 4.21 Variation of under-head contact surface pressures along the y-axis (a) and representation of effective diameters according to 2 different methods (b)

In order to match the model with the friction coefficient, firstly, the results of the friction coefficient change under repeated tightening were obtained experimentally. Accordingly, μ_{th} is the thread friction coefficient, μ_b is the under head friction coefficient, and μ_{tot} is the total friction value. As can be seen from the results in Table 4.3 and Figure 4.22, it was observed that there was a gradual increase in the friction coefficients after the first tightening.

According to the repeated tightening results, although the change trend in the friction coefficient is similar to the thread and under-head friction value, it is seen that the threaded region friction coefficient value is lower than the under head region. This reveals that the friction values of the under-head and threaded areas will make a big difference during modelling.

Table 4.3 Experimental friction coefficient test results, mean of 5 test

Repeated tightening	μ_{th}	Standart deviation	μ_b	Standart deviation	μ_{total}	Standart deviation
1.	0.098	0.011	0.134	0.009	0.113	0.011
2.	0.113	0.014	0.152	0.010	0.132	0.011
3.	0.124	0.019	0.168	0.013	0.143	0.015
4.	0.129	0.022	0.173	0.019	0.149	0.020
5.	0.137	0.026	0.181	0.023	0.155	0.024

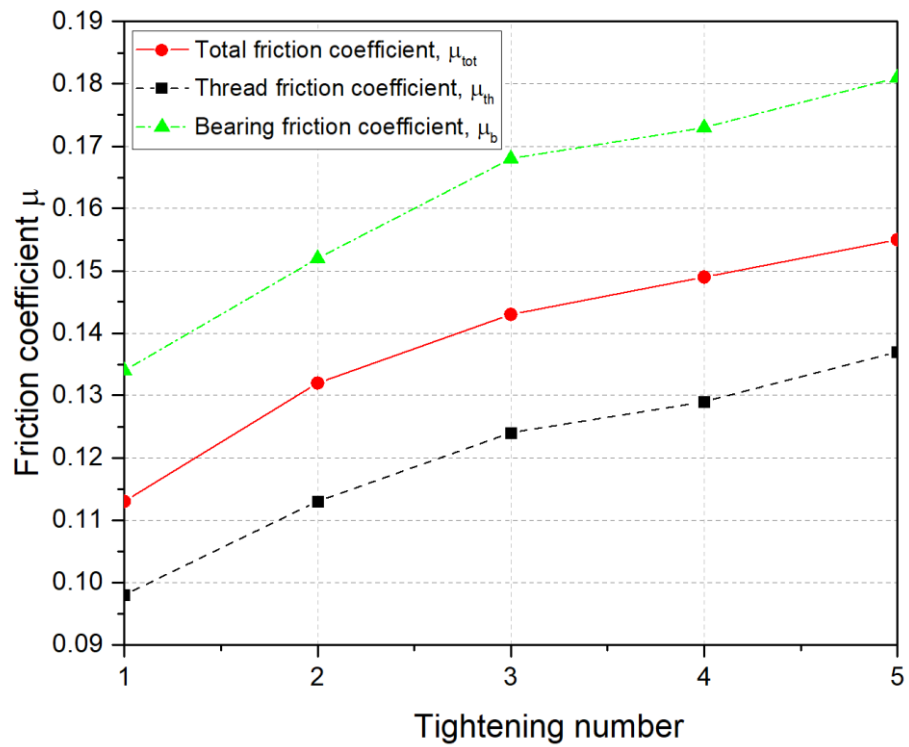


Figure 4.22 Friction changes under repeated tightening

During the process of converting the friction coefficient acquired from the number of tightening into a parameter that can be inputted into the simulation model, cumulative sliding distance data indicating the change in the friction coefficient of the "SLDI"-based model were developed. This was possible because sliding distances and rotation angles are directly proportional. For the purpose of determining the sliding distances under the bearing surface and in the threaded region during each tightening, the method described in section 2.3 was applied. Consequently, the cumulative sliding distance values of the thread surface and under head up until the fifth tightening are displayed in Figure 4.23. The obtained data indicates that the sliding distances between the surface of the thread and the bearing surfaces are significantly different. This circumstance demonstrates that each region requires a unique sliding distance-friction coefficient model.

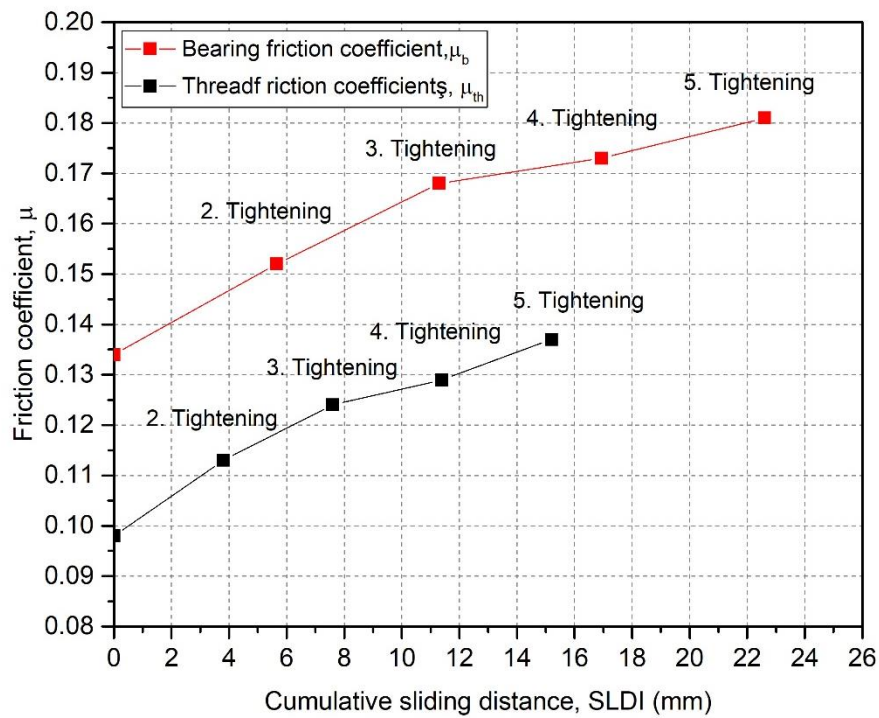
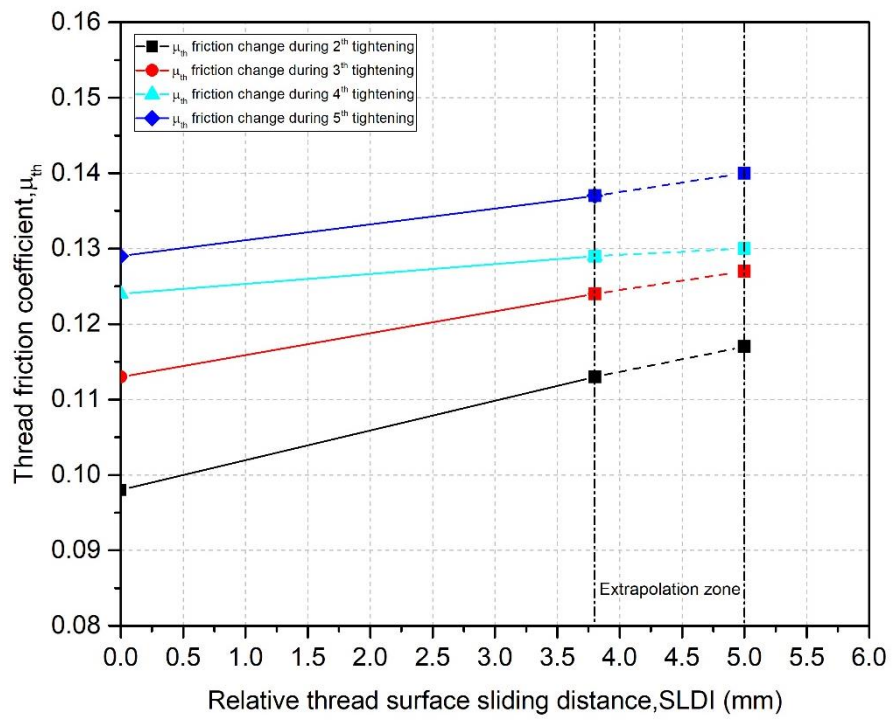
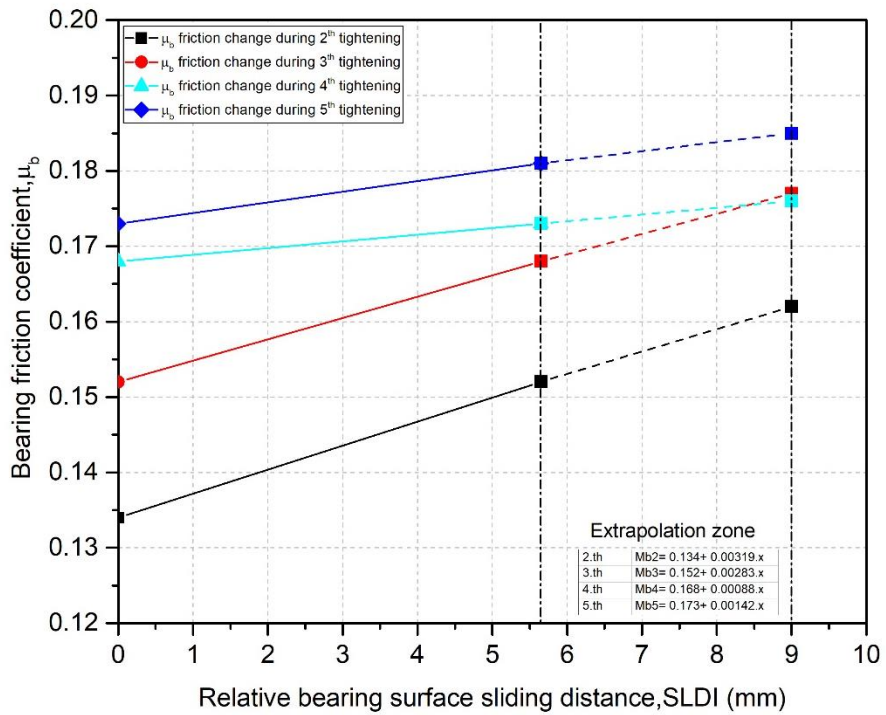


Figure 4.23 Cumulative sliding distance versus friction coefficient under repeated tightening

To determine the behavior of each simulation model at the appropriate number of repeated tightening, the relative sliding distance was utilized to calculate the changes in the friction coefficients of the under head and thread areas while holding the sliding distances constant for each tightening value. The model incorporates the linear change in friction coefficient during tightening in accordance with the sliding distance. As a result, Figure 4.24 presents sliding distance-based diagrams derived from experimental data.



(a)



(b)

Figure 4.24. Friction coefficient change based on relative sliding distance during tightening, thread (a), under head (b)

4.4.3 Tightening Simulation Outcomes

The spring stiffness of the numerical simulation model was validated in the initial phase, and the experimental test system's stiffness value was incorporated into the simulation model by comparing it to the data obtained from the torque angle graph during the first tightening. In this particular context, the initial simulation was executed with a value of 120000 N/mm and was designated as model 1. To validate the torque angle graph generated by model 1 against the experimental data, model 2 was constructed with a value of 300000 N/mm. The experimental data input coefficient was derived by interpolating the angle difference between the experimental model and the two models; this value was subsequently utilized as the input for model 3. Following this, model 3 was constructed utilizing the value 202000 N/mm. By correlating experimental data with the rotation angle, output torque, and resulting clamp load values, the accuracy of the primary stiffness model was validated. The dataset comprising the outcomes of the simulation and experiments is presented in Figure 4.25.

