

T.R.
GEBZE TECHNICAL UNIVERSITY
ENERGY TECHNOLOGIES INSTITUTE

**INVESTIGATION AND APPLICATION OF THE TORQUE
ANGLE TIGHTENING METHOD ON CRITICAL BOLTED JOINTS
OF GAS TURBINE ENGINE**

YASİN OKAN KÖSTERİT
**A THESIS SUBMITTED FOR THE DEGREE OF
MASTER OF SCIENCE**
**DEPARTMENT OF APPLIED PROPULSION SYSTEM
DESIGN& ENGINEERING FOR AEROSPACE TECHNOLOGIES**

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THESIS SUPERVISOR
PROF. DR. MEHMET ALİ ARSLAN

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2022



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JÜRİ

ÜYE

(TEZ DANIŞMANI) : Prof. Dr. Mehmet Ali ARSLAN

ÜYE

: Prof. Dr. Osman Saim DİNÇ

ÜYE

: Prof. Dr. Ata MUĞAN

ONAY

Gebze Teknik Üniversitesi Enerji Teknolojileri Enstitüsü Yönetim Kurulu'nun
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SUMMARY

Bolted joints are widely used in rotor and carcass structures of gas turbine engines of modern aircraft. The high accuracy of the preload value is crucial for the critical bolted joints of gas turbine engines to provide high-pressure gas sealing and joint integrity at the same time. The torque-angle tightening method has lower preload variation than the torque-controlled tightening method and is less complex, easier to verify and control comparing the other alternative tightening methods. In this study, the process of the torque-angle tightening method is studied and it is shown how to determine its tightening values. It is also demonstrated that the design improvement can be achieved by simply changing the tightening method rather than an expensive design change on a critical bolted joint.

The study is carried out on 0.25-28UNJF ISO3161 Inconel 718 bolts. The test parts are designed to reflect the same stiffness values as the estimated T800-LHT-800 design. Tightening values are determined for torque and torque-angle control tightening methods from torque-tension tests which are performed using the Labmaster3200 test system. In addition, a correlation is found between finite element analysis and torque-tension test results.

Using the tightening values determined as a result of torque-tension tests, 10 assembly tests are carried out for both methods with the help of a high-precision tightening wrench. By comparing the results of the two tightening methods, it is observed that the minimum and average preload values increased, while the preload variation decreased, as claimed, by only changing the tightening method.

The torque-angle tightening method can be used to achieve lower preload variation in critical bolted joint designs. Even if the torque-controlled tightening method was chosen at the design phase for a bolted joint, costly design changes can be avoided by simply changing the input of the tightening wrench after any field problem related to insufficient preload.

Key Words: Aerospace propulsion systems, tightening methods, bolted joints, torque-angle tightening, turn of nut tightening, preload

ÖZET

Modern hava araçlarının gaz türbinli motorlarının dönen ve sabit parçalarında cıvatalı bağlantılar yaygın olarak tercih edilmektedir. Ön yük değerinin doğruluğunun yüksek olması, gaz türbinli motorların kritik cıvatalı bağlantılarının yüksek basınçlı gaz sızdırmazlığını sağlarken öte yandan bağlantı bütünlüğünü de koruyabilmesi için büyük önem arz etmektedir. Tork-açı sıkma metodu, tork kontrollü sıkma metoduna göre daha düşük ön yük varyasyonuna neden olurken, diğer alternatif sıkma yöntemlerine kıyasla uygulaması daha basit, kontrolü ve doğrulaması da daha kolay olan bir metot olarak öne çıkmaktadır. Bu çalışma kapsamında, tork-açı sıkma yöntemi incelenmiş ve bu yöntemin sıkma değerlerinin nasıl belirlenebileceği gösterilmiştir. Ayrıca kritik cıvatalı bağlantılar için tasarım iyileştirmelerinin, pahalı tasarım değişiklikleri yerine sadece sıkma yönteminin değiştirilerek nasıl elde edilebileceği, seçilen kritik bir cıvatalı bağlantı tasarımı üzerinden doğrulanmıştır.

Çalışmalar Inconel 718 malzemesinden 0.25-28UNJF ISO3161 cıvataları ile gerçekleştirilmiştir. Test parçaları, referans alınan T800-LHT-800 tasarımıyla aynı direnginlik değerlerini yansıtacak şekilde tasarlanmıştır. Tork kontrollü sıkma ve tork-açı kontrollü sıkma yöntemleri için sıkma değerleri, Labmaster3200 test sistemi kullanılarak gerçekleştirilen tork-gerilim testleriyle belirlenmiştir. Ayrıca gerçekleştirilen sonlu eleman analizleri ile tork-gerilim test sonuçları arasında bir korelasyon bulunmuştur.

Tork-gerilim testleri sonucu belirlenen sıkma değerleri kullanılarak, yüksek hassasiyetli sıkma anahtarı yardımıyla her iki yöntem için 10'ar montaj testi gerçekleştirilmiştir. İki sıkma metodunun sonuçları kıyaslanarak, iddia edildiği gibi sadece sıkma yöntemini değiştirerek minimum ve ortalama ön yük değerlerinde artış sağlanırken ön yük varyasyonunun azaldığı gözlemlenmiştir.

Kritik cıvatalı bağlantı tasarımlarında düşük ön yük varyasyonu elde etmek için tork-açı sıkma metodu kullanılabilir. Bir cıvatalı bağlantı için tasarım aşamasında tork kontrollü sıkma metodu tercih edilmiş olsa dahi ön yük yetersizliğiyle alakalı herhangi bir saha problemi sonrasında sadece sıkma anahtarının girdisi değiştirilerek, maliyetli tasarım değişikliklerinin önüne geçilebilir.

Anahtar Kelimeler: Havacılık ve uzay itki sistemleri, sıkma yöntemleri, cıvatalı bağlantı, tork-açı sıkma metodu, somun döndürme metodu, ön yük

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LIST of ABBREVIATIONS and ACRONYMS

<u>Abbreviations and Acronyms</u>	<u>Explanations</u>
A	: Cross-sectional area
A_s	: Cross-sectional area of the section
d	: Nominal diameter
d_c	: Nominal diameter of the nut seating surface
D_m	: Average of the major and minor diameters
E	: Modulus of elasticity
F	: Tensile force, Preload
F_j	: Preload value obtain from the j-th test
F_{nom}	: Nominal preload value of the tests
k	: Stiffness
k_B	: Bolt stiffness
K	: Torque coefficient
k_{ji}	: Stiffness of i-th joint member
L	: Length of the section
L_e	: Effective length
m	: Sample size
P	: Thread pitch
P_i	: Pressure of the i-th station
T	: Torque
T_i	: Temperature of the i-th station
V_i	: Volume of the i-th station
α	: Tightening angle
Δx	: Change of the x, deflection
μ	: Friction coefficient
μ_c	: Friction coefficient of the nut seating surface
σ	: Unbiased sample standard deviation
θ	: Cone angle
CS-E	: Compliance for Engines
DTI	: Direct Tension Indicator
EASA	: European Aviation Safety Agency
FAA	: Federal Aviation Administration
FHA	: Functional Hazard Assessment
HPC	: High Pressure Compressor
IPC	: Intermediate Pressure Compressor
LPC	: Low Pressure Compressor
PSSA	: Preliminary System Safety Assessment
SSA	: System Safety Assessment

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1. INTRODUCTION

Bolted joints basically work by inducing an initial preload on the joint that keeps the clamped parts in contact through the life of the joint [1]. They are designed to withstand structural loads as well as provide engine alignment and high-pressure gas sealing. In line with safety expectations, they even have to withstand the ultimate loads that may arise from structural failures [2]. Although torque-controlled tightening is the most traditional and most commonly used tightening method, design requirements, flight environments, and manufacturing concerns create the need for lower preload variation [3]. Torque-controlled tightening highly depends on friction coefficient which also depends on numerous variables such as; type and hardness of materials, speed of tightening, surface finishes, lubricant types etc. Typically, between 10 percent of the applied torque led the bolt to be stretched and 90 percent is converted into energy spent on friction between the threads and the seating surfaces [4]. Although the torque-angle tightening method is very similar to the torque tightening method in terms of application, there is no dependence on friction since there is a linear relationship between the applied angle and the stretch of the bolt [5]. While many articles demonstrate plastic zone tightening uses of the torque angle tightening method, Fukuoka has shown that elastic zone applications are also applicable for critical bolted joint connections [6].

Following the failure of the T800-LHT-800 turboshaft engine's gyroscopic test, one of the design changes was to increase the bolt length to have a higher clamp load between power turbine and stub shaft to reduce gyroscopic deflections [7]. In this study, it is shown that with the estimated T800-LHT-800 design, the design of a critical bolted joint can be improved by simply changing the tightening method rather than a costly design change.

1.1 Motivation of Research

The performance, fuel efficiency, availability, and safety requirements of today's gas turbine engines demand designers to sharpen their pencils. The necessity of designing light enough to reduce fuel consumption, durable enough to minimize maintenance frequency, and safe enough not to cause any injuries even in case of failure has made the design of gas turbine engine parts challenging.

Although bolted joints have been the most frequently preferred connection method in gas turbine engines for many years, numerous failures have occurred in bolted joints due to excessive or insufficient preload. One of the main reasons for this situation is the high preload variation values of torque-controlled tightening, which is the most commonly used tightening method to provide preload.

The main motivation of this study is to explore the possibility of lower preload variation of the torque-angle tightening method as an alternative to the torque-controlled tightening method. Additionally to investigate whether the tightening parameters of the torque-angle method can be estimated with a finite element analysis before the joint-specific tests.

1.2 Objective and Impact of Research

The objective of this research is to examine the process of the torque-angle tightening method, to provide ease of use and to demonstrate how to determine the tightening parameters, and also to prove that a design improvement in a critical bolted joint can be achieved by simply changing the tightening method, rather than an expensive design change.

In this study, tightening parameters are determined for both methods with the torque-tension tests. The linear relationship between the angle and preload in the test results is obtained by finite element analysis. In addition, a method is proposed to decide the snug torque value. Finally, it has been shown that for the critical bolted joint design of the T800-LHT-800 engine, an increase in minimum and average preload values can be achieved by simply changing the tightening method.

1.3 Thesis Outline

This thesis is divided into four chapters: Chapter 1 addresses the purpose, impact, and motivation of research and introduces the importance of preload accuracy for critical bolted joint design. In Chapter 2, detailed literature information is shared about gas turbines, bolted joints and their tightening methods. Chapter 3 details the methods of this research which includes torque tension testing, determining tightening values and comparison of the torque and torque-angle controlled tightening methods through an assembly testing. Finally, Chapter 4 summarizes the research and its contribution to the literature and discusses possible future extensions of the research.

2. THEORY

2.1 Gas Turbine Engine

Gas turbine engines are undoubtedly one of the most important inventions of the 20th century. Before the Second World War gas turbine development began, but these applications were for electrical energy applications, not for engines as alternatives to diesels and steam turbines. With the first military jet engine developed towards the end of the Second World War, this technology provided a significant increase in flight speed compared to existing engines. Although the early applications of these engines had many negative outcomes such as low fuel efficiency, excessive noise, and safety problems, they became the standard form of propulsion for aircraft in as little as two decades. Today, with the development of turbofan engines with high bypass ratios, it has made possible the wide-body aircraft, which is the most important means of transportation in civilian applications. Today, gas turbine engines are used as a power source in many markets such as power generation, oil and gas industry and marine propulsion due to their high power and thermal efficiency in land applications [8].