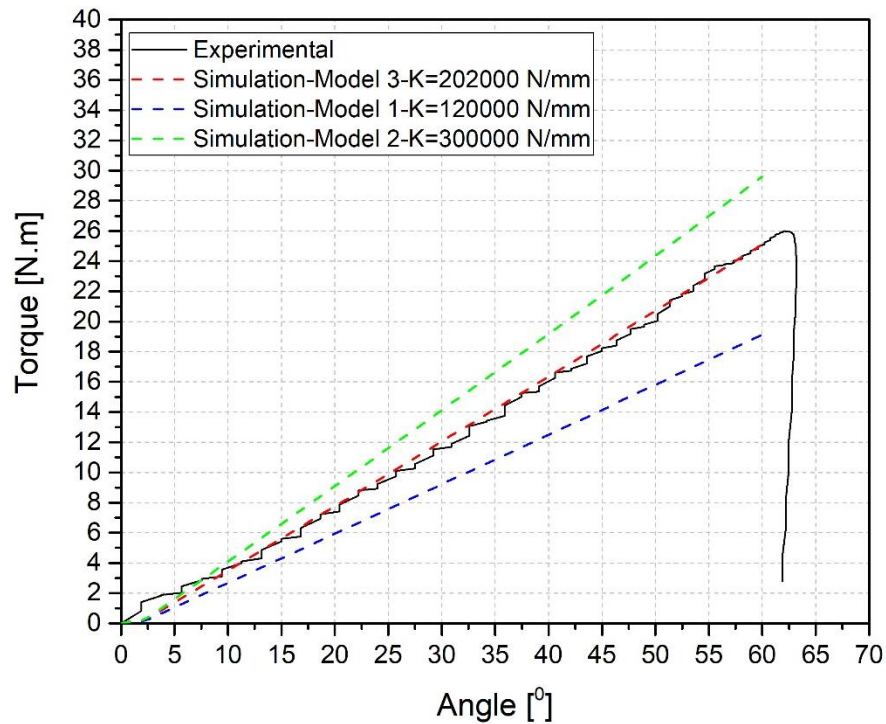


Figure 4.25. Torque angle values at different spring constants

In order to integrate the friction coefficient model, which was our objective, into the simulation, two distinct inputs were established using the APDL code for the simulation model and the sliding distance (SLDI) value for the thread and bearing areas, respectively, after the primary model had been designed. The applicable region's friction coefficient can be entered linearly based on the sliding distance using APDL code. Thus, end interpolation was avoided by entering the under-head friction coefficient for the second tightening as 0.134 when SLDI was 0 mm, 0.152 when SLDI was 5.65 mm, and 0.162 when SLDI was 9 mm, in accordance with the code provided in Figure 4.26.

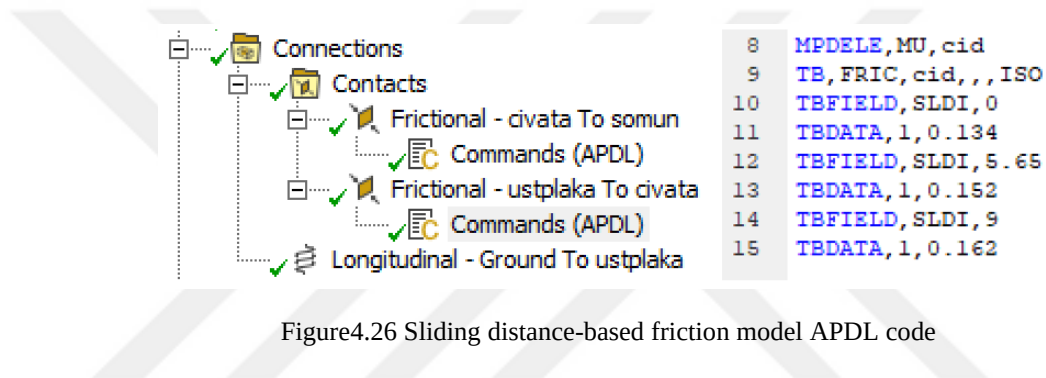


Figure4.26 Sliding distance-based friction model APDL code

The same input was created separately for each simulation model, the friction coefficient values were integrated into the model and the simulations were run. As a result of the simulation, it was observed that there was an increase in torque values depending on the amount of tightening's at the same axial force values, as expected. The data obtained is presented graphically in Figure 4.27. Findings: It was determined that there was a difference of 7.5 Nm between the 1st and 5th tightening.

Following further examination of the simulation model's outputs, it was determined that the predicted von Mises equivalent stress distribution was in a predicted range. Thus, the initial three engaged thread regions that make contact with the nut threaded region experience the greatest stresses. The distribution of tension during tightening is illustrated in Figure 4.28.

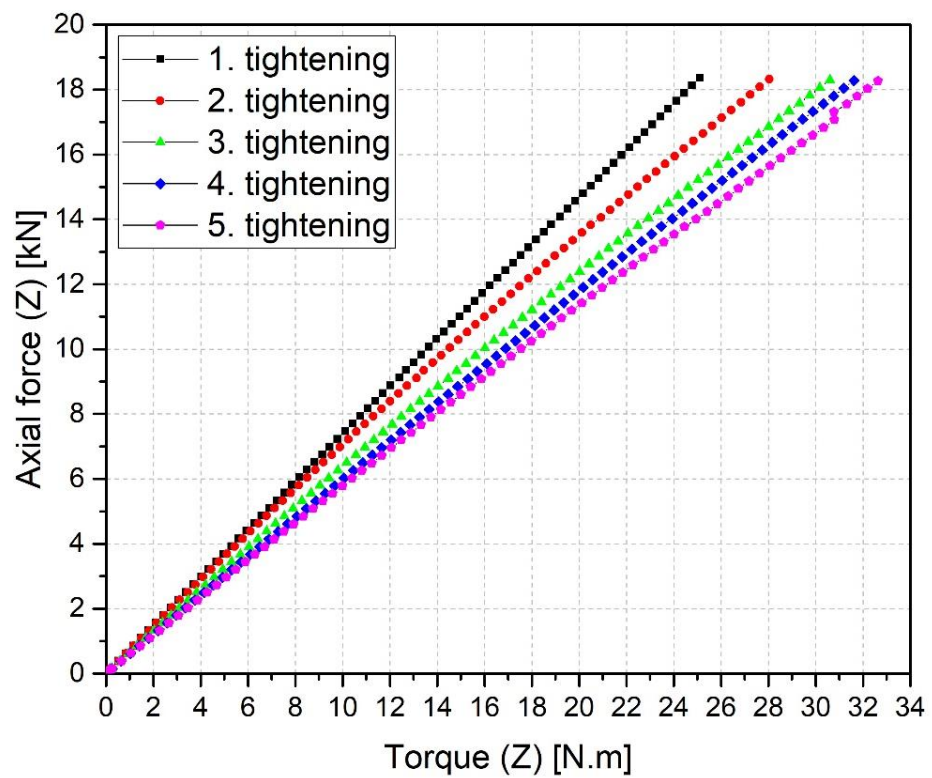


Figure 4.27 Simulation output of axial force-torque change based on the number of repeated tightening

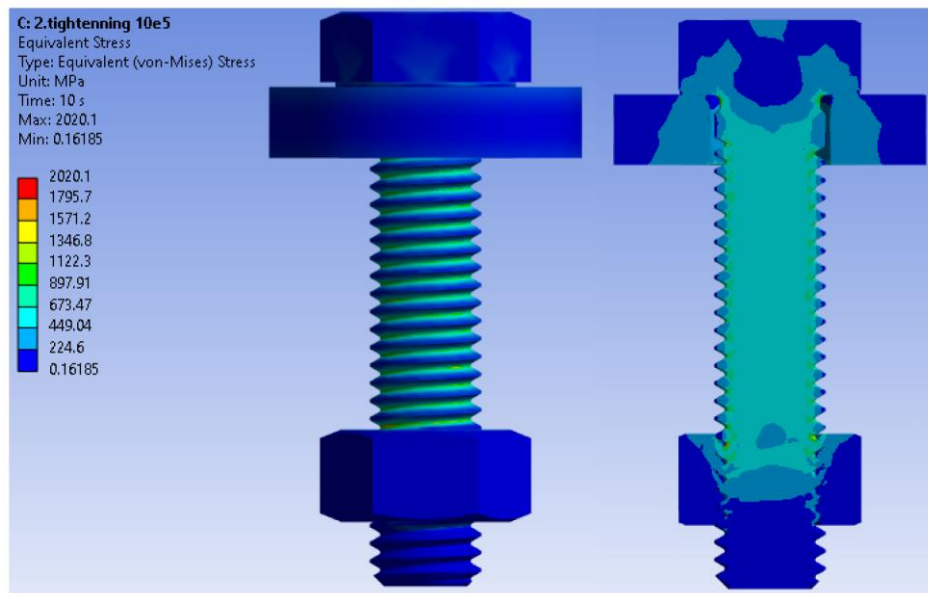


Figure 4.28 Von Mises equivalent stress distribution

When the obtained outputs are examined in detail, it is understood that the sliding distances in the surface contact areas are very different, as expected, and therefore the importance of the SLDI-based friction model is more apparent. A similar situation arises when the contact surfaces of real test samples are examined. As seen in Figure 4.29, it is seen that the slippage in the regions close to the outer diameter of the bottom of the skull is much higher.

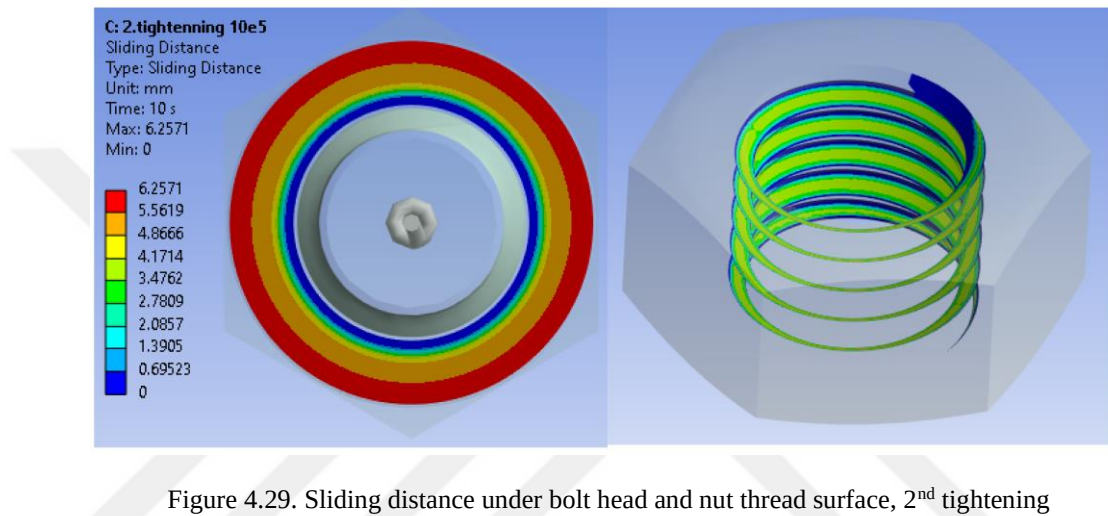


Figure 4.29. Sliding distance under bolt head and nut thread surface, 2nd tightening

Although experimental data supported in the development of the primary model, Table 4.4 provides the error margins for the friction coefficient assumed to be constant and the slip-based friction coefficient change model discussed in the section.

Table 4.4 Error percentage values

Model type	Torque (Nm)	Clamp load (kN)	Error %
Experimental fifth tightening	31.2	18.2	----
No friction change	25.1	18.3	19.5%
Head and thread friction with same value	33.7	18.2	8.01%
SLDI model	32.6	18.2	4.48%

4.5 Simulation Based Vibration Model Including Friction Changes

With the development of the tightening simulation model with contact interfaces, it is now possible to develop a simulation model for vibrational loosening under transvers loading conditions. For this purpose, again a transient analysis was used based on time inputs which correspond to our cycle inputs. A similar spring constant-based approach has been used to create a simulation model. Model was divided into 2 main steps. First step is focusing on the tightening operation where the main clamp load is generated by rotation of the bolt. In the second step, a transvers displacement in the top plate in Y axis was applied to simulate to plate cyclic displacements. Transvers displacement was set to ± 0.45 mm which the test setup load condition during the experimental studies. Loading case was represented in Figure 4.30 for the first 10 cycle.

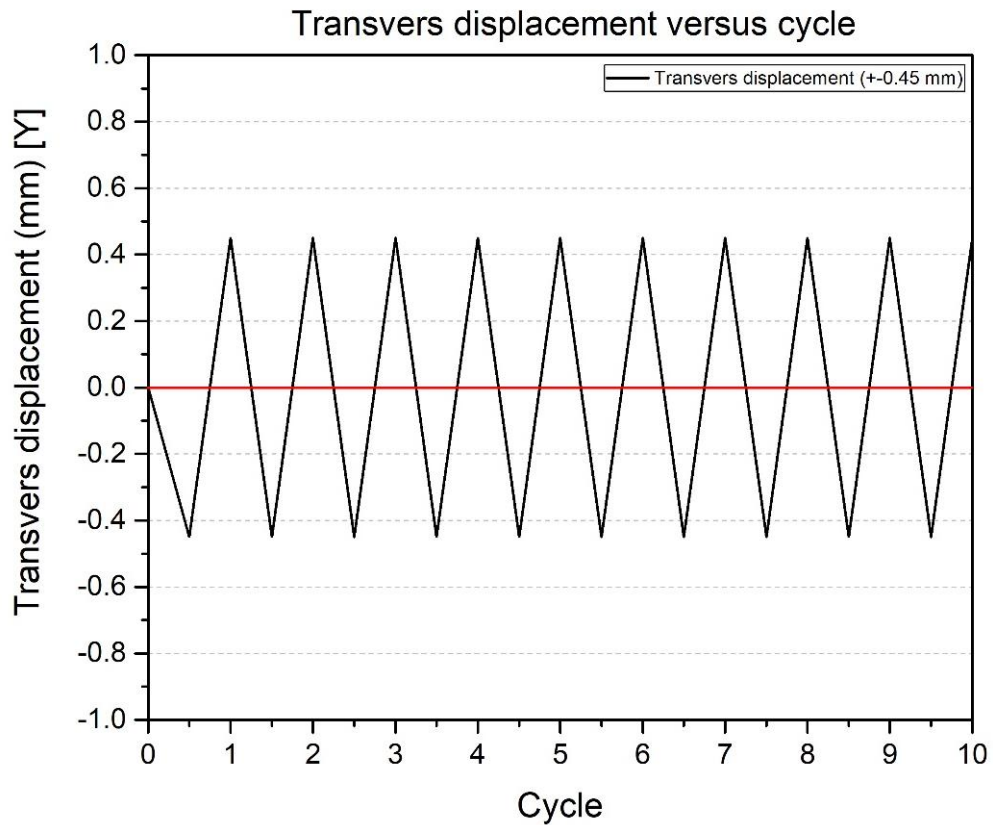


Figure 4.30 Transvers loading condition in terms of displacement in Y axis

One additional input in the boundary condition was the limitation of the plate rotation and movement in X axis. This was determined by entering a zero-rotation degree during the simulation to make the top plate to move only in Y axis. Displacement was set by remote displacement boundary condition in the software by entering tabulated data which corresponds to the cyclic displacement based on sub steps. Remote displacement was applied into side section of the movable top plate as shown in the Figure 4.31.

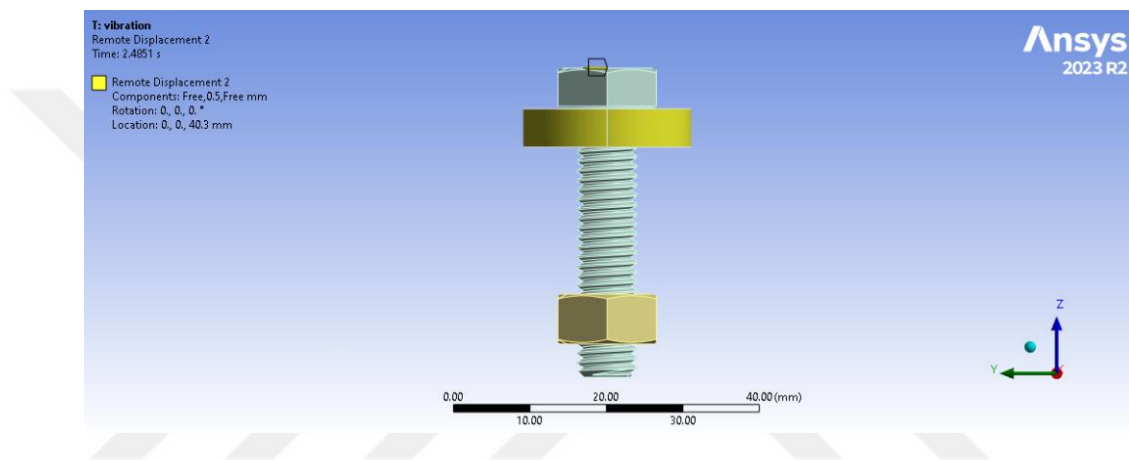


Figure 4.31 Remote displacement application on the moving plate

Model was set as previously stated to 2 main steps in transient solver. Total step count is set to 500 in which each step corresponds to 0.5 second. First step consists of 5 seconds and the vibrational loading step which is the second main step consist of 495-time step. First step is focusing on the tightening step which is targeted for the first 5 seconds in simulation. First load steps do not include any transversal displacement while the load case is only 60° rotations of the bolt head only in Z axis. This delivers a clamp load as discussed in detail in the previous sections. On the second main step, the transversal displacement load case starts to act starting from the end of 5th second. A zoomed in section for the load case in simulation software was given in Figure 4.32 where the figure shows the start of transvers loading case at fifth second and a total run of cyclic loading condition till 18th second in a zoomed in section. A load case for both boundary conditions and steps are given in Table 4.5 where Y, is the transvers displacement in Y axis and R_x , R_y , R_z are the rotational displacement in

their respective axis with a unit of degree. For our load condition, we do not have any rotational movement in R_x and R_y axis, while only rotation happens in R_z for 60° thru a period of 5 second in total.

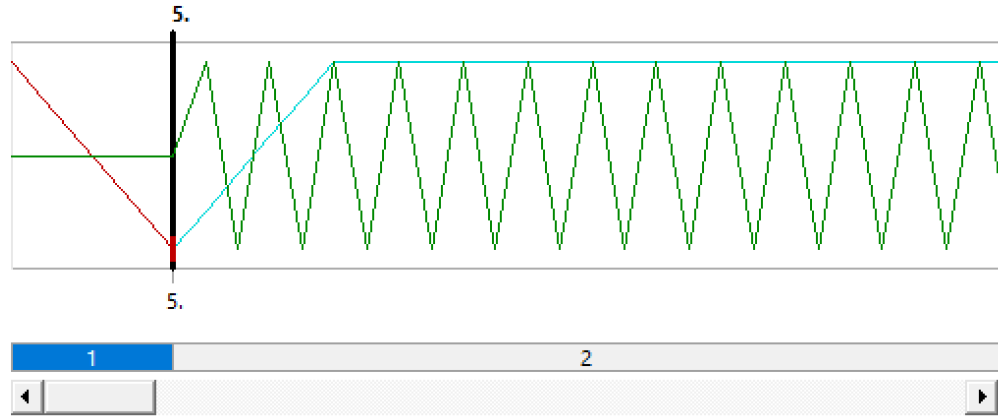


Figure 4.32 Load case based on main steps and sub steps in ANSYS based on the input values

Table 4.5 Load conditions based on time steps

Steps	Time [s]	Y [mm]	R_x [°]	R_y [°]	R_z [°]
1	0	0	0	0	0
1	5	0	0	0	60
2	6	0.45	0	0	0
2	7	-0.45	0	0	0
2	8	0.45	0	0	0
2	9	-0.45	0	0	0
...	...	...	...	...	...
...	...	...	...	...	...
2	250	0.45	0	0	0

In order to reduce the compute timing, a decision was given to limit the cycle count by 250 cycle due to the extensive length of calculation. Due to the nature of low cycle

loosening under transversal loads, 250 cycles will be suitable to determine the trend of loosening for the linear curve fit in order to obtain loosening rate values. Results for a load case given in the above section with a clamp load target of 18kN after tightening and $\pm 0.45\text{mm}$ displacement with a clamp length of 25 equivalent stress outputs are given in Figure 4.33. It can be seen clearly that, transvers displacement causes bolt to bend, and as a result to have stress values increased along the shank, thread root and under head radiuses. Repetitive load cycle in the transvers direction cause contacting surfaces to re distribute its pressure depending on the positional shift of the head based on the axis of the bolt. This causes micro slips in the bearing and thread contacting areas which then causes bolt rotation in each cycle. This rotation causes a decrease in the preload and loosening phenomena occurs. In order to evaluate model precision, experimental findings are used to compare the outcomes of the simulation model. Clamp load versus cycle graphs obtained from the Junker vibration test device is compared with the same outcome obtained from the simulation model.

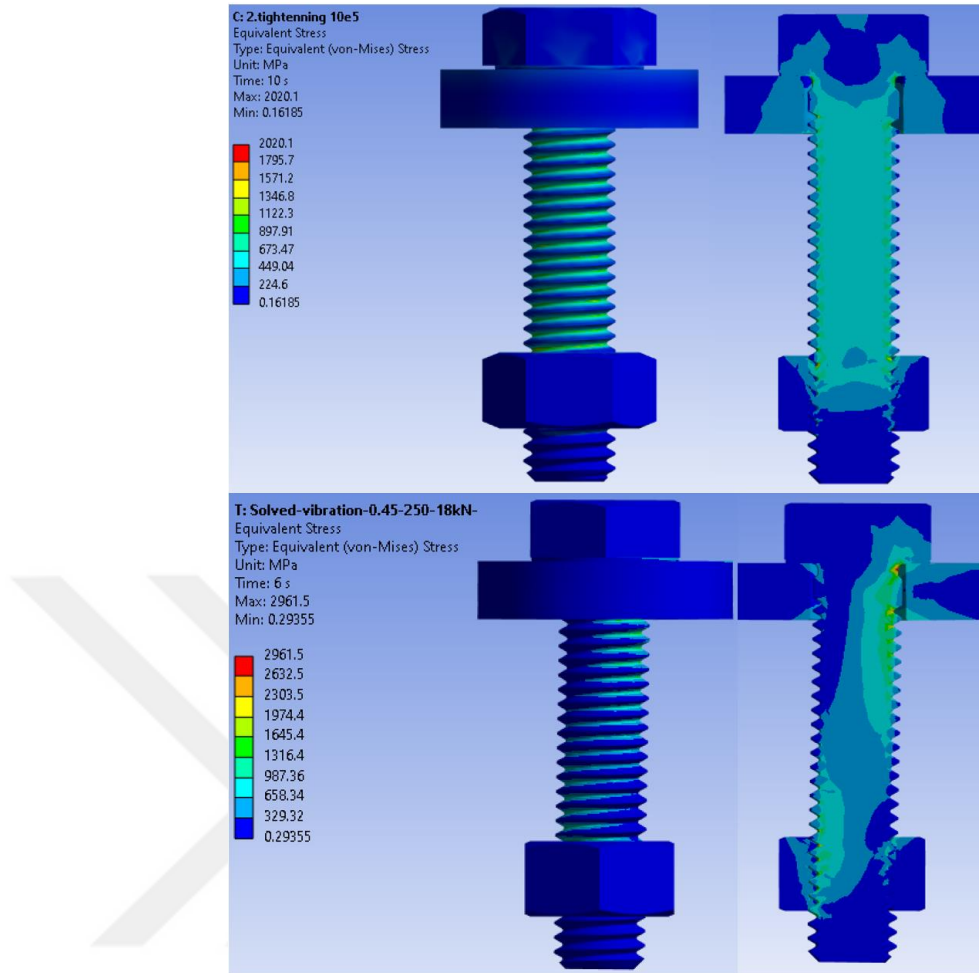


Figure 4.33 Equivalent von Mises stress distribution along the bolt under transvers displacement on the bottom and in nominal state at the top

Bending moments caused by the transvers displacement also cause a non-uniform pressure distribution on the bearing surface of the bolt. Each repetitive cycles redistribute this pressure variation along the contacting surfaces. Results at 5th and 6th are given in the Figure 4.34 for the bearing surface and same representation for the threaded portion is given in Figure 4.35.