2.1.1 Gas Turbine Thermodynamic Cycle

The thermodynamic cycle of gas turbine engines derives from the ideal Brayton cycle. The Brayton cycle, which is a constant pressure cycle for the perfect gas, is also known as the Joule cycle. In Figure 2.1, the representation of the Brayton cycle and T-S, P-V diagrams are given. From station 1 to station 2, air is compressed isentropically from pressure P_1 to pressure P_2 . The temperature rises from T_1 to T_2 without heat flow occurring. While passing from station 2 to station 3, the heat entering the system increases the volume of air from V_2 to V_3 under constant pressure, while the temperature rises from T_2 to T_3 . From station 3 to station 4, the air expands isentropically from pressure P_2 to pressure P_1 . The temperature drops from T_3 to T_4 without heat flow occurring. Finally, the heat is removed from the system, and the volume of the air decreases from V_4 to V_1 , and the temperature decreases from T_4 to T_1 . The efficiency of the ideal Brayton cycle is obtained by dividing the work done by the heat energy entering the system [9]

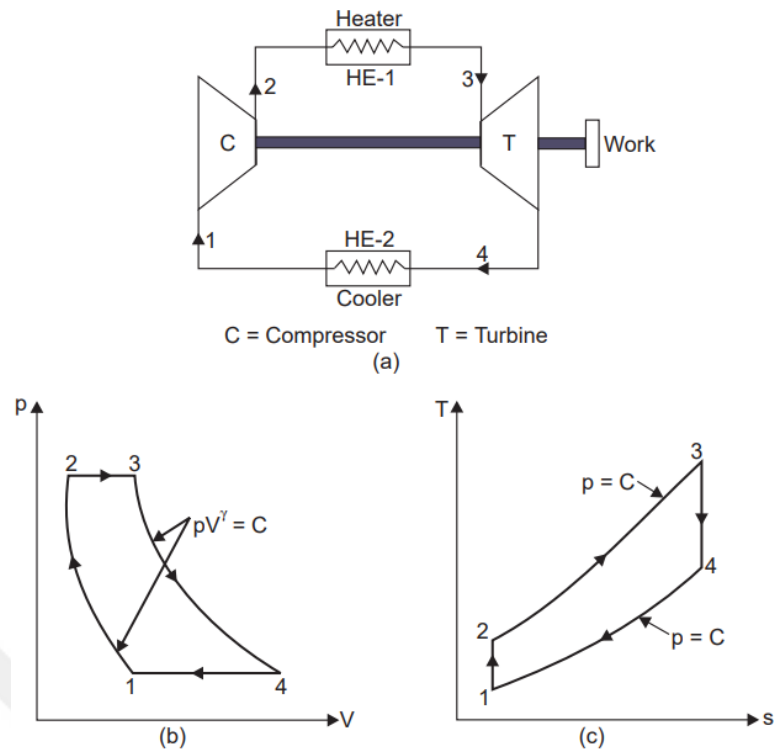


Figure 2.1 (a) Brayton cycle (b) P-V diagram (c) T-S diagram [9].

The gas turbine engine cycle works on the principle of the Brayton cycle but is an open cycle, no air is recirculated. The ideal open cycle gas turbine is given in Figure 2.2. Accordingly, the pressure and temperature of the ambient air drawn into the compressor increases. High-pressure air is burned at constant pressure in the combustion chamber. Then, as the high-pressure and high-temperature air exiting the combustion chamber expands to atmospheric pressure in the turbine, power is generated. Finally, the exhaust gases from the turbine are not recirculated and thrown into the atmosphere [10].

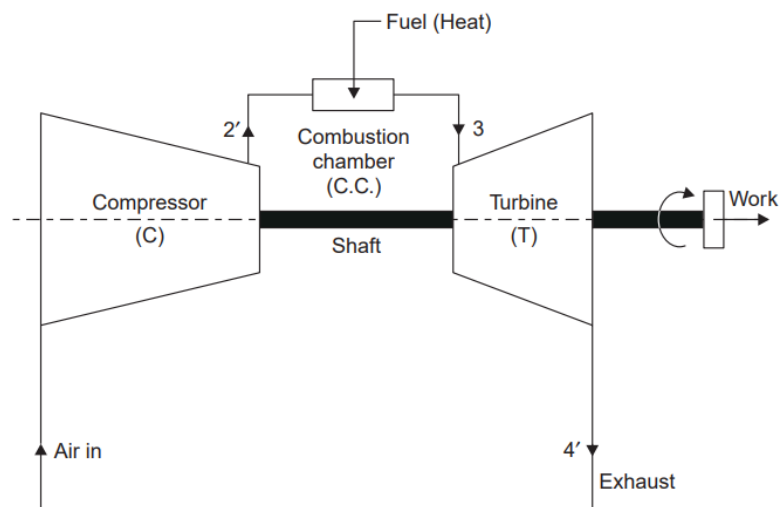


Figure 2.2 Open cycle gas turbine [9].

2.1.2 Gas Turbine Design

Gas turbine has many components that limit the engine life in all applications. Thermodynamic cycle efficiency is not the only parameter for gas turbine efficiency. Acceptable durability limits and maintenance periods of structural elements are also important efficiency criteria for gas turbines. If a gas turbine with very high thermodynamic efficiency is damaged by structural phenomena such as low cycle fatigue, creep, high cycle fatigue, oxidation, the damage can be catastrophic for the rest of the engine. For safe and reliable operations, the durability of the engine is critical [11].

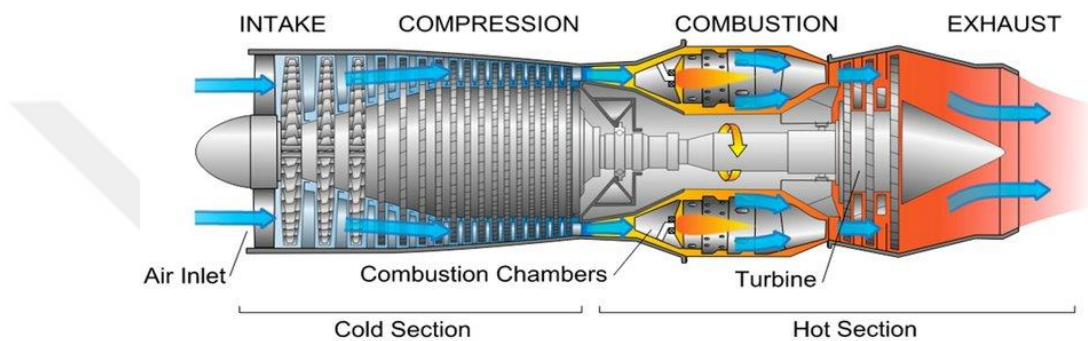


Figure 2.3 Gas turbine engine components [12].

Gas turbines basically consist of inlet, compressor, combustion chamber, turbine and exhaust main parts given in Figure 2.3 [12]. Compressor and turbine components shown in ideal cycle can consist of many stages in gas turbine engines. Compressor and turbines are connected to each other by shafts. Aircraft engines and aircraft-derivative engines may have more than one shaft. These shafts can contain high, medium, and low-pressure compressor units (HPC, IPC, LPC). Compressor parts on the respective shaft are connected to the turbine parts in multi-shaft configurations as well as in single shaft configurations. In addition to the parts that support these rotating parts, gas turbine engines have complex designs, as shown in Figure 2.4, when considering the parts that will perform the necessary subsystem functions such as secondary air systems and lubrication systems [13].

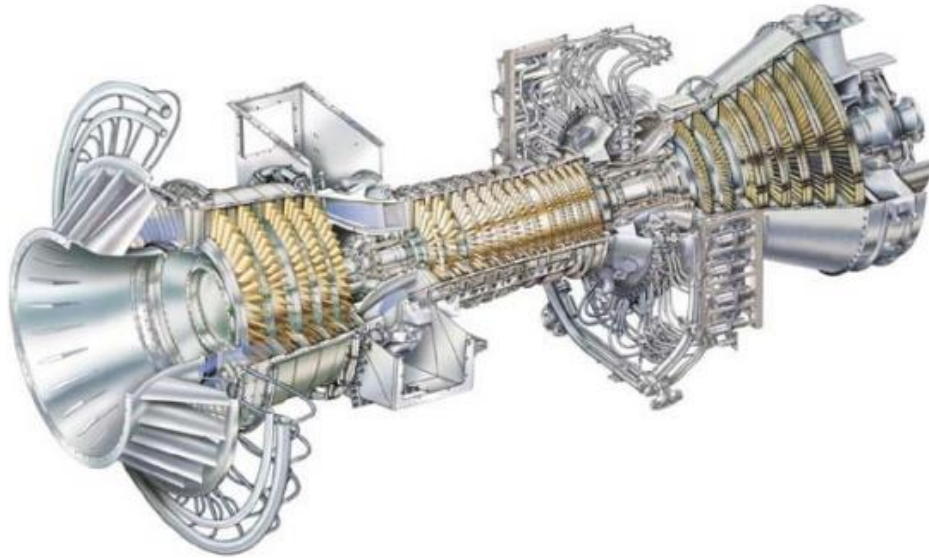


Figure 2.4 The GE LM6000 (aeroderivative of the CF6-80C2) [14].

Gas turbine engines are designed with a modular design philosophy to reduce maintenance costs, shorten maintenance time, meet maintenance requirements more easily and reduce transportation costs [15]. Aligning the modules, matching the contact surfaces, and maintaining the clearances is a challenging task when reassembling in modular gas turbine engines consisting of many parts, as in Figure 2.5 [16].

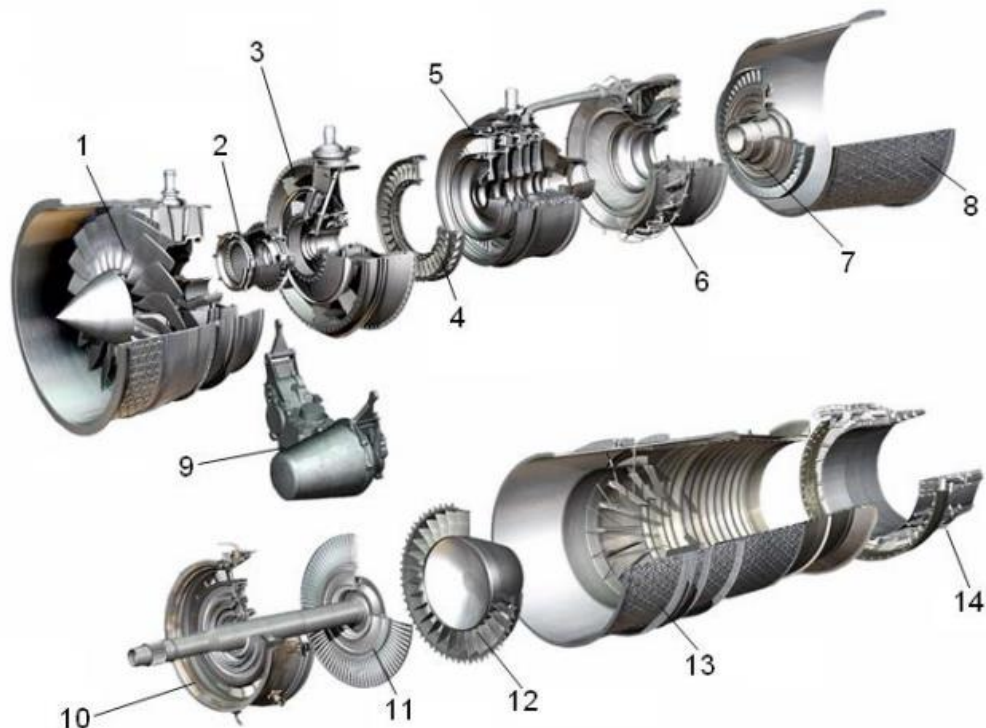


Figure 2.5 Modules GTE EJ 200 1...8, 9 - cold part; 6, 7, ...14 - hot part [17].

2.1.3 Bolted Joints in Gas Turbine Structures

Bolted joints connections, which have the ability to maintain their structural integrity under high level of loads, and are more reliable than alternative connection types in terms of sealing and engine alignment, are preferred in the many connection interfaces found in modern gas turbine engines [2]. Bolted joints with low-density materials are used in the cold sections of gas turbine engines in order to have engines with low weight and with low fuel consumption. Bolted joints with high temperature resistant material are also frequently preferred in the hot section of gas turbine engines [18]. Figure 2.6 shows schematically the major structural bolted joint connections of gas turbines [2].

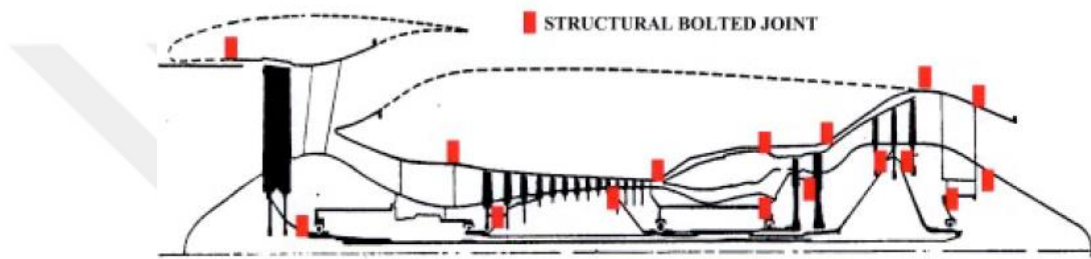


Figure 2.6 Major structural bolted joints in a gas turbine engine [2].

2.1.4 Component Classifications

An overview of the safety assessment process, which includes System Safety Assessment (SSA), Preliminary System Safety Assessment (PSSA), and Functional Hazard Assessment (FHA), is given in Figure 2.7. This figure also shows the place and relationship of safety assessment methods in the process [19].

Gas turbine engines must meet the safety requirements determined by authorities such as the European Aviation Safety Agency (EASA) and the Federal Aviation Administration (FAA), as well as capability requirements such as performance, fuel consumption, and durability. There are many parts in gas turbine engines that will not cause major damage to the engine and aircraft, even if they lose their functionality, the effect of which will only result in a decrease in the performance of the engine. These parts are called "in condition" and are replaced in the event of damage detection or according to the standard maintenance schedule. In case of loss of functionality, the parts that can damage the engine or the aircraft are called engine critical parts [20].