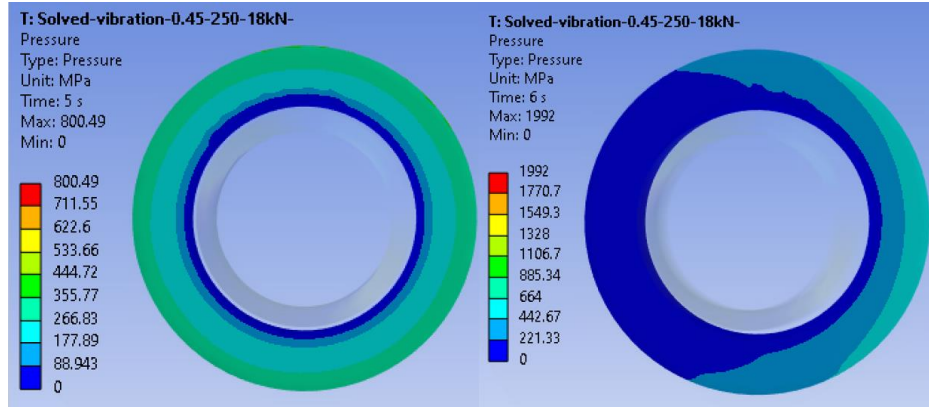


Figure 4.34 Pressure distribution on the bearing surface with transvers displacement

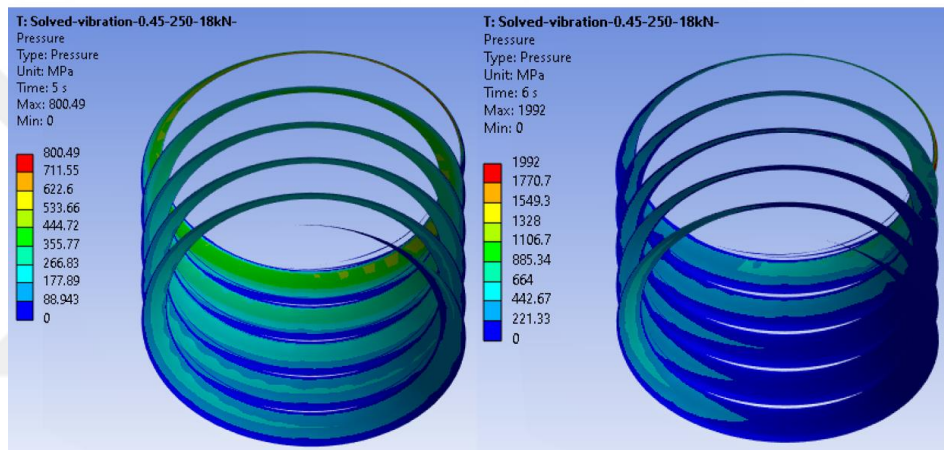


Figure 4.35 Pressure distribution on the thread surface with transvers displacement

Comparison of clamp load versus cycle outcomes from experimental and simulation software is given in Figure 4.36 to validate our simulation model precision. According to the obtained values, loosening rate and behavior of the curve is in well correlation with the experimental results. An example graph of a fully loosened curve is given is Figure 4.37.

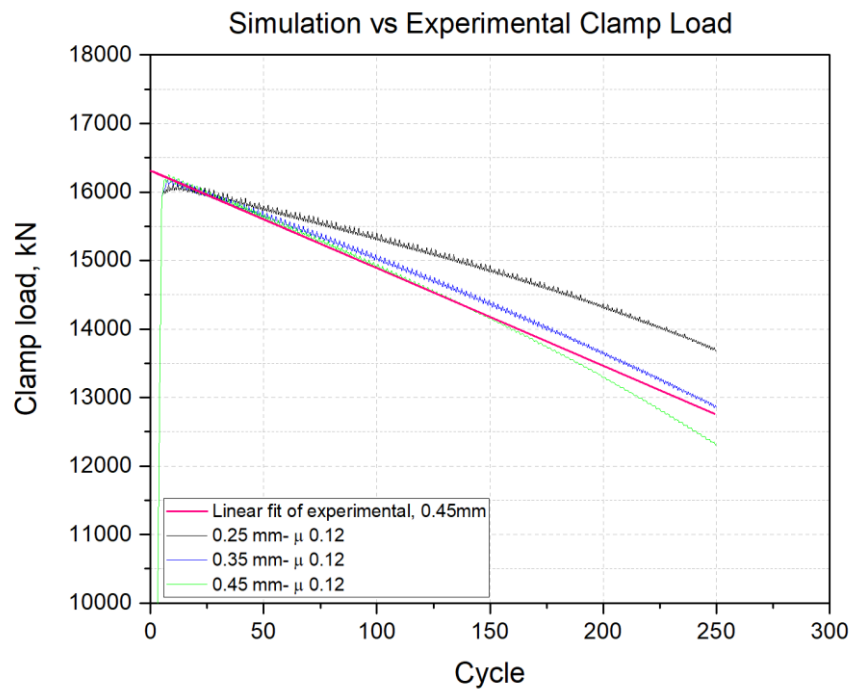


Figure 4.36 Loosening curve comparison of experimental and simulation result

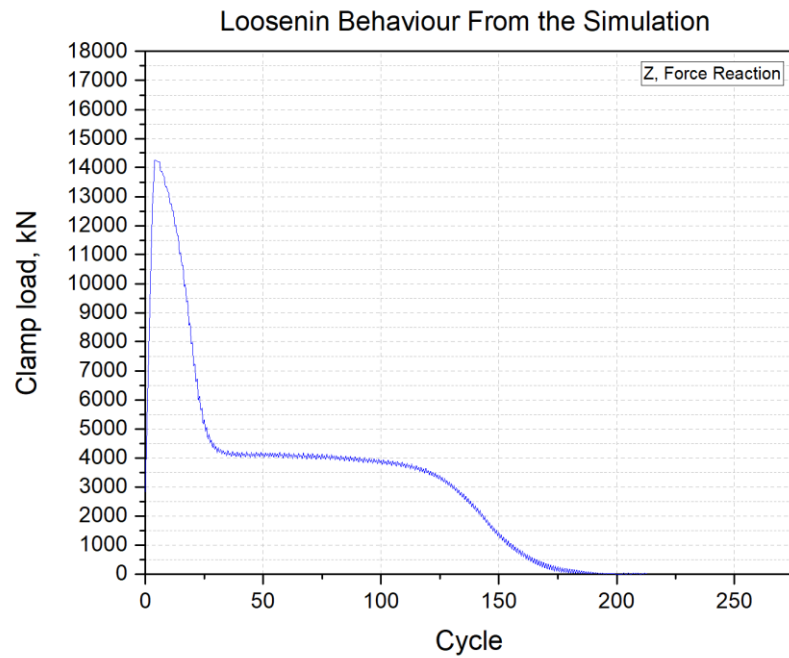


Figure 4.37 Loosening curve output of a simulation model

4.6 Loosening Rate Prediction Model with Neural Networks

Due to the various parameters effecting loosening rates and difficulties to test all the combination, a suitable prediction model is needed to estimate these values for all load cases. For this purpose, neural network based deep learning algorithm was developed in order to predict loosening rates which are needed for prediction or fastener life cycles. In order to achieve this, a structure of a model was first designed based on several performance criteria. Data for model training was obtained from previous experimental findings and loosening rates obtained from the simulation method. For the input layer five main parameters were set. These are, clamp load, clamp length, friction coefficient, transvers displacement and coating type. These main 5 inputs were used to train the main model in order to have an output of loosening rates. A split of 70% train and 30% test was set in the neural network code for the overall model. Main structure of the code was set to iterate the best outcome which can be reached with trialing different layer count and layer densities. As a performance predictor, MSE which can be described as mean square error, MSA, mean absolute error and R squared values are used. Equations for all three types of performance evaluation methods are given in Equations 4.9, 4.10 and 4.11 respectively. Where the variables f_i , y_i , N , and $\bar{y}$ represent the predicted value given by the moodel, true value of datas, number of the sample and evarag of the true data respectively.

$$\text{MSE} = \frac{1}{N} \sum_{i=1}^N (f_i - y_i)^2 \quad 4.9$$

$$\text{MAE} = \frac{1}{N} \sum_{i=1}^N |f_i - y_i| \quad 4.10$$

$$R^2 = 1 - \frac{\sum_{i=1}^N (f_i - y_i)^2}{\sum_{i=1}^N (f_i - \bar{y})^2} \quad 4.11$$

During the calculation loop to find the best combination, iterations were calculated automatically and their outcomes were read. Once the range of the described iteration is complete, system automatically gives the best performing neural net layer and density counts for the main prediction model which I called as the best model. Best model is then saved and kept as the main neural network for loosening prediction rates as long as there is not any additional new data inputs. That loop iteration ensures that the neural network concept introduced here can be adjustable in time and deliver a better performance in time with more data introduced. This code structure also allows user to modify their data counts and outcomes of the neural networks in time. Current model has been trained with 150 input data. Due to the nature of neural networks, with the introduction of more experimental or simulation data, there is a possibility that the predictions will be improved. Above code, ensures that with a new data layer introduction, layer densities and counts could be automatically adjusted depending on the best performance variations.

Second section of the code structure aims to focus on loosening rate prediction using the saved best trained model. Once code is launched, system asks the input parameters to be manually implemented. Once the 5 inputs values which we would like to have the loosening rate estimations are introduced, system ask for how many cycles do we need to calculate the clamp load reduction? With all the data provided manually, system starts the cycle loop prediction method. In this phase, for the first cycle it uses the input values for the outcome prediction of cycle 1. With the predicted loosening rates from end of cycle 1, system automatically subtract that calculated loosening rate from the input clamp load values which was defined and set as the new clamp load value for cycle 2. This ensures that the change of clamp load is taken into account while the new loosening rate was predicted. This loop continues for the cycle amount which was introduced in the first step. After the inputted cycle amount, a clamp load cycle graph is drawn. Section of the main developed code is given in Figure 4.38 in which model re calibrates its layer counts and densities based on new input data for finding the best model.

On the following sections, model is developed and re-trained based on the best outcome which the software is able to define. Main prediction function is set to use

Keras libraries with a mean square error loss function and ‘Adam’ optimizer. Screen shot of the code structure for main prediction mode is given in Figure 4.39.

With the current data set which we have, a prediction versus actual data plot can be seen in Figure 4.40. This is based on a best trained model which results in R squared to be 0.73 with Relu function. Actual data versus prediction in loosening rate comparison is also given in Figure 4.41 for the same trained mode.

Outcomes of the performance evaluation metric can be obtained with the code structure given in Figure 4.42. Figure also includes the performance evaluation function outcomes of the trained mode.