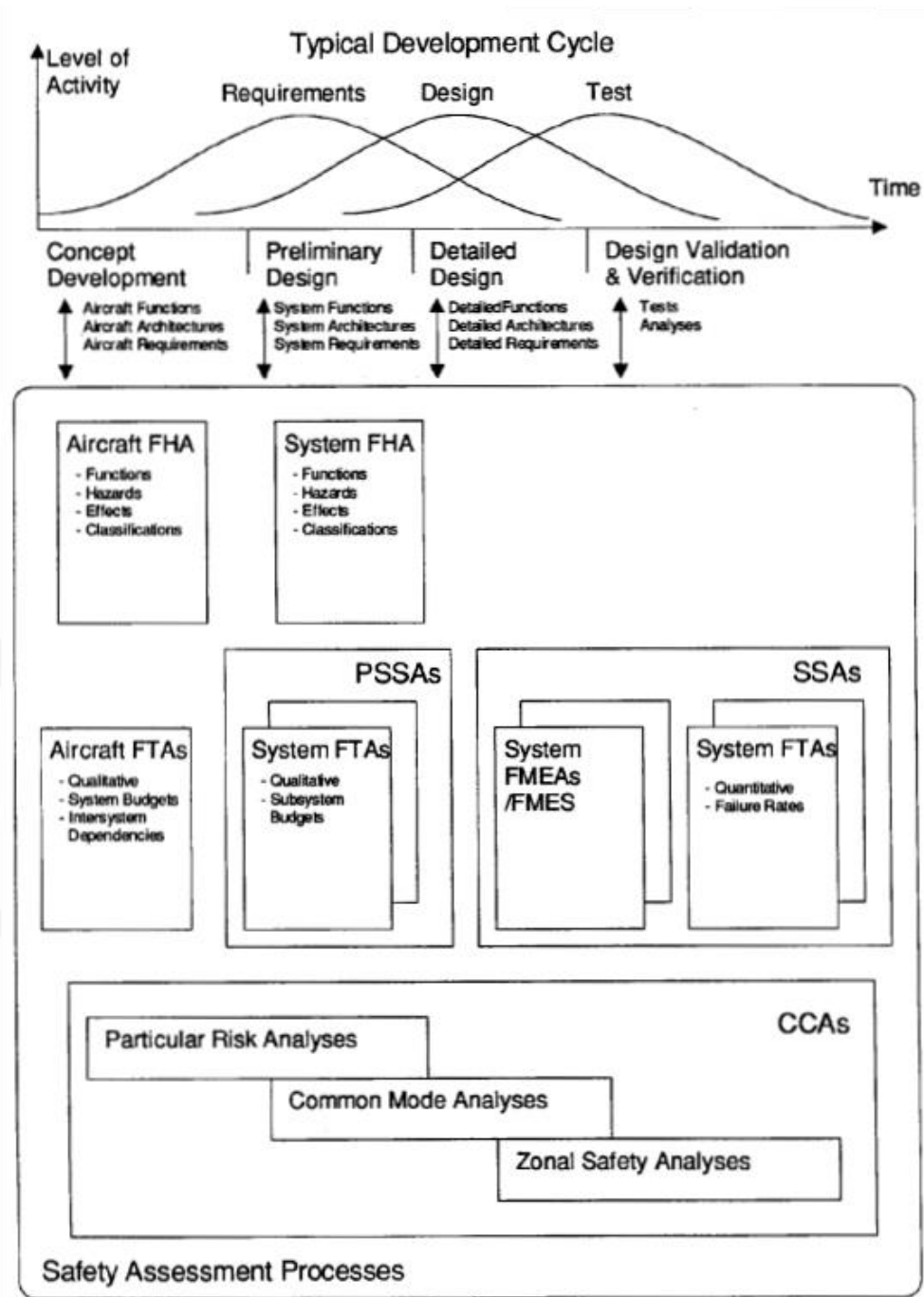


Figure 2.7 Overview of the safety assessment process [19].

Compliance for Engines (CS-E) Amendment 5 identifies a critical part as one whose failure could result in hazardous engine effect [21]. The effects that may cause dangerous engine effects can be listed as follows; non-containment of high-energy debris, high concentrations of toxic products in the bleed air, sufficient to incapacitate passengers and crew, significant thrust in the opposite direction to the direction commanded by the pilot, complete inability to turn of the engine and uncontrolled fire [19].

2.2 Bolted Joints

Correctly designed bolted connections are frequently used in modern engines and engineering structures due to their low cost and reliability [22]. In the structure and the engine of an aircraft, hundreds of bolt designs of different types, sizes and materials are used. Taking the Boeing 747s as an example, this plane has 2.4 million fasteners apart from its engines, of which 22% are structural bolts [23].

Regulation agencies, such as the FAA and the EASA require a set of requirements that must be met in order to guarantee the airworthiness of these parts [24]. In normal operation, bolted joints are designed considering high internal pressures, high rotational speeds, and high temperature levels, as well as weight and packaging concerns. In addition, transient operating conditions during engine starting, acceleration or deceleration are taken into account, which may cause the bolted joints to be subjected to various combinations of loads during normal operation. Moreover, bolted joints must be designed in such a way that they are not subjected to plastic deformation even under the limit loads caused by the maneuvers of the aircraft. Finally, one of the most challenging regulation requirements for the bolted joint design in modern engines is the requirement to maintain the integrity of the rotor and carcass structure, even under ultimate unbalance, which may occur due to loss of fan blades, bird strike or any abnormal event [2].

2.2.1 Basics of Bolted Joint Design

Most general bolted joint design which is given with Figure 2.8 consists of four main parts such as an external threaded bolt, clamped components, also called joint members, an internal threaded nut and washer. The bolted joint works according to the principle that when the nut is advanced by twisting, the stretched bolt creates a clamping force called preload on the clamped parts [25].

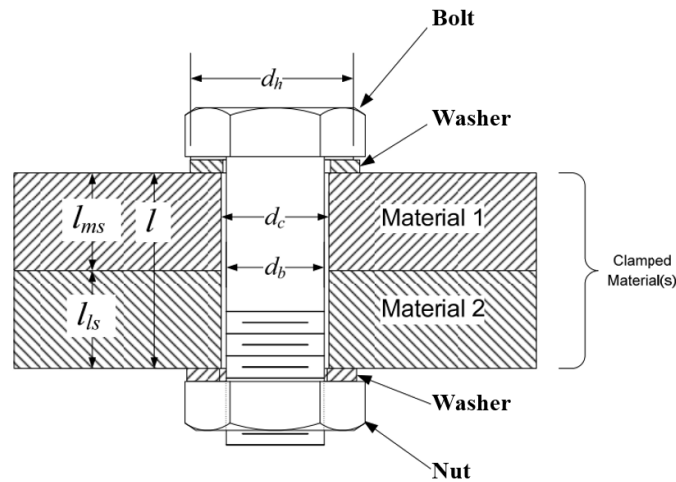


Figure 2.8 General model of the bolted joint design [23].

Tension joints and shear joints are the types in which bolted joints are named according to the load types they are subjected to. In the tension joints shown with the upper drawing in Figure 2.9, the load is parallel to the bolt axis. In the shear joints in the lower drawing, the load is perpendicular to the bolt axis. Tension joints preferred for the joint designs due to the complex loads in gas turbine engines must be clamped together with sufficient preload to prevent joint separation and leaks. Tension joints in gas turbine engines may have been exposed to shear loads, so these bolts are also responsible for preventing the joint elements from slipping [5].

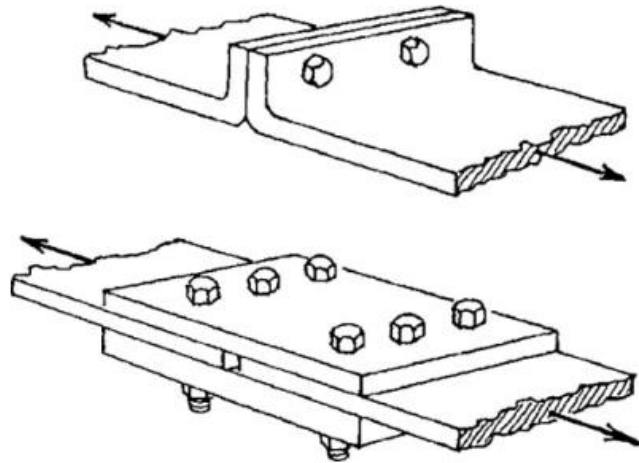


Figure 2.9 Main joint designs according to subjected load types [5].

Bolted connections are actually alive structures with a certain potential energy. This energy must be large enough that it is not completely lost under vibrations, thermal cycles and shocks that may be exposed during operation. However, the preload that will provide this energy must be low enough not to damage the joints and the bolt. Both the bolt and joint members can be thought of as springs as shown in Figure 2.10.

While a bolt assembly takes place, the structure generally behaves elastically. The energy generated between the tensioned spring (bolt) and the compressed spring (joint) during assembly must be maintained throughout the life of the connection [5].

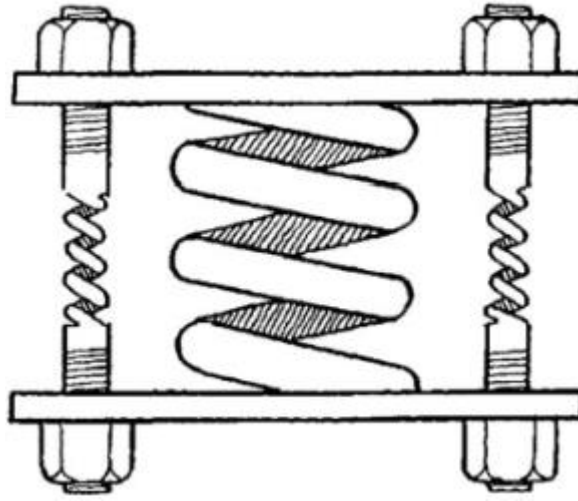


Figure 2.10 Spring representation of bolted joints [5].

2.2.2 Bolted Joint Stiffness

Although bolted joint designs are used frequently, they are structures that require attention and provide the promised reliability as long as they are designed correctly. The spring constants of the bolt and joint members determine the characteristics of bolted joints and their behaviour. The relative size of the spring constants of bolts and joint members relative to each other is also an important factor for bolted joint design under external loads. These spring constants are called stiffness [26].

2.2.2.1 Bolt stiffness

Calculating the stiffness of the bolt is much easier than calculating the stiffness of the joint members, thanks to its uncomplicated behaviour [26].

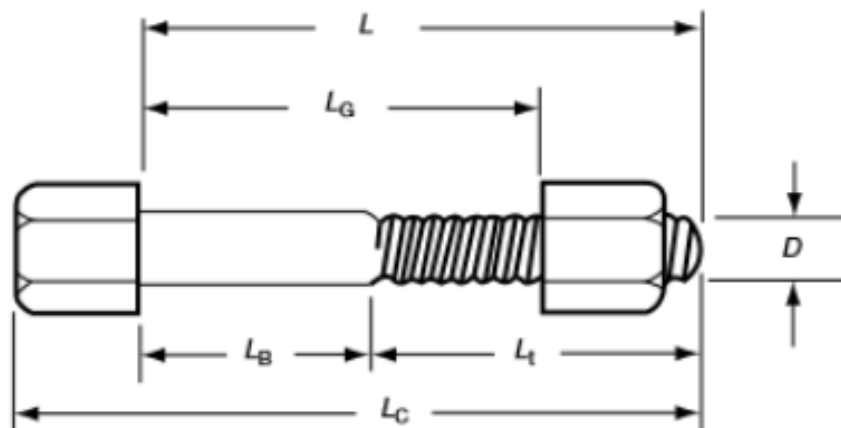


Figure 2.11 Bolted joint dimensions [5].

According to Hooke's law, the deformation of a bar of constant cross-section is defined by Equation (2.1) where F is the applied tensile force, E is the modulus of elasticity, L is the length of the section and A is the cross-sectional area [27].

$$\Delta L = \frac{FL}{AE} \quad (2.1)$$

Considering the another simple material principle, in a standard tensile test when an axial tensile load is applied to a straight cylindrical bar, a linear relationship between load and deflection is obtained up to the yield point. If this linear relationship given by Equation (2.2) is combined with Equation (2.1) and rewritten as given in Equation (2.3), the stiffness value, k , is obtained.

$$F = k \Delta L \quad (2.2)$$

$$k = \frac{AE}{L} \quad (2.3)$$

The stiffness of the bolt depends on the modulus of elasticity, the number of sections, and the cross-sectional area and length of each of these sections. Assuming that the material chosen for the bolted joint design is homogeneous, only the length and the cross-sectional area are variable. Although the cross-sectional area and length calculations can be easily obtained for many sections of the bolt, it is difficult to determine the contribution of the bolt head and the threads engaged with the nut to the stiffness. However, according to the experiments, it is obvious that a section of more than the length defined as the grip length between the bottom of the bolt head and the beginning of the nut contributes to the stiffness. The length of the bolt that affects the stiffness is defined as the effective length [28]. VDI 2230 [29], an accepted standard for bolted joint design, recommends using 0.4 times the nominal bolt diameter as the length value for the bolt head and engaged threads when calculating the stiffness of both regions, while Bickford [5] recommends 0.5 for this coefficient.

The formulation in Equation (2.4) simply expresses the stiffness of the bolt if the difference between the thread diameter and the nominal diameter is neglected, or if an all-thread bolt is assumed.

$$k_B = \frac{EA_s}{L_e} \quad (2.4)$$

Although much more complex formulations can be derived for the stiffness value of the bolt (k_b), it is possible to make a good estimation by using the effective length (L_e) and the tensile stress area (A_s) for calculating the stiffness value of the bolt. There may be a 4% difference between the theoretical stiffness calculations and the actual

design due to geometric tolerances, and 3% due to the variation in the elasticity modulus. These variation values can be reduced according to the need for precision, but this will create extra cost for the design [5].

2.2.2.2 Joint members stiffness

Many different studies have been carried out to approximate the stiffness of the joint members, whose behaviour is much more complex than the bolt. The most important reason why many studies have been carried out on estimating the stiffness of joint members is that this value is crucial for the accuracy and reliability of the design of the bolted joint. One of the most common methods for calculating the stiffness of fasteners was proposed by Rotscher. In this approach, illustrated in Figure 2.12, the stress area of the bolt is assumed to be in the form of a cone [26].

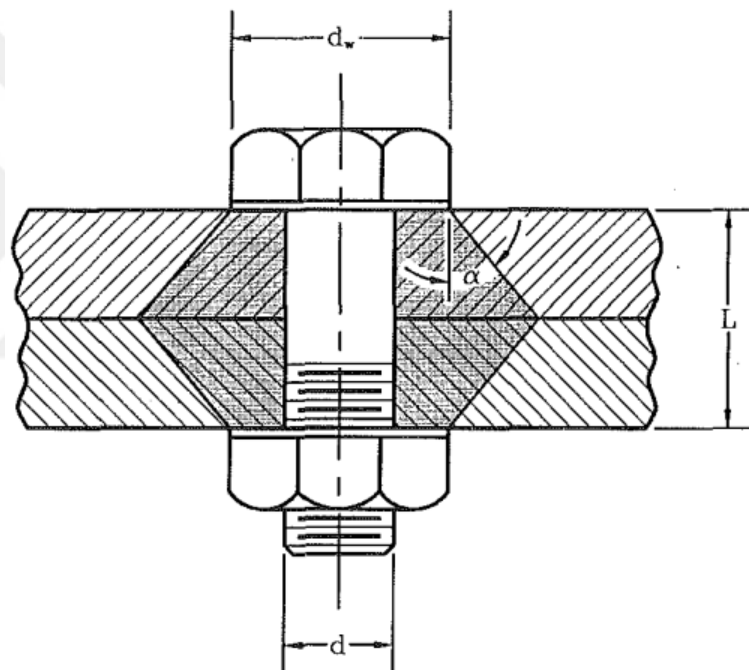


Figure 2.12 Schematic of a typical bolted joint connection [26].

Today, although there are studies that suggest different angle values by preserving this conical approach, or studies that develop different approaches, all of these studies are based on arbitrary approaches. The study of Wileman, Choudury and Green comparing validated finite element analyses with other methods is given in Figure 2.13 [26].

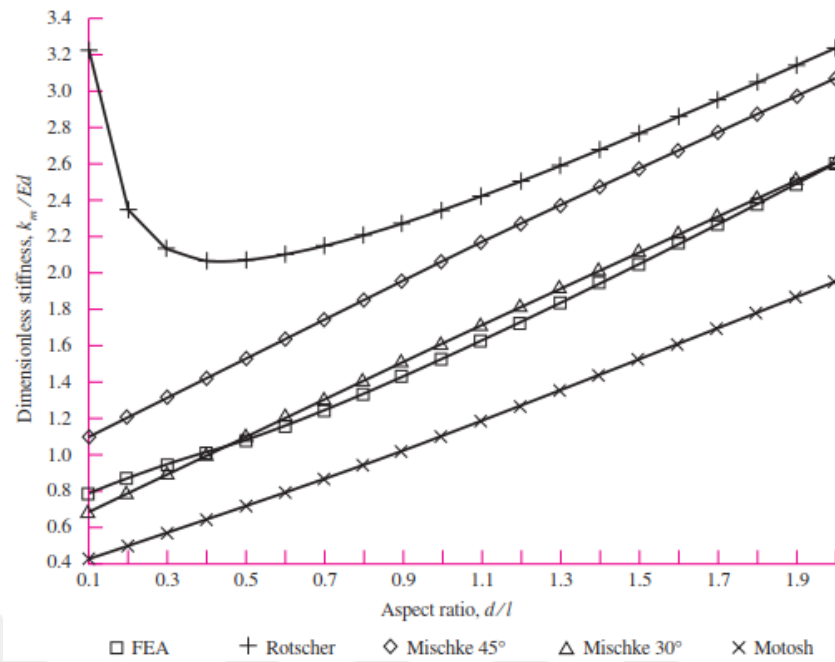


Figure 2.13 Comparison of different approaches for joint member stiffness [1].

2.2.3 Assembly of Bolted Joints

The main purpose of the assembly process is to create the preload between the bolt and joint members by providing the necessary energy. The accuracy of the given preload is critical, as the life and behaviour of the bolted joint is highly dependent on it. While high preload values must be applied to the bolt to prevent many failure modes, at the same time the preload must not exceed certain maximum values to meet the strength expectations of the bolt and the joint. The most common example of this trade-off is the bolted joint, which is subjected to tensile loads, a situation that is most undesirable but inevitably occurs in gas turbine engines. When this happens, the tensile load increases while the clamping load between the flanges decreases. In fact, it is expected that the preload given to this structure at the assembly stage should be low enough not to exceed the yield strength of the bolt and high enough that the flanges will not separate. Other problems that low-precision preload can cause can be listed as follows: static failure of the fastener, static failure of joint members, vibration loosening of the nut, fatigue failure of the bolt, stress corrosion cracking, joint separation, joint slip, excessive weight, excessive cost [5].

Since the accuracy of the preload is an important parameter for the design of the bolted joint, many assembly methods such as torque control, turn control, torque-angle control, load-indicating washer, bolt stretch control, have been developed to control this parameter [30].

Methods such as bolt stretch control, strain gauged bolt and load-indicating washers are methods that are generally used in experimental studies, providing low preload variations, but with high application cost and difficulty. The most frequently preferred method in the industry is the torque-controlled tightening method due to its ease of use. Since it has been used for many years, even standard torque tables have been created for different bolt sizes, but the preload variation of this method is between 25% and 35% according to various sources in the literature [5].

2.2.4 Tightening Process

The tightening process, in its most basic terms, is the storage of a certain amount of energy on the bolted joint by creating a certain preload on the clamped parts while the bolt is stretched, with the energy given to the system. The energy given to the system to tighten the bolted joint is the area under the torque angle diagram given in Figure 2.14 [4].

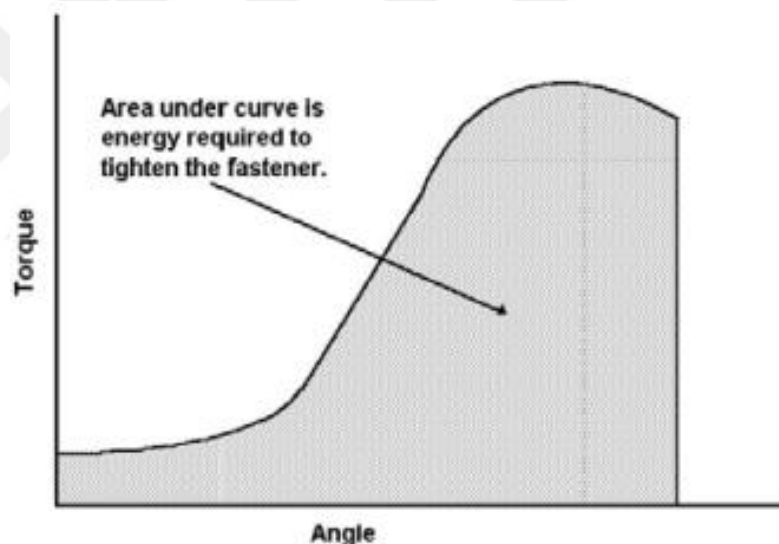


Figure 2.14 Tightening fasteners transfers energy [4].

In the tightening process, 90% of the energy given to the system is lost by turning into heat with friction losses. Forty percent of this loss is due to friction between the threads, and 50% is due to friction between the mating surfaces of the bolt, nut and the joint members. As a result, only 10% of the energy given to the system is transferred to the structure as a preload. The energy conversion is shown schematically in Figure 2.15 [5].

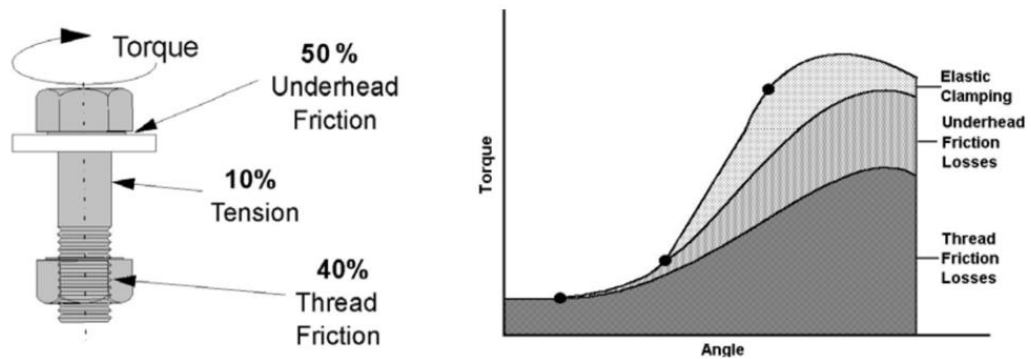


Figure 2.15 Energy conversion in tightening [4].

Before examining the tightening methods, it is necessary to correctly understand the nature of the tightening process and the relationship between torque, angle, and preload parameters. The most general graphic showing the tightening process is given with Figure 2.16. The tightening process basically consists of four main zone. The first region is the rundown zone, which is the process until the bolt or nut surface contacts the seating surface. If there is no locking mechanism, torque and preload values are not expected to be observed in this region. However, if there is a locking mechanism or deformed threads, a prevailing torque is expected in this region. The second region, called alignment zone, is the non-linear region where the parts begin to deform and preload begins to occur. There are many effects, called micro and macro, that affect the behaviour of this region. Manufacturing or design defects can be detected by examining the behaviour of this region. The third region is the elastic clamping region, where the torque-angle and preload-angle values increase linearly. For each bolt, the slope of the curve in this region is characteristic. Finally, in the post yield region, the bolt starts to show plastic deformation and goes towards the failure [4].

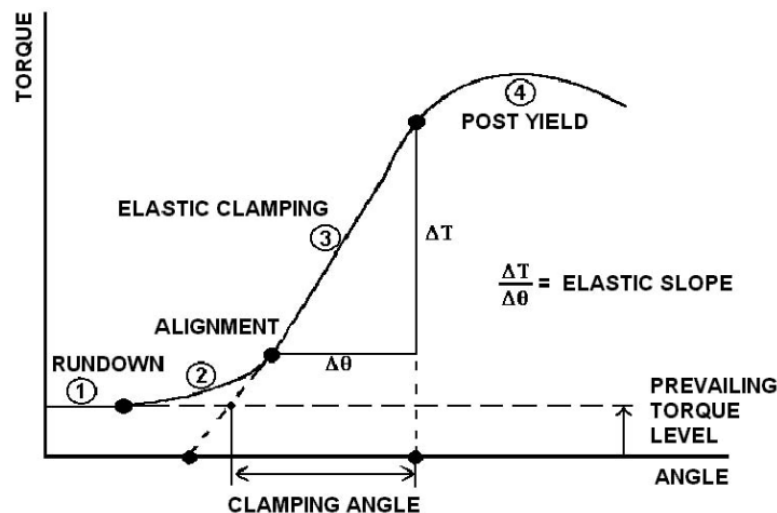


Figure 2.16 Regions of the tightening process [4].

2.2.4.1 Rundown zone

The rundown zone is the beginning of the tightening process, which actually corresponds to manually inserting the bolt into the joint. If there is no locking mechanism, thread forming, or break through torque in the design, preload and torque are not expected in this region.

Except for the cases given in Figure 2.17, undesirable situations such as misalignment of parts, out-of-tolerance threads may also cause prevailing torque to be observed. Thus, production or design problems can be detected. Chips remaining between the teeth may also cause low amounts of torque to be observed in this region [31].