With the trained model, it is possible to draw a clamp load loss versus cycle graphs based on the input values asked before running the code. For this purpose, software asks to input our main parameters and a cycle count which we would like to predict. For the example given in Figure 4.43 inputs are as follows. Outcome of this inputs based on the best trained model is given in Figure 4.44. Code structure computing the new resultant loosening rates based on the new clamp load generated at the previous cycle. This allows us to re iterate loosening rates based on the outcomes of the previous cycles and thus gives us more realistic curve behaviors.

```

1 import pandas as pd
2 import numpy as np
3 import tensorflow as tf
4 import itertools
5 from sklearn.model_selection import train_test_split
6 from sklearn.metrics import r2_score
7
8 # Input datas are loaded from the excel files
9 file_path = '/content/Looseningratedata.xlsx'
10 data = pd.read_excel(file_path)
11
12 # Split data into input (X) and output (y)
13 X = data[['clamp load', 'clamp length', 'friction coefficient', 'lateral displacement']]
14 y = data['loosening rate']
15
16 best_r_squared = -float('inf')
17 best_config = None
18 tried_configs = set()
19 max_trials = 1000 # Define max_trials here
20
21 # Define a function to generate all configurations
22 def generate_all_configs():
23     layer_counts = list(range(3, 12))
24     layer_dense = list(range(5, 25))
25     activations = ['relu'] # Randomly select activation function
26     return list(itertools.product(layer_counts, layer_dense, activations))
27
28 # Get all possible configurations
29 all_configs = generate_all_configs()
30
31 # Define a function for model training and evaluation
32 def train_evaluate_model(layer_count, layer_dense, activation, X_train, X_test, y_train, y_test):
33     model = tf.keras.Sequential()
34     model.add(tf.keras.layers.Dense(layer_dense, activation=activation, input_shape=(X_train.shape[1],)))
35     for _ in range(layer_count - 1):
36         model.add(tf.keras.layers.Dense(layer_dense, activation=activation))
37     model.add(tf.keras.layers.Dense(1)) # Output layer (1 neuron for regression)
38
39     model.compile(optimizer='adam', loss='mean_squared_error')
40
41     model.fit(X_train, y_train, epochs=100, batch_size=32, verbose=0)
42
43     predictions = model.predict(X_test)
44     r_squared = r2_score(y_test, predictions)
45
46     return r_squared

```

Figure 4.38 Main optimization function for the best model investigation

```

76 # After finding the best configuration, retrain the model with the best configuration
77 best_layer_count, best_layer_dense, best_activation = best_config
78 X_train, X_test, y_train, y_test = train_test_split(X, y, test_size=0.2, random_state=42)
79
80 # Retrain the model with the best configuration
81 model = tf.keras.Sequential()
82 model.add(tf.keras.layers.Dense(best_layer_dense, activation=best_activation, input_shape=(X_train.shape[1],)))
83 for _ in range(best_layer_count - 1):
84     model.add(tf.keras.layers.Dense(best_layer_dense, activation=best_activation))
85 model.add(tf.keras.layers.Dense(1)) # Output layer (1 neuron for regression)
86
87 model.compile(optimizer='adam', loss='mean_squared_error')
88 model.fit(X_train, y_train, epochs=100, batch_size=32, verbose=0)
89
90 # Predict on test data using the trained model
91 predictions = model.predict(X_test)

```

Figure 4.39 Main prediction model for loosening rate outcomes

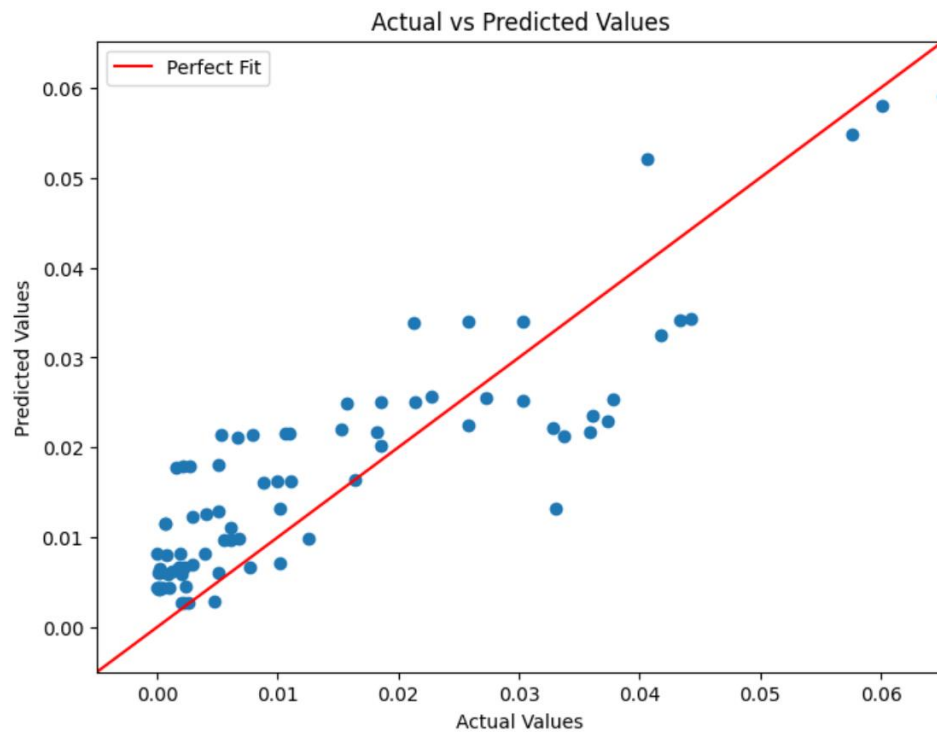


Figure 4.40 Predicted versus actual plot

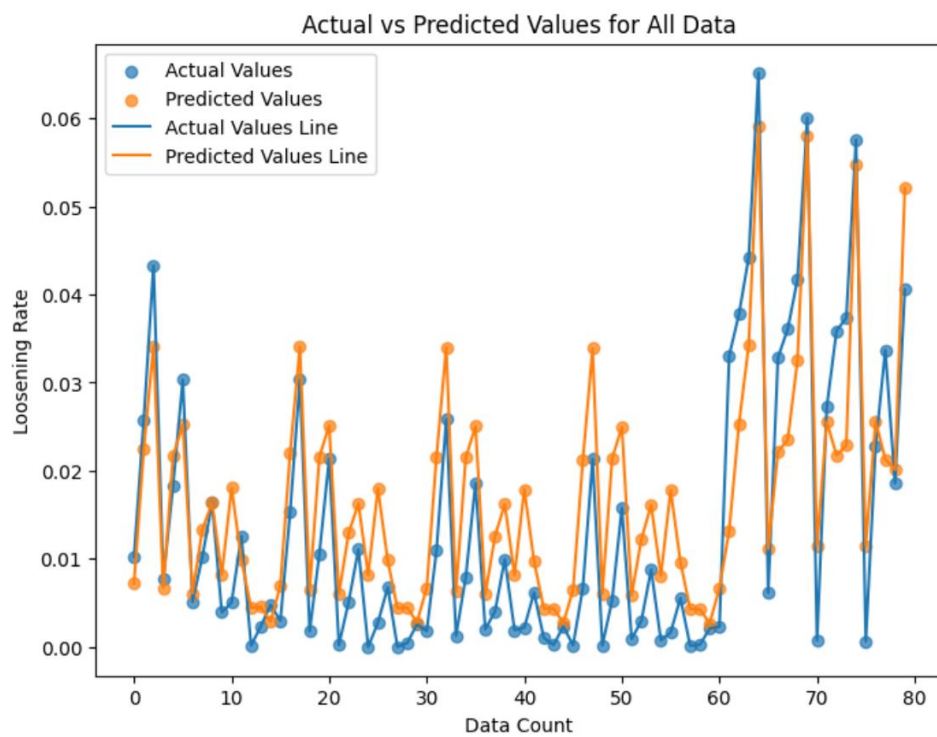


Figure 4.41 Loosening rate values of predicted versus actual data

```

1 from sklearn.metrics import r2_score, mean_absolute_error, mean_squared_error
2
3 # Calculate R-squared
4 r_squared = r2_score(y, all_predictions)
5 print(f"R-squared (R²): {r_squared}")
6
7 # Calculate Mean Absolute Error
8 mae = mean_absolute_error(y, all_predictions)
9 print(f"Mean Absolute Error (MAE): {mae}")
10
11 # Calculate Mean Squared Error
12 mse = mean_squared_error(y, all_predictions)
13 print(f"Mean Squared Error (MSE): {mse}")
14
15
16
17

```

```

R-squared (R²): 0.7367940262989163
Mean Absolute Error (MAE): 0.006797878259108011
Mean Squared Error (MSE): 6.670913032410099e-05

```

Figure 4.42 Performance evaluation of the trained model

```

Enter the number of cycles: 500
Clamp Load: 20
Clamp Length: 25
Friction Coefficient: 0.1
lateral displacement: 0.45

```

Figure 4.43 Input values for loosening curve prediction

Due to the nature of the neural network, there is a need to have a broad range of input data which can be used for training the model. Higher amount of data will result to better trained models and better prediction rates. Although our current model is delivering a reliable outcome, there is still areas to improve with a more data counts. This is why we developed an open-ended loop for the main structure which can be fed as long as we have new experimental or simulation data which can be used as a train set.

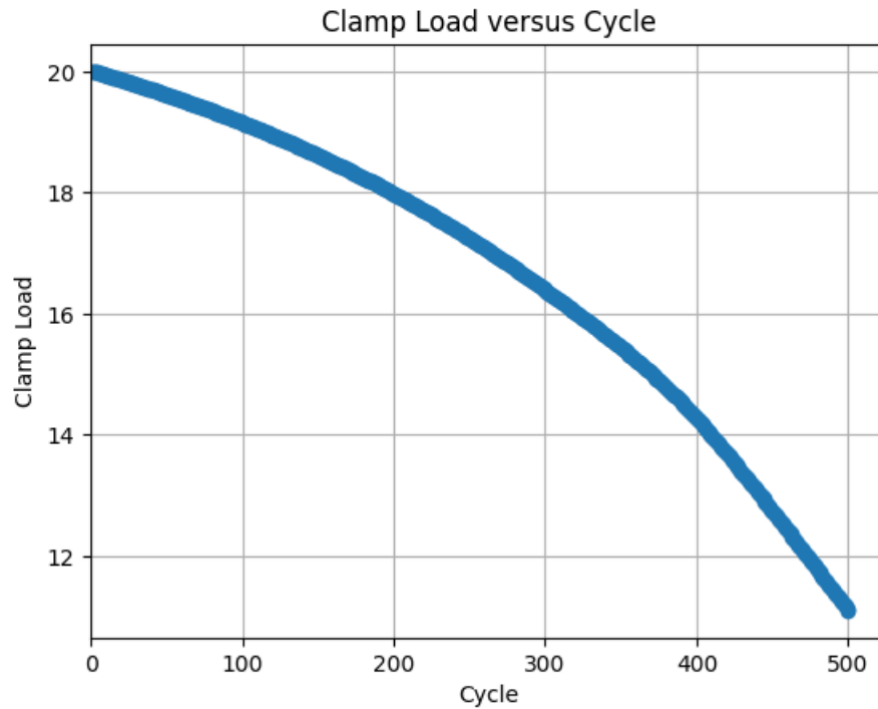


Figure 4.44 Predicted loosening curve

4.7 Loosening and Fatigue Effected Prediction Model with Reusability.

The objective of our study is to develop a methodology for accurately forecasting the fatigue life of bolts subjected to the effects of loosening over time. To ascertain the fatigue life, it is important to possess knowledge on the loosening properties shown by bolts under vibrational conditions. Due to this reason, the first experimental investigations mostly concentrated on determining and predicting the rate at which a bolt becomes loose when subjected to lateral motion. As evidenced in the literature, fatigue in bolts is significantly influenced by the average load and amplitude load experienced over its operational lifespan. Loosening refers to the process of adjusting the average load value of a bolt and affecting its fatigue behavior over time. Previous studies conducted in this field have proved that the Palmgren-Miner methodology is a great technique for predicting the cumulative fatigue life of bolts subjected to varying loads. In the present case, the utilization of the updated Palmgren Minor procedures was employed on our dataset as a result of the fluctuations seen in the clamp load over a period of time.

Palmgren-Miner linear damage hypothesis states that each loading cycle contributes to the damage of a component. Thus, the life cycle of a component is expected to reach to its fatigue life once the cumulated fatigue damage reaches to 1. Equation of this rule is given in Equation 4.12.

$$\sum_{i=1}^k \frac{n_i}{N_i} = C \quad 4.12$$

For design purposes, assumption for the failure is considered when C is equal to 1 in a cumulative failure model. In our situation, each cycle has a distinct load range when bolts are exposed to time-dependent loosening. As a result, the Palmgren Minor approach is expanded and enhanced to make assumptions on each load instance throughout time. This technique was initially utilized in our experimental investigation, which was done on bolts and nuts subjected to repetitive tightening situations. In a condition involving repeated tightening, there will be varying loosening rates for each repeated test cycle due to the fluctuating friction coefficient. In such a circumstance, the cumulative fatigue prediction model must account for each case separately. In order to create accurate predictions, the F_m - N_i life curve for lateral loads must be known. For these purposes, previous sections focused on determining these inputs.