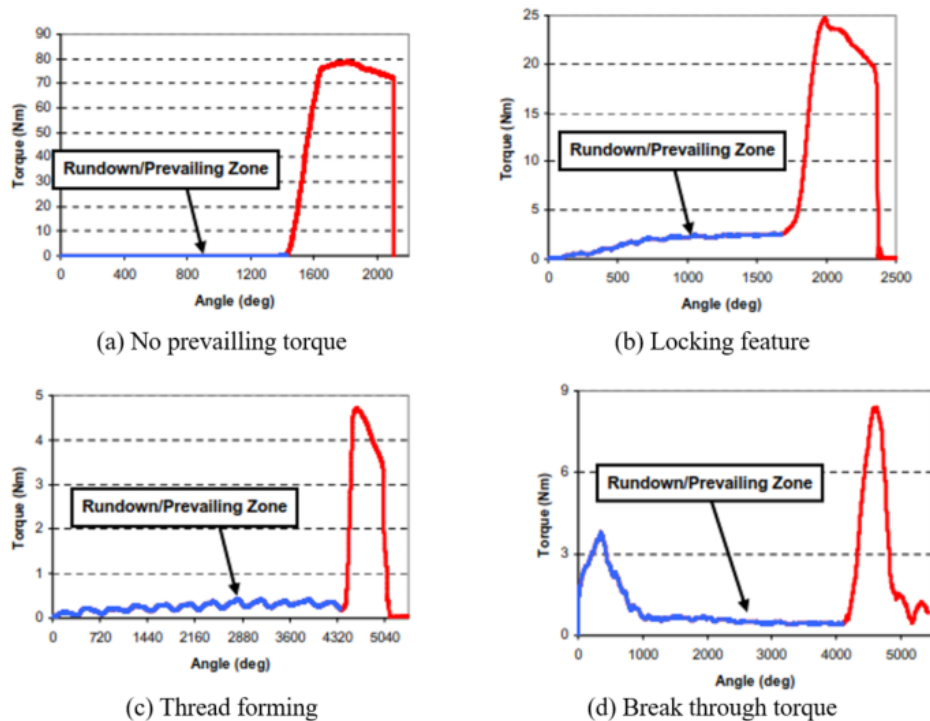


Figure 2.17 Different rundown region behaviours [31].

2.2.4.2 Alignment zone

With the end of the rundown zone, the alignment zone begins where the joint members and the bolt begin to deform together. In this region, torque and preload values start to increase in bolted joint, but there is no linear relationship between torque and angle or between preload and angle. Due to the non-linear behaviour of the bolted joint in this zone, the data in this region is not preferred as a control parameter in tightening methods [4].

The width of the alignment zone depends on the stiffness of the bolt; this region is expected to be narrow for bolted joints with high stiffness, while it is expected to be high for those with low stiffness [31].

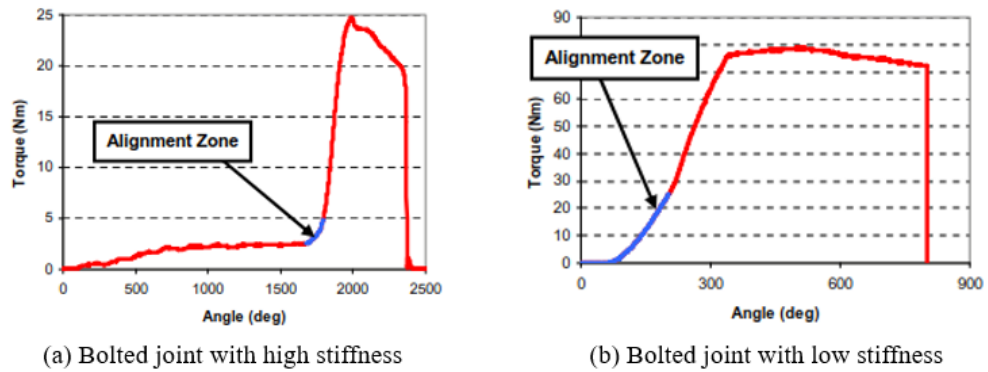


Figure 2.18 Different alignment zone behaviours [31].

In addition, many factors, categorized as micro and macro, affect the behaviour of this region. While macro effects can be listed as the factors that have an important effect on the general behaviour of the structure, such as bending of the parts together, pulling the parts together and bending of the bolt due to the non-parallel surface, factors such as surface hardness and thread deformations create micro effects [31].

Since the torque-angle plots directly reflect the behaviour of the bolted joint, design and manufacturing errors can be detected upon careful inspection. Errors that cause macro effect drastically change the behaviour of the alignment region.

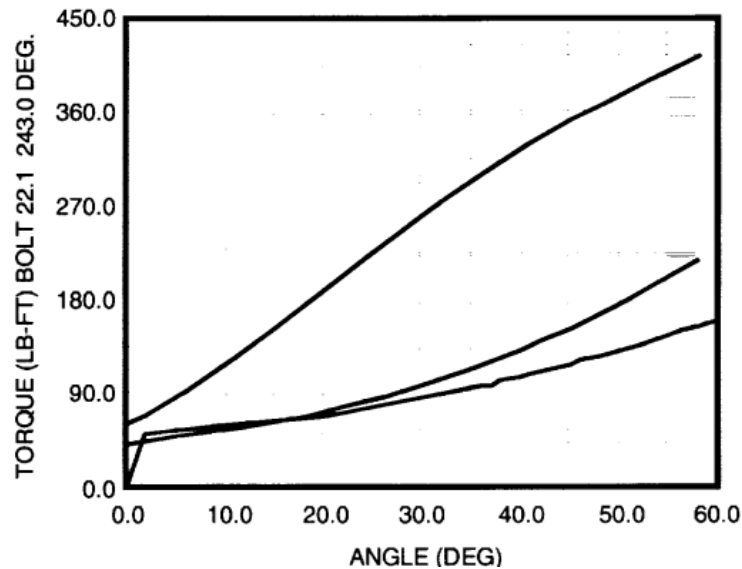


Figure 2.19 Abnormal torque-angle curve [3].

Thompson, Nayate and Smith in their study, where they showed that the preload variation of a critical bolt can be reduced by using the angle of twist method, they examined the torque-angle graphs and detected many errors arising from design and

production. They demonstrated that one of the situations that caused the abnormal behaviour given in Figure 2.19 was due to the interference of the washer radius with the bolt shank radius. The given energy was spent on deforming the washer radius instead of stretching the bolt. Again, in the same article, it was claimed that problems such as mismachined holes and incorrectly manufactured bolts were determined by examining the torque-angle graphs [3].

2.2.4.3 Elastic zone

With the end of the alignment zone, the elastic region begins, in which a linear relationship is observed between the torque-angle and between the preload-angle. The transition point from the non-linear region to the linear region between the two regions is called snug point.

The slope of the linear curve is characteristic for each bolted joint and is a function of the stiffness of the structure. While the slope is high in bolted connections with high stiffness, the slope is low in bolted connections with low stiffness [5].

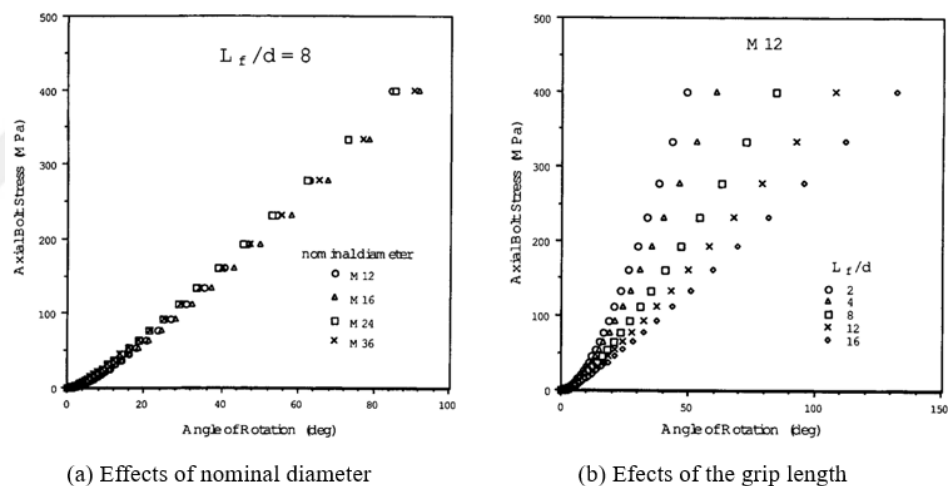


Figure 2.20 Effects of nominal diameter and grip length on the elastic curve [6].

In one of his studies, Fukuoka established very low changes in the elastic region in his experiments for standard bolts of different sizes with the same grip length divided by nominal diameter ratio. On the other hand, he reported that there are significant differences in the elastic region when the grip length is changed while keeping the nominal diameter constant. As shown in Figure 2.20, lower sloping curves were obtained for soft bolts with higher grip lengths [6].

The slope of the torque-angle curve changes if the coefficient of friction is changed under constant stiffness. In higher frictions, the desired energy is obtained by

turning the bolt less, while in low frictions the same energy is obtained by turning the bolt more. Torque-angle graph is given in Figure 2.21 when the bolted connection with the same stiffness is tightened to the same torque value under different coefficients of friction [31].

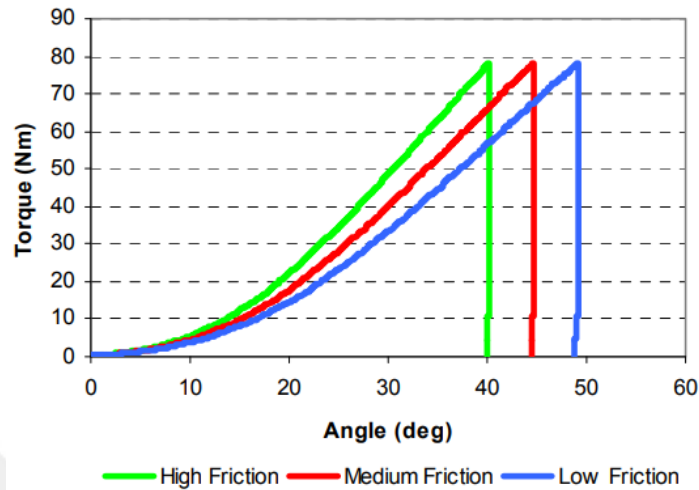


Figure 2.21 Friction effect on torque angle relationship [31].

Although many sources say that the coefficient of friction has no effect on the slope of the preload angle elastic curve, this is actually not true. This can only be achieved if the angle of rotation in the theoretical formulation is controlled relative to the bolt, which is very difficult. For this reason, the coefficient of friction will also have some effect on the preload-angle curve [5].

The comparison of Fukuoka's numerical models verified with experimental results for different friction coefficients is given in Figure 2.22. The first of these graphs is an indication that the comparison under different friction coefficients is made under equivalent stiffness. The second graph shows the effect of the friction coefficient directly. In this study, it has been reported that the coefficient of friction has an effect on the slope of the curve, but this effect is quite small [32].

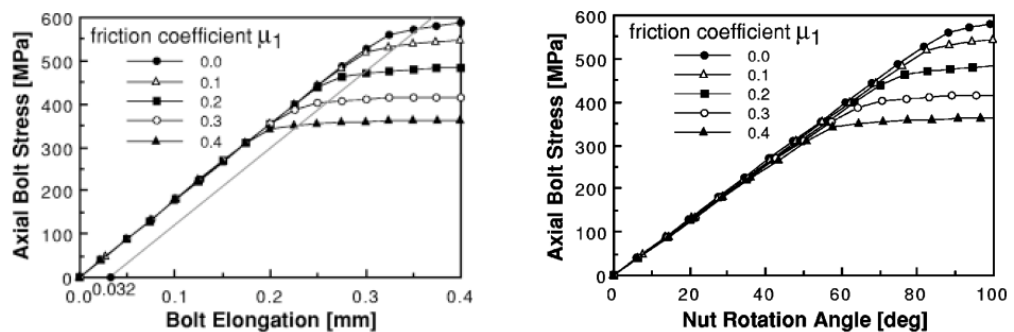


Figure 2.22 Friction effect on preload angle relationship [32].

Another factor affecting the behaviour of the elastic curve is called embedment. Theoretically, the seating surfaces can be described as perfectly flat, but in reality, they have a rough structure consisting of hills and valleys, albeit at a microscopic level. Rough surfaces cause plastic deformation of the seating surfaces of the contacts that rub against each other during the tightening process. Although these plastic deformations do not completely change the stiffness of the bolted joint, they somewhat distort the linear structure of the elastic curve as shown in Figure 2.23 [5].

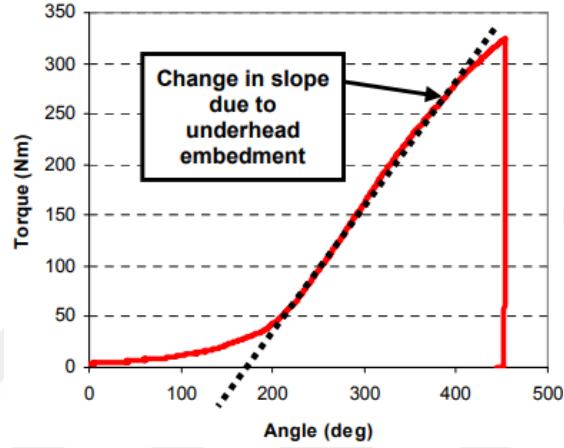


Figure 2.23 The effect of embedment on elastic slope [31].