In our experimental study, log-linear scale found to be more appropriate for prediction. Because of this reason Palmgren miner method was updated accordingly. Our first input, which is changing by each cycle, is the value of F_{mi} , which is our clamp load value. Due to the lateral load acting in each cycle, this value is decreasing by the amount of loosening rate. Change in the F_{mi} is causing a fatigue life change in each cycle. For this reason, we are in need to take account each cycle based on their respective fatigue life. Main equation for a log linear scale is given below. Under Equation 4.13.

$$F(x) = a.\log_{10}(x) + b \quad 4.13$$

Where $F_{m,i,j}$ is the clamp load for the respective cycle, N_i is the fatigue life for the respected cycle. Equation becomes as described in Equation 4.14. Equation 4.15 is the sub step to obtain Equation 4.16 which is the general, form of a N_i dependent of $F_{m,i,j}$ and slope of the curves. This equation is only valid where I is bigger or equal to 1 as states in Equation 4.17. There is not any possibility of having a cycle 0 or minus cycle in the equation.

$$F_{m,i,j} = a.\log_{10}(N_i) + b \quad 4.14$$

$$\frac{F_{m,i,j} - b}{a} = \log_{10}(N_i) \quad 4.15$$

$$10^{\frac{F_{m,i,j} - b}{a}} = N_i \quad 4.16$$

$$F_{m,i,j} = F_{m(i-1),j} - LR_j, \quad i \geq 1 \quad 4.17$$

Where i is cycle, j is the test sequence for the respective load case. Palmgren Miner equation for log-linear curve can be expressed as given in Equation 4.18 for each Junker vibration test cycle set.

$$\sum_{i=1}^n \frac{1}{10^{\frac{F_{m(i-1),j} - LR_j - b}{a}}} = C_{cycleset} \quad 4.18$$

A general equation can be derived with taking all install instances cumulative fatigue values in order to achieve the full cycle life time. This generalized equation is given in Equation 4.19 where n is the maximum cycle value for corresponding cycle set and k is the total cycle set amount till fracture.

$$\sum_{j=1}^k \sum_{i=1}^n \frac{1}{10^{\frac{F_{m,(i-1),j} - LR_j - b}{a}}} \quad 4.19$$

In our experimental study, failure occurred at fifth set, which means that the LR_j value is changing for each set. Total cumulative damage based on this assumption was calculated with using excel macros for each test set and cycles. Junker vibration tests with repetitive load cases are given in Figure 4.45 and Figure 4.46.

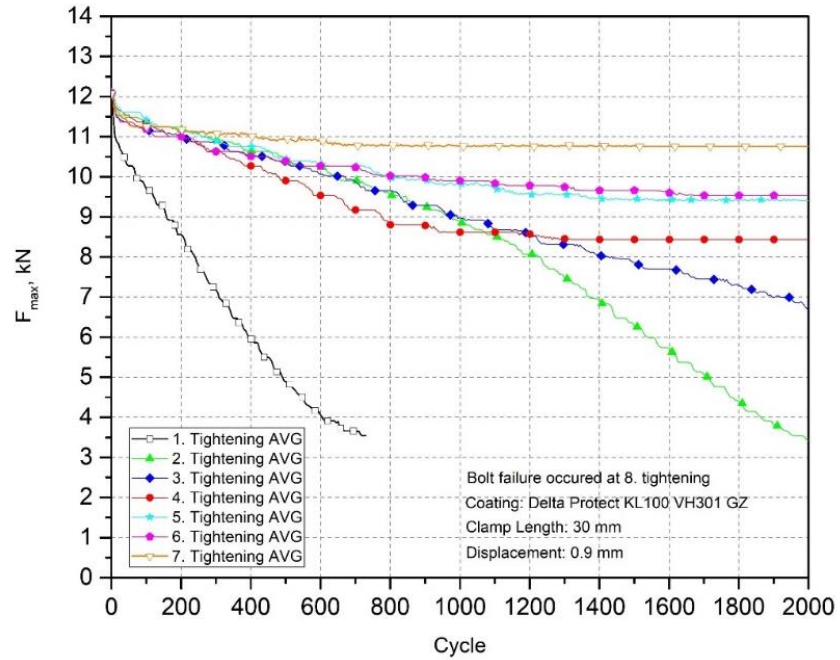


Figure 4.45 Junker Vibration test result for 12.1 kN with repeated tightening

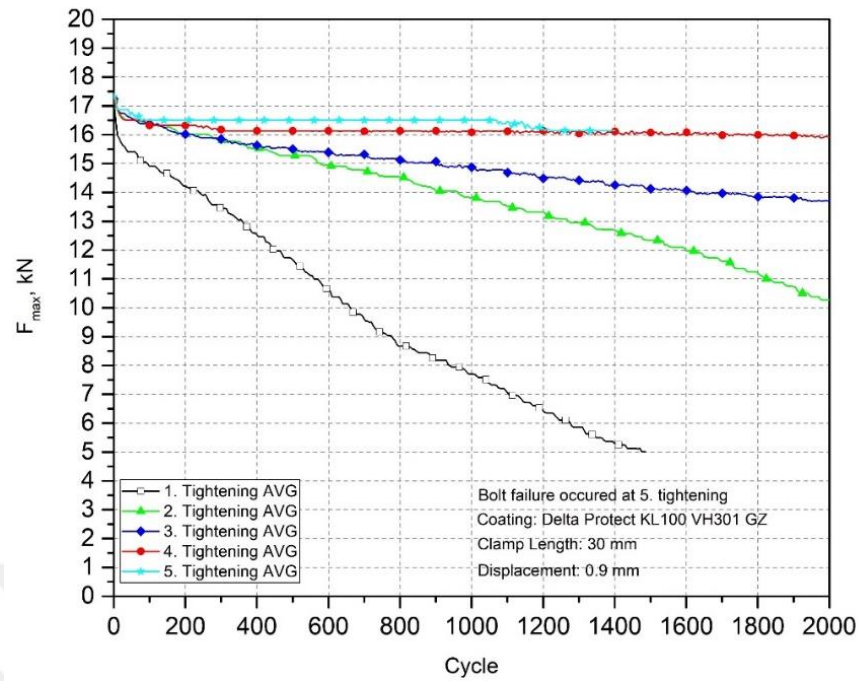


Figure 4.46 Junker vibration test result for 17.1 kN with repeated tightening

In the experimental setup of the Junker test bench, it is notable that each distinct load case corresponds to a unique loosening rate. The phenomenon of loosening rates can be quantified by measuring the reduction in clamp load experienced during each repeated cycle. Loosening rate values obtained from the experimental study based on repetitive Junker vibration test is given in Table 4.6.

Experimental test results in lateral loading condition is given in Figure 4.47 in linear log scale. $F_{\max}$ - N_i cycle graphs are obtained by running different load cases until fatigue failures occurs for each individual load cases for 10 sets. Results of the experimental tests demonstrate a clear relationship between the reduction of $F_{\max}$ values resulting from lateral loads and an increase in the fatigue life of the bolt. Finding obtained by the experimental study herein demonstrate a notable agreement with a prior investigation that examined the fatigue endurance of bolts subjected to transverse loading conditions (Hashimura et al., 2017; L. Yang et al., 2021).

Table 4.6 Loosening rates for various load cases and repeated usages

Maximum fastener load, $F_{\max}$, (kN)	Number of repeated usages, J	Loosening rates, LR, (kN/Cycle)
17.1 kN	1	0.007630
17.1 kN	2	0.003080
17.1 kN	3	0.001370
17.1 kN	4	0.000172
17.1 kN	5	0.000152
15.6 kN	1	0.010530
15.6 kN	2	0.004520
15.6 kN	3	0.001980
15.6 kN	4	0.001470
15.6 kN	5	0.000780
15.6 kN	6	0.007700
12.1 kN	1	0.010780
12.1 kN	2	0.006310
12.1 kN	3	0.002950
12.1 kN	4	0.002710
12.1 kN	5	0.001850
12.1 kN	6	0.001830
12.1 kN	7	0.001830
12.1 kN	8	0.001830

Estimation of cumulated fatigue lives in repetitive bolt usages is conducted using the modified Palmgren Minor method, as presented in equation 7. Results of these predictions are tabulated in Table 2. Pictures of the bolt failure due to fatigue during Junker (a), bolt without experiencing any damage at the end of the test (b) are given in Figure 4. The present analysis illustrates the visible appearance of fatigue fractures resulting from the cumulative damages incurred during repeated usage and cycles, once the fatigue limit of the bolt has been surpassed. Development of fatigue cracks are within the initial two thread roots, where localized maximum stresses are observed.

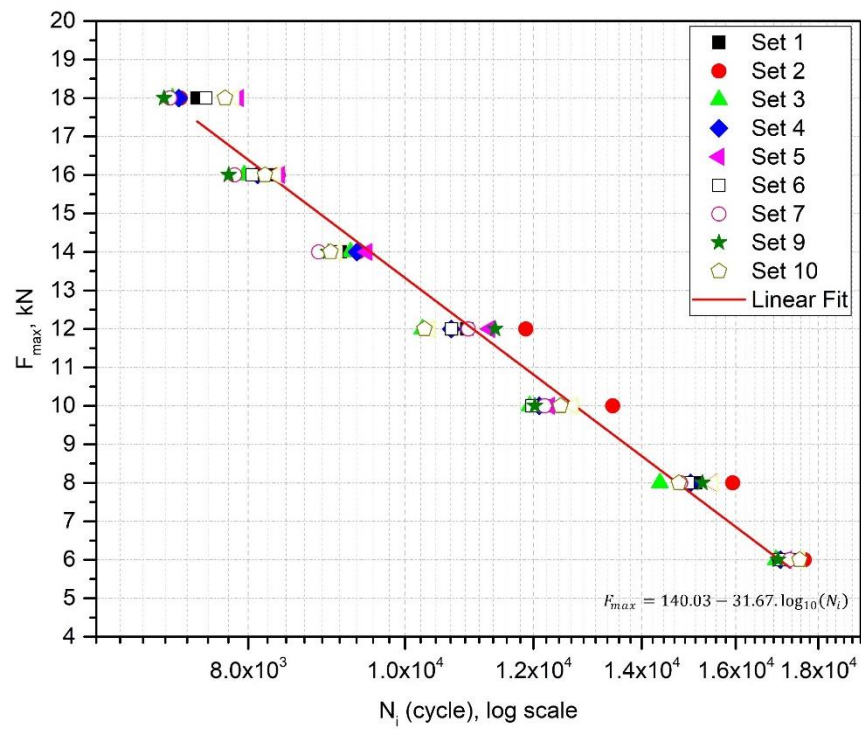


Figure 4.47 F_{max} - N_i curve at lateral loading condition with restricted loosening

Table 4.7 Cumulative fatigue damage prediction with repeated tightening

Maximum fastener load, F_{max} , (kN)	Number of repeated usages, j	Cycle until test end or failure, i	Results of the test	Cumulative damage for load case, D_i	Cumulative damage with repeated usages, $D_{j,i}$
17.1 kN	1	1782	Loosening	0.13596	0.13596
17.1 kN	2	2000	Test end	0.21171	0.34768
17.1 kN	3	2000	Test end	0.23816	0.58584
17.1 kN	4	2000	Test end	0.25942	0.84527
17.1 kN	5	1416	Fatigue	0.18453	1.02982
15.6 kN	1	1477	Loosening	0.10413	0.10413
15.6 kN	2	2000	Test end	0.17261	0.27675
15.6 kN	3	2000	Test end	0.2046	0.48140

Table 4.7 Continued

15.6 kN	4	2000	Test end	0.21206	0.69346
15.6 kN	5	2000	Test end	0.22267	0.91614
15.6 kN	6	1431	Fatigue	0.11539	1.0315
12.1 kN	1	719	Loosening	0.05017	0.10413
12.1 kN	2	2000	Test end	0.11949	0.16966
12.1 kN	3	2000	Test end	0.14848	0.31815
12.1 kN	4	2000	Test end	0.15092	0.4690
12.1 kN	5	2000	Test end	0.16011	0.62918
12.1 kN	6	2000	Test end	0.16011	0.78937
12.1 kN	7	2000	Test end	0.16011	0.94945
12.1 kN	8	212	Fatigue	0.01901	0.96845

Based on the data presented in Table 4.7, it is evident that the observed damage accumulation during each load cycle aligns with the anticipated patterns. The effect of reduced clamp load in bolted joint resulting from loosening has been identified as a significant factor contributing to reduced damage accumulation in bolts. However, it is important to consider that with repeated use of bolts, the rate of loosening gradually declines, leading to an increase in the accumulation of damage with each subsequent cycle. Damage accumulation for each repetitive installations and cycles are given in Figure 4.48, Figure 4.49 and Figure 4.50 for 17.1 kN, 15.6 kN and 12.1 kN respectively.