2.2.5 Torque-Controlled Tightening

Among the tightening methods, the most frequently used method is the torque tightening method due to its ease of use and cost-effectiveness. It is a method of estimating the applied preload by controlling the torque value applied to the bolt.

Due to many variables, high variations are observed in the preload value [33]. Although the torque tightening method is the most commonly used method, it is difficult to use in applications with high preload sensitivity due to the high preload variation [34]. Different values given by different sources for the preload variation vary between 25% and 35%. Bickford gave an average of 30% for the preload variation, while Lyndon separated lubricated and unlubricated bolts, reporting 25% and 35%, respectively [5], [35]. Ford, on the other hand, separated the assemblies as on the operator's judgement and wrench controlled, and published the preload variation values of 35% and 25%, respectively [36].

According to Shigley, Equation (2.5) is a good estimation for calculating corresponding torque to the required preload.

$$T = \frac{F d_m (\mu \pi d_m + L \cos \theta)}{2 (\mu \pi \cos \theta + \mu L)} + \mu_c F \frac{d_c}{2} \quad (2.5)$$

Shighley made a few manipulations in this equation and rewritten it as in Equation (2.6) where $\gamma = L/\pi d_m$, $d_c = 1.25d$.

$$T = \left[\left(\frac{d_m}{2d} \right) \left(\frac{\tan \gamma + \mu \sec \theta}{1 - \mu \tan \gamma \sec \theta} \right) + 0.625 \mu_c \right] Fd \quad (2.6)$$

And finally, the equation is simplified as in Equation (2.7), and the part of the equation in square brackets is defined as the K factor [1].

$$T = KFd \quad (2.7)$$

The formula shows that there is a certain ratio between torque and preload. The K factor is not the friction coefficient, but a simple representation of each factor that directly affects the ratio between the two parameters when the relationship between torque and preload is examined experimentally. Examples of effects that are part of the K factor are friction coefficient, bolt size, torsion, bending, plastic deformation of threads etc. Although the theoretical formulation of the K factor gives a correct approach to the effect of the relevant parameters on the K factor, the K factor is usually determined by experimental results due to the difficulty of determining these parameters. In fact, in some studies, tool accuracy, bolting procedure and operator skill are also included in the K factor calculations [5].

In his study, where he examined the effect of bolt sizes on the K factor, Brown showed that the effect of bolt size on the K factor was limited in bolts under $1^{1/4}$ size. He also stated that the K factor and friction factor showed parallel behaviour for all the bolt types he used. Figure 2.24 shows Brown's study for two different bolt materials under different lubrication conditions and different grip lengths [37].

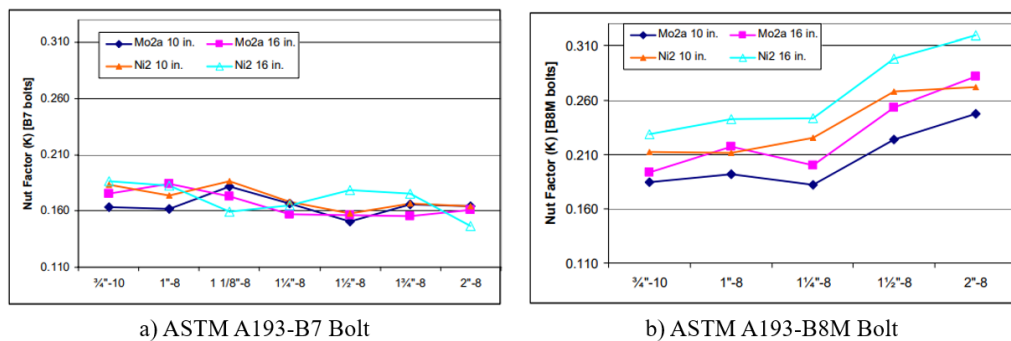


Figure 2.24 Effects of bolt size on K factor for different materials [37].

The relationship between torque and preload highly depends on the coefficient of friction between the threads, and between the nut and the bearing surface [28].

2.2.5.1 Friction effect on preload variation

A total of 90 percent of the energy given to the system during the tightening process is lost as heat due to the friction coefficient which is the main reason for the high variation of the torque control tightening method. Bickford explains the variation in the preload value caused by the change in friction coefficient as follows. If the friction coefficient is 10% more, it can increase the energy lost due to friction from 90% to 95%. Although this situation does not seem significant when evaluated on its own, considering that the energy supplied to the system is constant, this means that the energy allocated for stretching the bolt has decreased from 10% to 5%. A 5% reduction in the energy devoted to stretching the bolt means a 50% reduction in the target preload value [5].

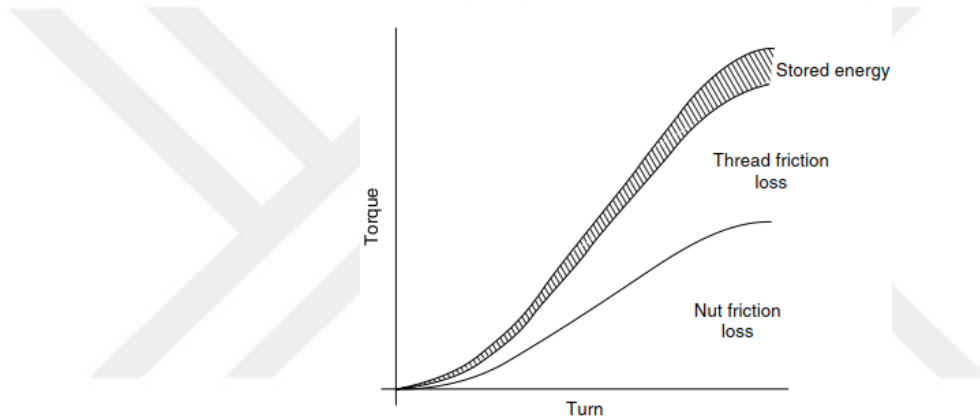


Figure 2.25 Energy stored compared to energy consumed during the tightening process [5].

It is estimated that the friction coefficient is a property dependent on nearly 200 different parameters [4]. Although there are different studies in the literature on many parameters on which the friction coefficient depends, they help us to understand which parameter affects the friction coefficient and how. It is quite difficult to parameterize the coefficient of friction in a formula. The friction coefficient is generally determined by experimental results [38].

Some of the factors affecting the friction coefficient can be listed as follows:

- Lubricant and its type, amount, condition, method of application, contamination, and temperature
- Hardness values of the components
- Type of materials
- Surface finishes
- Thickness, condition, and type of plating, if any

- Tightening speed
- Manufacturing process, cut versus rolled threads
- Hole clearance surface pressures
- Presence or absence of washers

Although it is not possible to control all of these factors, control accuracy can be increased by keeping these factors under control in order to reduce the variation they will create. Otherwise, a randomly selected bolt under the same specified torque value as shown schematically in Figure 2.26 may break due to excessive preload if it is a well-oiled bolt, and if it is a rusty or dirty bolt, it may not perform its function due to insufficient preload [5].



Figure 2.26 Different preload values due to different coefficients of friction under same specified torque [39].

Despite all this, the torque-controlled tightening method is frequently preferred by design engineers because it is cheap and easy to apply. For connections other than critical bolted connections, it may be more feasible to overdesign considering the preload scatter.

2.2.6 Torque-Angle Tightening

The torque-angle tightening method takes advantage of the constant slope of the curve, called the elastic curve, in the tightening process and provides less variation than the torque-controlled tightening method by eliminating the friction dependence thanks to the thread advance geometrically related to the angle [6]. It has been shown in many different studies that the torque-angle control tightening method is one of the most reliable methods of tightening. The main reason for this is that this method is affected very low by many parameters that the torque-controlled tightening method is highly affected by [18].

Torque-angle tightening have two variables which are the snug torque and the tightening angle. First, torque is applied until it almost reaches the starting point of the elastic behaviour, which is called the snug point which is shown with Figure 2.27. Then it is tightened at an angle that will provide the desired preload value [6].

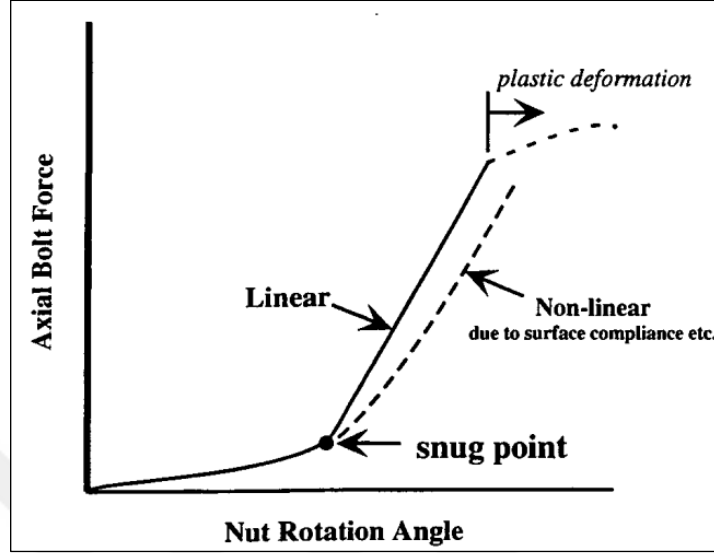


Figure 2.27 Nut rotation angle axial bolt force graph [6].

Before the torque-angle tightening method, there were angle tightening methods that were used by defining certain references before the angle tightening process, but these methods had much higher variation values than the torque-angle tightening method due to the low accuracy of the reference points. In the torque-angle tightening method, the snug point is used as the start of the angle tightening process. Snug torque is used to ensure that the alignment region is complete because the behaviour and width of this region can vary from bolt to bolt, assembly to assembly, application to application. It is not possible to make a theoretical prediction in this region. Also, if a initial torque value is not used, preload control will not be accurate due to different starting references when switching to the angle tightening process [5].

Joint-specific tests are performed to determine the snug torque value. Since the tightening angle value is determined within the elastic region, it can be predicted with Equation (2.8). The relationship between preload and angle is given by the following formula where α is turning angle, P is thread pitch, F is preload and k_B , k_{J1} , k_{Ji} are stiffnesses of the bolt, joint 1, joint i respectively [40].

$$F = k_R \Delta L \rightarrow F = k_R \frac{\alpha}{360} x P \rightarrow F = \frac{\frac{\alpha}{360} x P}{\frac{1}{k_B} + \frac{1}{k_{J1}} + \frac{1}{k_{Ji}}} \quad (2.8)$$

$$\rightarrow \frac{\Delta F}{\Delta \alpha} = \left(\frac{1}{k_B} + \frac{1}{k_{j1}} + \frac{1}{k_{ji}} \right) \frac{P}{360}$$

Equation (2.8) is written according to the approximation that fasteners compared to bolts can be considered rigid. In addition, the current formulation does not include twisting in the bolt shank. Crescimanno and Keller rearranged the above formula to include these two effects [40]. Fukuoka, on the other hand, rearranged the Equation (2.8) by adding the effects of surface hardness and inclination angle on the nut surface. He claimed that he improved the accuracy of the torque angle tightening method with the new formula he proposed [6], [41]. However, Bickford defined these and similar effects as negligible factors that are included in the variation caused by the torque-angle tightening method [5].

Although one of the main problems of the torque angle tightening method is that if for any reason preload cannot be created in the bolted joint, it is difficult to control it, today this concern has been resolved thanks to the high-tech tightening wrenches that also show the torque value after the angle input is provided. Even many anomalies can be detected if these values are followed [5].

2.2.7 Other Control Methods for Provide Accurate Preload

In the literature, there are many application methods used to provide high precision preload to bolted connections. Many of these are methods that are not preferred for industrial applications used in experimental studies, both in terms of cost and feasibility.

2.2.7.1 Yield control tightening

The tightening method in which the tightening is stopped when the yield point is reached by following the bolts to the yield point with various strategies is called the yield-controlled tightening method. There are studies claiming that this method has very low variation between 3% and 8%. In addition to being costly compared to the alternatives of this method, the most important disadvantage is that the bolt, which is tightened to the yield point, has difficulty in meeting the strength criterion under operational loads, even if the torsional stress on it is relieved when the assembly completes [5].

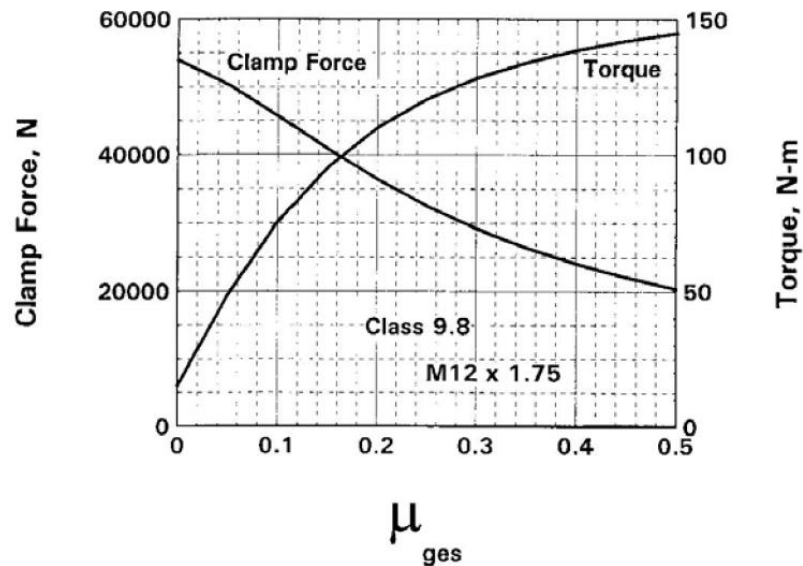


Figure 2.28 Friction effect on yield control tightening [4].

In Figure 2.28, the graph of the bolts tightened to the yield point with different friction coefficients for the M12 bolt is shared. As the coefficient of friction increases, the yield point is reached at higher torques, while lower preload values are achieved in the bolt despite higher torque levels. Control of friction is very important when using this method [4].

2.2.7.2 Stretch control

The stretch control methods are basically a method of estimating the preload value by measuring the elongation of the bolt. Despite the fact that effects such as part tolerances, plastic deformations and temperature changes reduce the accuracy of this method, it has much lower variation values compared to other application methods. Although there are many different application methods for stretch control, these methods are generally not suitable for mass production. However, it is preferred for some critical bolted joints because of its high accuracy values.

The most common stretch control method is measuring with a micrometer. The disadvantages of this method are generally the accessibility problems and the human factor. For stretch control, some companies produce bolts with self strain gauges, but these bolts are generally high-dimensional bolts. On the other hand, when it is evaluated in terms of both suitable material and cost, it is not frequently preferred in industrial applications.

Ultrasonic measuring devices, which have been used in the last 30 years, are devices that can predict stretch at high accuracy values, without being affected by many parameters that will cause variation in other methods [5]. Ultrasonic measuring

devices determine how much the bolt is stretched by measuring the return time of the sound wave given to the bolt to the device. The elongation or preload of the bolt is calculated according to the time change between the two measurements, taking into account the material velocity, stress and temperature factors specific to the material, with the measurement taken before and after the assembly [42].

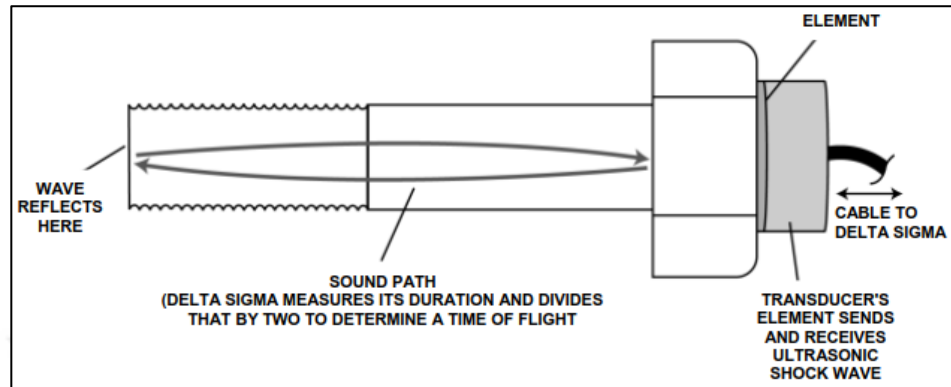


Figure 2.29 Ultrasonic measurement device [42].

2.2.7.3 Direct tension indicators

Direct tension indicator (DTI) is a method in which the preload is estimated by the measurements taken after the compression of the arch-like protrusions placed between the bolt and the nut, acting like small springs. One of the major disadvantages of this method is that the preload loss cannot be measured due to the plastic deformation of the protrusions [43]. The DTI can be placed under the bolt or nut head. However, in cases where it is placed under the nut as shown in Figure 2.30, it is recommended to use an extra conventional washer [5].

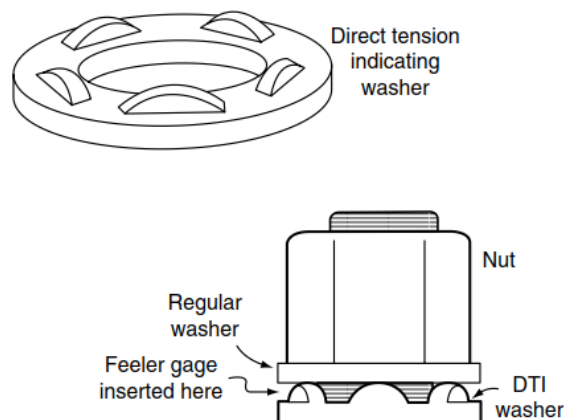


Figure 2.30 Direct tension indicator [5].

3. METHOD

The methodology followed in the study is given in Figure 3.1 as a flowchart. Accordingly, while creating the base design, test components with the same stiffness value as the reference design were created. Afterwards, 54 torque-tension tests were performed and tightening parameters were determined for both methods according to these test results. In addition, the preload-angle relationship in the test results was compare with the finite element analysis. Finally, two tightening methods were compared with the assembly tests performed for both methods.

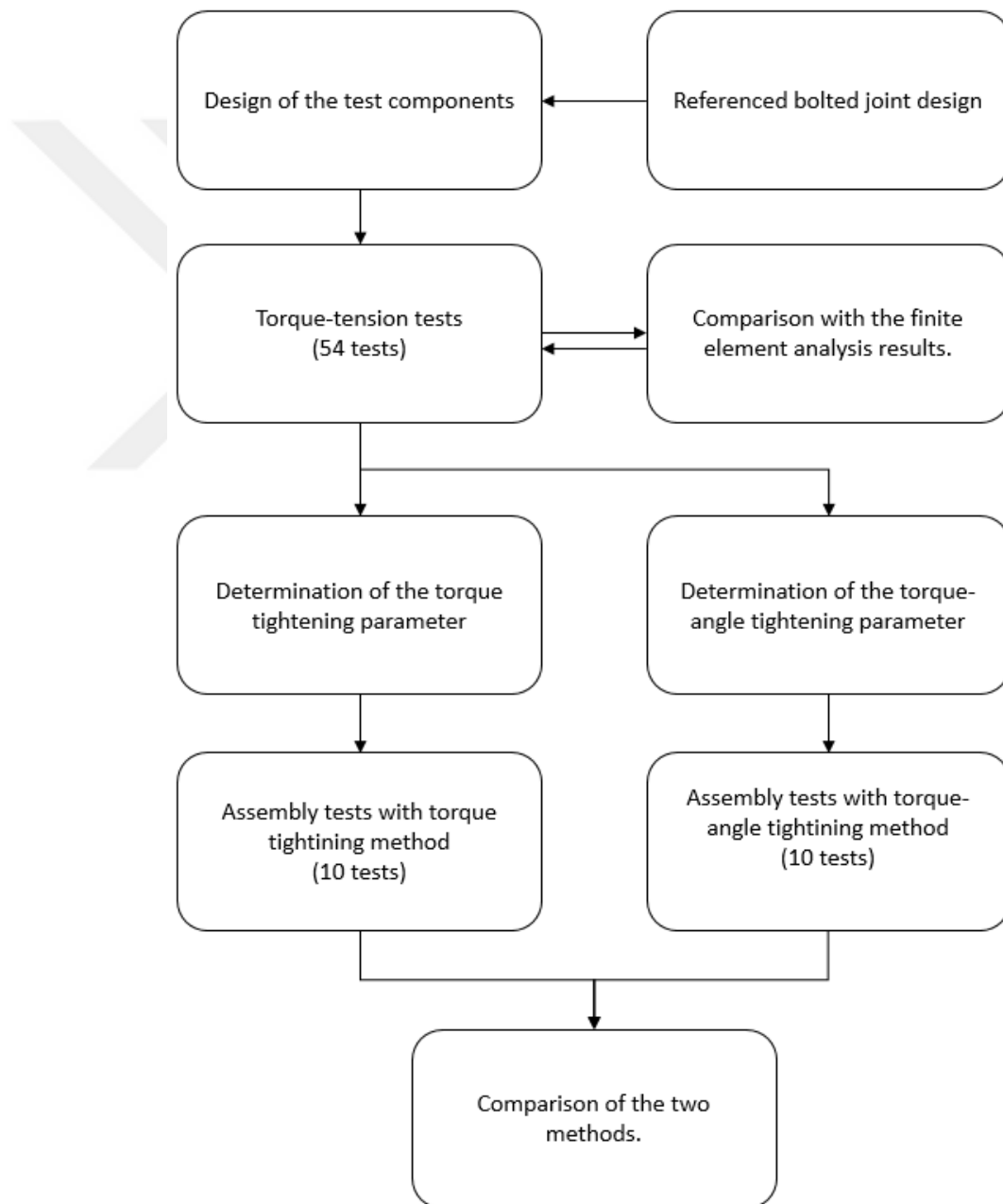


Figure 3.1 Flow chart of the study.

3.1 Referenced Bolted Joint Design

The T800-LHT-800 is a modern turboshaft engine developed for highly maneuverable helicopters. The engine is expected to give infinite life at 0.9 rad/s gyroscopic maneuvers, while it is expected to withstand 2.5 rad/s gyroscopic maneuvers for 30 seconds. T800-LHT-800 consists of followings: two stage centrifugal compressors, a reverse flow combustion, two stage high pressure turbine, two stage low pressure turbine. The engine's cross section is given with Figure 3.2.

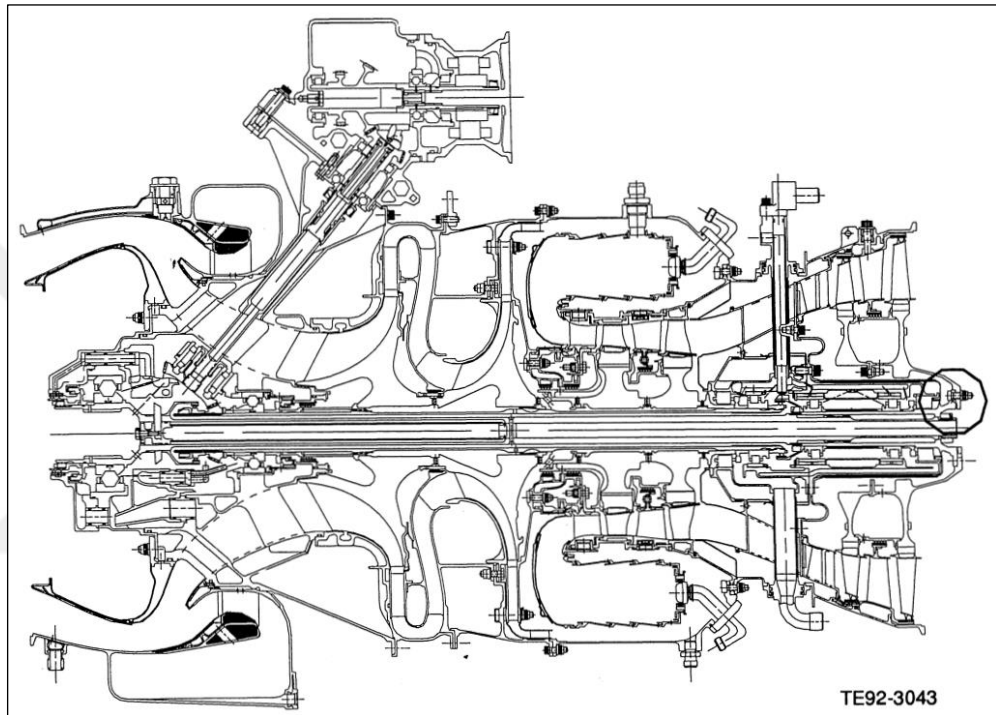


Figure 3.2 T800-LHT-800 PFR design cross section and the failure location [7].

A gyroscopic maneuver test with a preliminary flight rating (PFR) phase design was planned to demonstrate that the engine satisfies the system requirements against gyroscopic loads. In this test, it was found that the connection between the power turbine and the intermediate shaft caused the power turbine rotor to be released due to the fatigue damage. After the tests, some design changes were made. In addition to changes such as increasing the flange thickness, increasing the critical fillet size, the bolt length was also increased to increase the clamping of the power turbine flange to the stub shaft. Figure 3.3 shows the failure region and the proposed new design. The proposed new design change to increase the clamping of the critical bolted joint of the T800-LHT-800 engine is anticipated to reduce gyroscopic deflections [7].