(a)



(b)

Figure 4.48 Bolt failure due to fatigue during Junker (a), bolt without experiencing any damage at the end of the test (b)

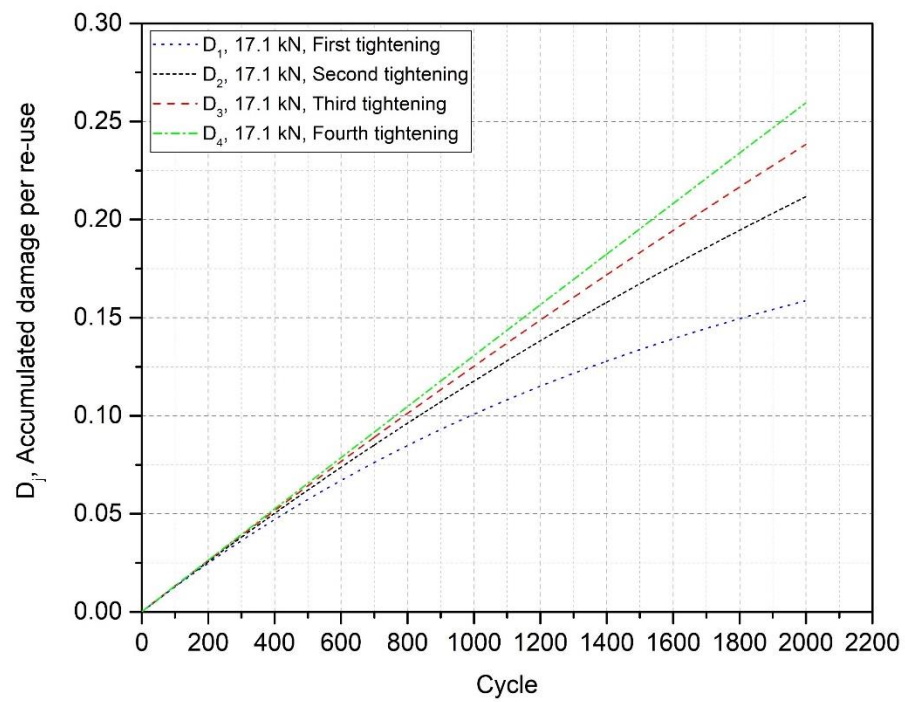


Figure 4.49 Accumulated damage per load case versus loosening rate for 17.1 kN

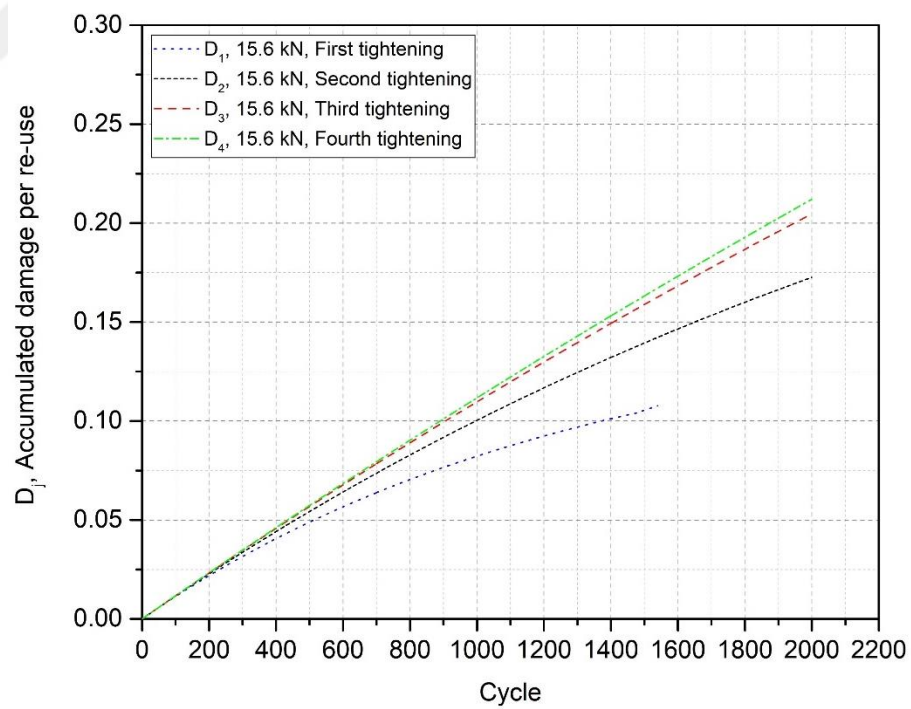


Figure 4.50 Accumulated damage per load case versus loosening rate for 15.6 kN

Although the aforementioned technique has demonstrated satisfactory predictive outcomes, there is a necessity for further enhancements that may be extended to cover a broader range of general load instances. In order to obtain an accurate forecast of the load scenario, it is necessary to have a loosening rate input obtained from experimental investigation, as well as knowledge of the transverse fatigue life for the projected parameters. To enhance the predictive approach over a wider spectrum, it is important to convert lateral displacement measurements into corresponding lateral force, resulting bending moment, resultant axial load, and stress values. Bolts loaded in transvers directions can be compares as a beam element. For this reason, a similar equilibrium can be drawn in order to calculate bending moments and lateral forces based on transvers displacements and clamp length. This assumption is only valid based on elastic beam theories and not covers plastic regions. A schematic representation of acting loads and stress in transversely loaded state are given in Figure 4.51. When bolted joints are loaded in transvers load, their shape becomes as an S. This is mainly because of the acting moments on the bolt due to lateral forces which forces bolt to lose its axially. As previously said, this moment value can be easily calculated with beam theories. Equation 4.20 shows the moment acting on the bolt under head radius while Equation 4.21 is for calculating the effecting moment in the first engaged threads according to (Hashimura, 2007; Hosoya et al., 2018; Kasei, 2007). Where l_g is the grip length, l_e engaged thread length, F_{TR} transvers force, C section modulus of the bolt.

$$M(y) = (F_{TR}) \cdot l_g - M_B \quad 4.20$$

$$M_B = C \cdot (F_{TR}) \cdot l_g \quad 4.21$$

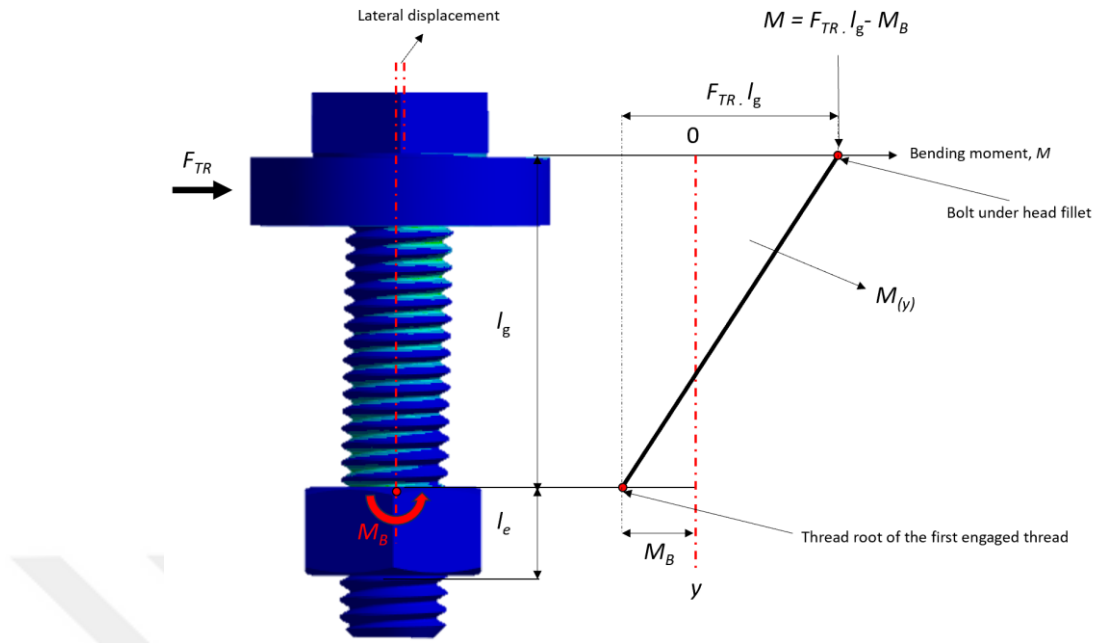


Figure 4.51 Load and moment diagram of a bolt loaded in transvers direction

A similar concept is defined in VDI 2230 as well without taking into account the thread engagement depths for standard bolt-nut connections. In that condition the mid-section of the moment graphs will be equally divided into two. Shear strains acting on the bolt in the transvers direction results a shear stress in the member. In order to obtain a force equation there is a need to solve the cantilever beam equations. Shear loading case of a bolt can be assumed as a cantilever beam fixed on one and free on the other with a moment acting at the free end. From a free body diagram, maximum deflection and reflecting force will happen at the free end of the cantilever beam. With taking the equation resultant transversal force acting on the bolt bearing surface can be described as Equation 4.22. Where F_{trs} is the transvers load acting in y axis, E_b is the bolt elastic modulus, I_b is the moment of inertia of the bolt, L_g is the grip length and δ_y is the deflection in y axis which corresponds to our transvers displacement values. After obtaining the transversal load, bending stress at the first thread root of the engaged threads can be achieved with the Equation 4.23. Where $\sigma_{t,y}$ is the bending stress caused by y displacement at y axis and D_3 is the thread root effective stress diameter.

$$F_{trs,y} = \frac{12 \cdot E_b \cdot I_b}{L_g^3} \cdot \delta_y \quad 4.22$$

$$\sigma_{trs,y} = \pm \frac{F_{trs,y} \cdot L_g}{2 \cdot I_b} \cdot \frac{D_3}{2} \quad 4.23$$

Bolts which are affected by transversal displacement exhibits an elongation in the Z axis as well. This elongation causes an increase on the bolt tensile stress and loads. For this purpose, it is essential to calculate that effect as well even if this is a relatively smaller value compared to the transversal stress value. In order to calculate this outcome a representation of the bolt shank elongation was given in Figure 4.52. Extension on the length of the bolt can be calculated with the equation given in 4.24. This value can be taken as a derivative to obtain the extension of the length with using the Equation 4.25. A general expression based on the length integration can be obtained as given in Equation 4.26 to 4.29 (Welch, 2018) .

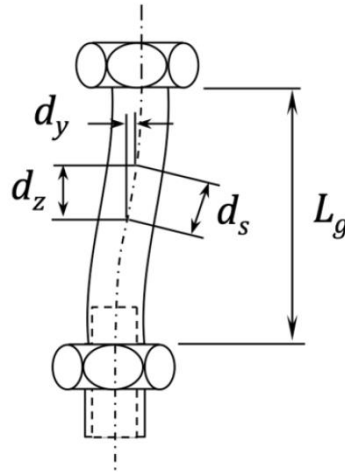


Figure 4.52 Schematic representation of bolt elongation (Welch, 2018)

$$ds = \sqrt{dz^2 + dy^2} \quad 4.26$$

$$ds = dz + \frac{1}{2} \cdot \left(\frac{dy}{dz} \right)^2 dz \quad 4.27$$

$$\text{Extended length} = L + \delta_z = \int_0^L ds \quad 4.28$$

$$\delta_z = \frac{1}{2} \cdot \int_0^L \left(\frac{dy}{dz} \right)^2 dz \quad 4.29$$

By solving the above integration, elongation on the bolt due to shear load can be achieved as given in Equation 4.30 for infinitely stiff joints. In order to take account, the compression distribution on the member areas for a more precise value a joint stiffness value can be used as given in Equation 4.31, a common range of 1.1-1.2 can be taken for the stiffness effect in the below equation.

$$\delta_z = \frac{F_{trs,y}^2 \cdot L_g^5}{240 \cdot (E_b \cdot I_b)^2} \quad 4.30$$

$$F_{A,z,bending} = \frac{F_{trs,y}^2 \cdot L_g^4 \cdot A_b}{240 \cdot E_b \cdot I_b^2 \cdot \left(1 + \frac{A_b \cdot E_b}{k_{jp} \cdot L_g} \right)} \quad 4.31$$

With the above equation it is now possible to find the resultant total axial stress and loads. In order to get the total axial load, including the preload and transvers effect, Equation 4.32 needs to be solved. For total resultant stress effect, which is also dependent on the load orientation, Equation 4.33 is given. Where F_{axial} is the total axial force acting on the bolt axis, σ_{axial} total axial stress value including the bending effect.

$$F_{A,z,total} = \sigma_{a,z} \cdot A_{b,z} + F_{A,z,bending} \quad 4.32$$

$$\sigma_{axial,total} = \frac{F_{a,z}}{A_{a,z}} \pm \sigma_{a,z,bending} \quad 4.33$$

It is also very important that the slip resistance of the bearing surface needs to be taken into account. If the static equilibrium of friction forces are passed beyond, joint will slip and not transmit the shear load the bolt shank. Bolt bearing load capability is a function of the friction coefficient value and the preload value in the joint. This relationship can be described as given in Equation 4.34. In which $F_{sliding}$ limits defines the maximum transvers force limits which can keep the plate without sliding. μ_b is the friction coefficient of the bearing surface and F_p is the preload value of the joint after assembly. A limit can be given as the Equation 4.35.

$$F_{sliding\ limit} = \mu_b \cdot F_p \quad 4.34$$

$$If\ F_{sliding\ limit} < F_{transvers} : F_{transvers} = F_{sliding\ limit} \quad 4.35$$

VDI 2230 defines and suggest to use von Mises stress for calculating the thread root stresses. An equation substituted for the above load case is given in Equation 4.36 and Equation 4.37. Where $\sigma_{VM,root,min}$ is the minimum von Mises stress at thread root, $\sigma_{axial,min}$ minimum axial stress and τ_{xy} shear stress. Same accounts for the maximum von Mises stress values as well. Both equations are given in 4.36 and 4.37

$$\sigma_{VM,root,min} = \sqrt{\sigma_{axial,min}^2 + 3 \cdot (\tau_{xy}^2)} \quad 4.36$$

$$\sigma_{VM.root,max} = \sqrt{\sigma_{axial,max}^2 + 3 \cdot (\tau_{xy}^2)} \quad 4.37$$

For fatigue calculations, there is a need to know the amplitude and mean stresses which are dependent on the above load cases. A general expression can be drawn based on the above inputs. Where $\Delta\sigma$ is the stress range, $\sigma_{root,max}$ is the maximum stress at thread root and $\sigma_{root,min}$ minimum stress at the thread root, σ_{amp} is the stress amplitude σ_{mean} mean stress as given in Equation 4.38, 4.39 and 4.40.

$$\Delta\sigma = \sigma_{root,max} - \sigma_{root,min} \quad 4.38$$

$$\sigma_{amp} = \frac{\Delta\sigma}{2} = \frac{\sigma_{root,max} - \sigma_{root,min}}{2} \quad 4.39$$

$$\sigma_{mean} = \frac{\sigma_{root,max} + \sigma_{root,min}}{2} \quad 4.41$$

After defining the stress amplitude due to bending, calculation can be made with using the previously developed S-N curve for bending load. Formulation of the experimentally derived S-N curve is given in Equation 4.42.

$$\log N = 12.51 - 3.27 \log \sigma_{amp} \quad 4.42$$

With the above equations and limitations, it is now applicable to predict bolt fatigue lives with taking the bending effects into considerations. Aim of the study was to take the loosening effects and reuse effects into the consideration. For this purpose, above calculations and their resultant effects needs to be calculated in each transvers cycles. With the calculations of each load conditions and resultant tensile and transvers

stresses acting on the bolt, a fatigue life estimation based on accumulative damage criteria can be achieved. In order to solve this complex problem, an excel file is created with equations to obtain above calculations with the input values. A table of the calculation method and its corresponded output from the calculator is given in Table 4.8.

Table 4.8 Shows the calculated values based on the equation given in the section for bolt stress calculation

Name	Equation	Unit	Value
Clamp length	L_g	[mm]	45
Clamp force	$F_{a,z}$	[N]	17100
Displacement, transvers,	δ_y	[mm]	0.15
Transvers force	$F_{trs,y} = \frac{12 \cdot E_b \cdot I_b}{L_g^3} \cdot \delta_y$	[N]	833.61
Bolt diameter	D_b	[mm]	8
Elastic modulus	E	[MPa]	210000
Second moment of area	I	[mm ⁴]	200.96
Bending stress at root due to transvers force	$\sigma_{trs,y} = \pm \frac{F_{Transvers} \cdot L_g}{2 \cdot I_b} \cdot \frac{D_3}{2}$	[MPa]	336
Thread root effective diameter	D_3	[mm]	7.2
Tensile force due to bending	$F_{A,z,bending} = \frac{F_{trs,y}^2 \cdot L_g^4 \cdot A_b}{240 \cdot E_b \cdot I_b^2 \cdot \left(1 + \frac{A_b \cdot E_b}{k_{fp} \cdot L_g}\right)}$	[N]	215.61
Tensile area of the bolt	A_b	[mm ²]	36.6
Slip Limited by Friction	$F_{sliding\ limit} = \mu_b \cdot F_p$ $If\ F_{sliding\ limit} < F_{transvers} : F_{transvers} = F_{sliding\ limit}$	[N]	1710
Bearing friction coefficient		----	0.1
Tensile stress amplitude due to bending		[MPa]	5.89
Total bolt load, preload + bending	$F_{A,z,total} = \sigma_{a,z} \cdot A_{b,z} + F_{A,z,bending}$	[N]	17315.6

Table 4.8 Continued

Total bolt stress, min , $\sigma_{total,min}$	$\sigma_{axial.total} = \frac{F_{a,z}}{A_{a,z}} \pm \sigma_{a,z,bending}$	[MPa]	137.10
Total bolt stress, max, $\sigma_{total,max}$	$\sigma_{axial.total} = \frac{F_{a,z}}{A_{a,z}} \pm \sigma_{a,z,bending}$	[MPa]	809.10
Tensile stress due F_p , σ_{mean}		[MPa]	467.21
Total tensile stress		[MPa]	473.10
Von Misses, at max thread root, σ_{vmax}	$\sigma_{VM.root,max} = \sqrt{\sigma_{axial,max}^2 + 3 \cdot (\tau_{xy}^2)}$	[MPa]	810.03
Von Misses, at min thread root, σ_{vmax}	$\sigma_{VM.root,min} = \sqrt{\sigma_{axial,min}^2 + 3 \cdot (\tau_{xy}^2)}$	[MPa]	142.67
Amplitude stress, σ_{amp}	$\sigma_{amp} = \frac{\Delta\sigma}{2} = \frac{\sigma_{root,max} - \sigma_{root,min}}{2}$	[MPa]	333.63
Mean stress	$\sigma_{mean} = \frac{\sigma_{root,max} + \sigma_{root,min}}{2}$	[MPa]	476.36
Bending Fatigue Life %50 Probability	$\text{Log}N=12.51-3.27 \times \text{Log}\sigma_{amp}$	Cycle	18139

In order to reevaluate above methodology, a special test series was conducted to compare calculated life cycle versus experimental fatigue life. A similar test method with a rotation limited fixture in the bolt head and nut portion was carried out with different length values under 0.15mm lateral displacement and with a friction coefficient of 0.1mm. Test with 35, 40 and 45 mm length values showed a good correlation with theoretical estimation. Clamp length versus cycle plot for the same lateral displacement value is given in Figure 4.53.

Above given methodology is covering the initial case based on the input's values. As previously stated, it can be seen from the calculations and formulas that, transvers load capability is directly dependent on the bearing friction coefficient value. Due to the shear load effect which results in bending, cause extremely high stress zones in the thread roots. With the increase of the bearing friction coefficient values, due to the increase of load capability of the joint, joint will get effected by a higher stress at the same transvers displacement values. This will trigger the fatigue failure in bending and cause a total fracture. Above equations based on cantilever beam theories and von

Mises stress transformations, allows to predict bolt fatigue life's, based on the previously given bending fatigue tests results and cumulative fatigue life theories.

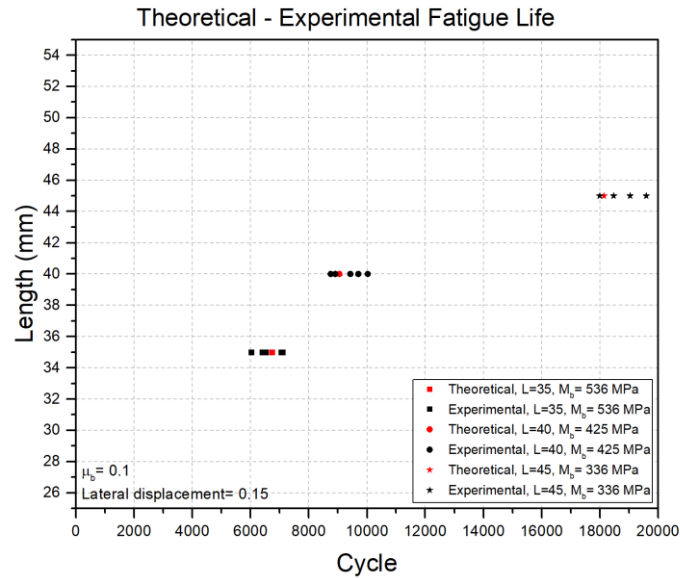


Figure 4.53 Length versus cycle graph for theoretical and experimental results

4.8 Conclusion

This study aims to contribute to the understanding of the bolt behaviors under dynamic load cases which are mainly affected by transvers loads. For this purpose experimental, theoretical, machine learning and simulation based researches were done in order to predict fastener outcomes under different load conditions. Findings shows that, bolts which are affected by vibrational loosening are more prone to have friction increases in their bearing and thread sections due to the wear in the contacting areas because of macro and micro slips. Thus, fastener reusability becomes questionable for such cases. Experimental findings revealed that, coating types also plays a key role in the outcome of their behaviors. Due to the difficulties of running multiple experimental tests, a simulation model which involves tightening, friction change and loosening was created. This simulation model allows researchers to predict their fastener outcomes in case of loosening risk persists. Results obtained from the simulations are in well agreement with the experimental results. It is suggested to use

simulation software with thread contact pair if the stresses in the bolt threads are critical. Due to thread mesh density on the loosening simulation, solver time can be very high depending on the mesh quantities. To resolve this issue, a neural network based deep learning algorithm and model was developed. Model was trained with the experimental and simulation outcomes and parameters. In order to control the function errors MSE, MSA and R square values are used. Model loop established in this context are set based on a loop which can find the best parameters while training and finding the best cases. This allows us to feed the neural network in the future with experimental and simulations data's which is going to increase the reliability of the program without touching to the base code. Current state of the neural network is working well if inputs are within the range of the trained parameters. In order to increase the model reliability and range, there is a need to include more data which is going to deliver more capability. Last, section of the study focuses more on the theoretical background of the bolt stresses and loosening lives. Calculations were made based on cantilever beam theories which is a common way for such load cases in bolted joints. Fatigue life prediction methodology developed based on the transversal fatigue life's under repeated tightening with using the cumulative damage theories. Findings obtained from the experimental study and theoretical assumptions are within a good error range. In order to expand the methodology developed in the initial case, a cantilever beam theory was introduced and transversal displacement inputs was transformed to lateral shear and force values for using in life predictions under loosening states. This method also involves bearing friction effects and load transfer capabilities into the calculation. Increase in the shear load capability, causes an increase on the bending moment under the same transversal displacement which is causing a reduction in the bolt fatigue life. On the other hand if the lateral displacement is a function of a steady displacement, with the increase of the bolt clamp length, resultant bending moments decreases on the threaded portion. Which causes an increase in the fatigue life of the bolt. That is why the input parameters of the system is critical and should be taken into account depending on the case. For high transversal loads, where loosening plays a key role, reuse of the fasteners should be limited due to the increase of the bearing friction which increase the lateral load capability of the joint. This impact might have an adverse effect on the total bolt life as proven in the given study.

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