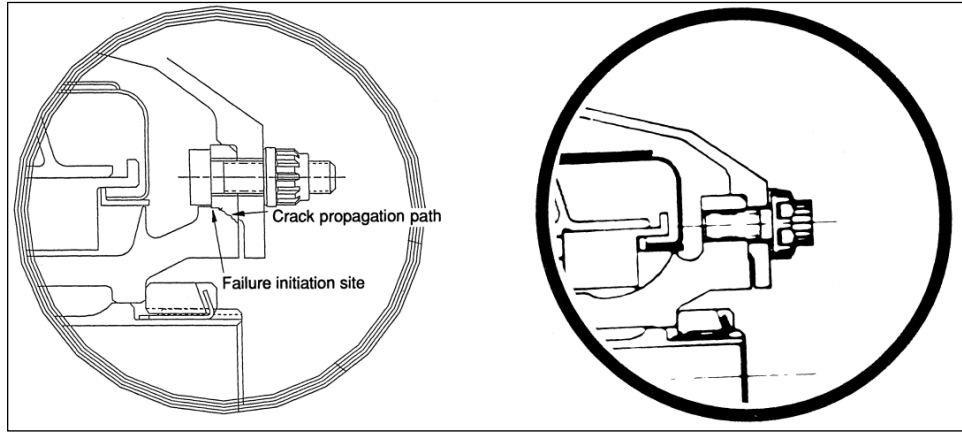


Figure 3.3 Failure region of the old design and the new proposed joint design [7].

Considering the engineering, production, manufacturing, and similar costs of the changes made in this region, T800-LHT-800 design is chosen for the study as it would be a powerful application to show that it is possible to increase the clamping by reducing the preload variation by simply changing the tightening method.

3.2 Design of the Test Components

The critical bolted joint of the T800-LHT-800 turboshaft engine is selected for the study to demonstrate the improvements of the torque-angle tightening method for critical bolted joint connections. Since the detailed dimensions of the design are not known, the dimensions of the design are created in accordance with 0.25 in bolt dimensions. The selected bolt is given in Figure 3.4.



Figure 3.4 0.25-28UNJF ISO3161.

The values shared in the gyroscopic test article are taken into account when creating the estimated base design [7]. Flange thicknesses are selected 9 mm, 5.5 mm and 0.6 mm for the stub shaft, disk flange of power turbine and the washer respectively. After the base design is created, test components which reflect the same configuration and equal stiffness are designed. Of the test parts shown in Figure 3.5, test bushing reflects the behaviour of the stub shaft while test plate reflects the behaviour of the disk flange.

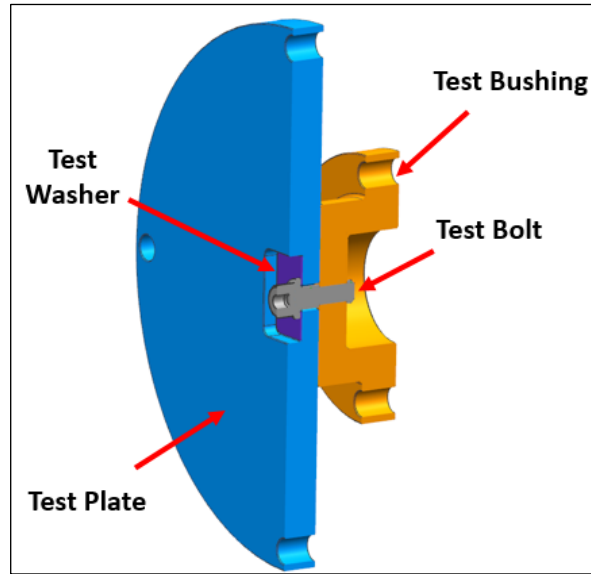


Figure 3.5 Designed components for torque-tension test system.

The interface of the torque-tension device used was taken as a reference in determining the basic dimensions of the design area outside the bolted joint region. The interface of the device used is schematically given in Figure 3.6. The figure is only for interfaces that affect the design space, it does not reflect the actual configuration and dimensions. Since the bolt used in the study is a cap screw without a nut, the bolt should be mounted in the opposite direction.

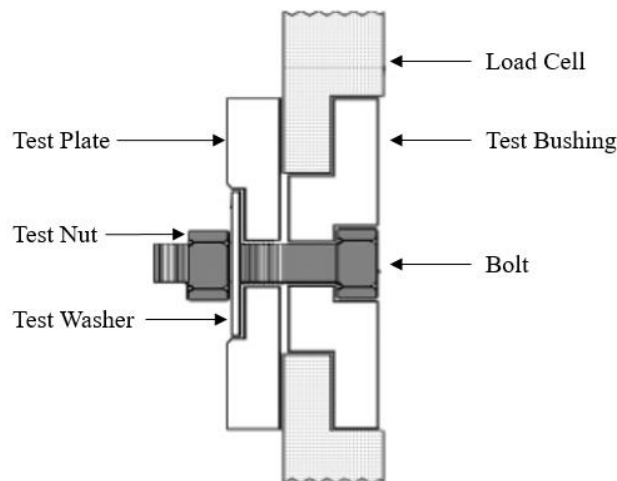


Figure 3.6 Torque tension device interface.

Becker and Frounfeldke stated that the material of the stub shaft is Inconel 718 in the study where the reference geometry is taken [7]. Although the materials of the other parts are uncertain, Inconel 718 is generally preferred in turbine discs of gas turbine engines due to its high yield and ultimate strength properties in high temperatures, creep resistance, LCF resistance, homogeneity and ductility. In addition, one of the most important reasons why Inconel 718 is frequently preferred in hot regions of gas

turbine engines is that it is cost effective compare to the alternative materials [44]. Considering all these, it is assumed that the material of the other parts are Inconel 718, such as the stub shaft, whose real material is specified. All components are manufactured from forged raw material and there is no coating or plating on any part.

3.3 Torque-Tension Testing

It is aimed to examine the characteristic of the bolted joint by performing torque-tension tests for the test configuration created. In these tests, the preload variation is determined for the torque control tightening method by examining the torque preload relationships directly. In addition, the same data is examined, and K factor calculation is carried out for the bolted joint. On the other hand, the torque-angle and preload-angle relationships are examined in order to determine the torque-angle control tightening parameters that will provide the determined preload value. Snug torque is determined by examining the torque-angle graphs, and the tightening angle is determined by examining the torque-preload relationship.

3.3.1 Test Equipment and Experimental Procedures

Tests are performed on a calibrated Labmaster 3200 test system. The test system given in Figure 4.6 consists of load cell to measure preload and thread torque, torque angle transducer to measure applied torque and bolt rotation, DC electric drive motor, servo controller. Additionally, Windows-based software is used for transducer signal conditioning, data acquisition and drive motor control [45].

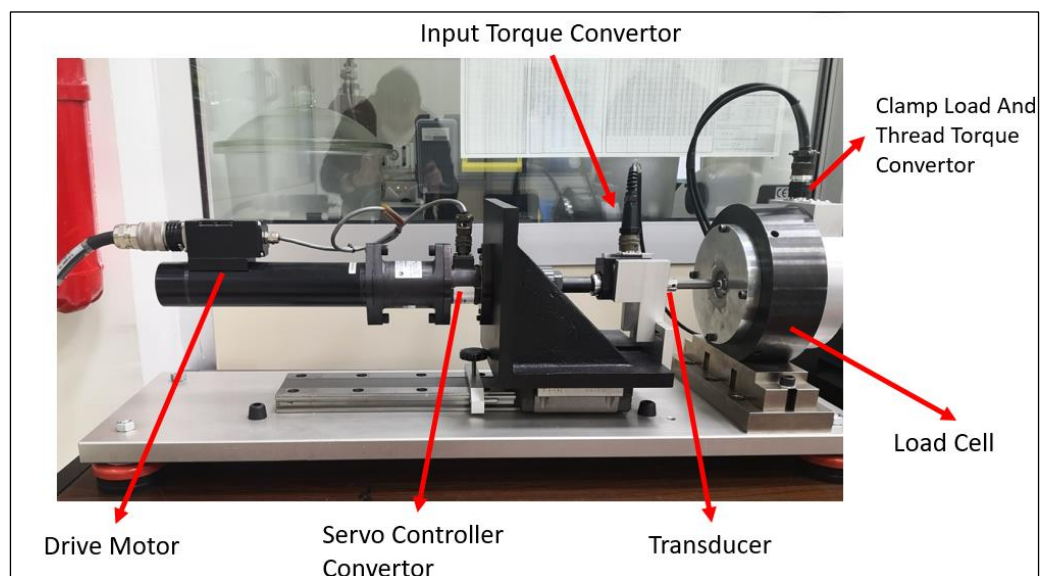


Figure 3.7 Labmaster 3200 torque tension test system.

Before starting the test, necessary information such as tightening speed, simple bolt dimensions, maximum input torque are entered into the Windows-based software. Afterwards, the bolt is turned by hand so that it enters its slot. When the test starts, as can be followed from the example given in Figure 3.8, the device rotates the bolt up to the specified torque value, when the relevant torque value is reached, the device starts to rotate in the opposite direction. When the disassembly torque value is reached, the preload on the bolted joint starts to decrease and stops when the device reaches its initial position. During the tightening process the device measures friction coefficients, preload, torque, and angle values.

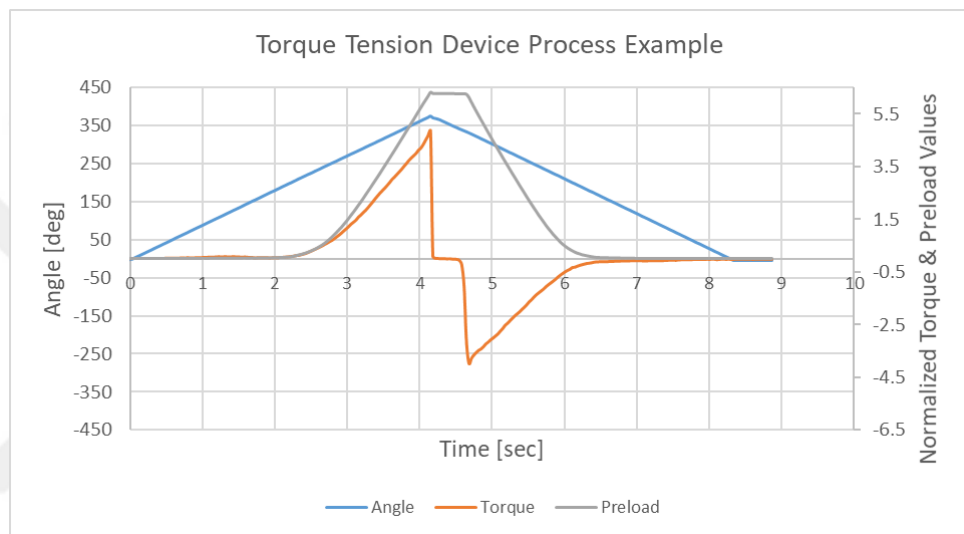


Figure 3.8 Torque-tension device – a process example.

The device also provides sufficient data to confirm that the bolted joint remains in the elastic region. The yield point can be detected easily thanks to the post process tool in the user interface of the device. For tests that do not exceed the yield point, the data obtained during unloading is sufficient to evaluate whether there is plastic behaviour.



Figure 3.9 Example of a bolt failure during the test.

Output data and graphs can also be tracked during testing. Even if the monitored data differs significantly from the average data due to an abnormal behaviour, the tests

are not stopped without being sure. Naturally, during the tests, various failures were experienced, as an example is given in Figure 3.9. Even if there is no visible damage on the bushing part, after each bolt failure, the bushing part is replaced with the new one.

3.3.1.1 Normalization of the test data

All torque values in the study are normalized with the torque value of the snug point obtained with the test results. The preload values are also normalized with the average preload value corresponding to the snug point.

3.3.2 Experimental Conditions

Tests are carried out in accordance with NASM-1312-15 [46] and ISO16047 [47] standards at room temperature. Bolts are fully lubricated with engine oil and all bolts tightened to the same torque value with a constant speed.

A total of 54 tests are performed with 18 samples of 0.25-28 UNJF ISO 3161 bolts, using each bolt 3 times with reference the NASA-STD-5020 [48]. Angle dependent preload and torque values are measured. Input torque versus preload values are also measured. For preload variation calculations two-sided three sigma distribution is used with unbiased sample standard deviation. Unbiased sample standard deviation, σ , is given with Equation (3.1) where m is the sample size, F_j is the preload value obtain in the j -th test and F_{nom} is the nominal preload value [48].

$$\sigma = \sqrt{\frac{1}{m-1} \sum_{j=1}^m (F_j - F_{nom})^2} \quad (3.1)$$

3.3.3 Torque-Tension Test Results

The torque tension graph is created with the data obtained from the tests is given in Figure 3.10. As can be seen in graph, all test data remain in the range of 3 sigma limits. The preload variation for the maximum torque value is obtained as 19 percent. Numerical values calculated for eleven data point are shared with Table 3.1. It is observed that as the torque increased, the preload variation decreased in percentage.

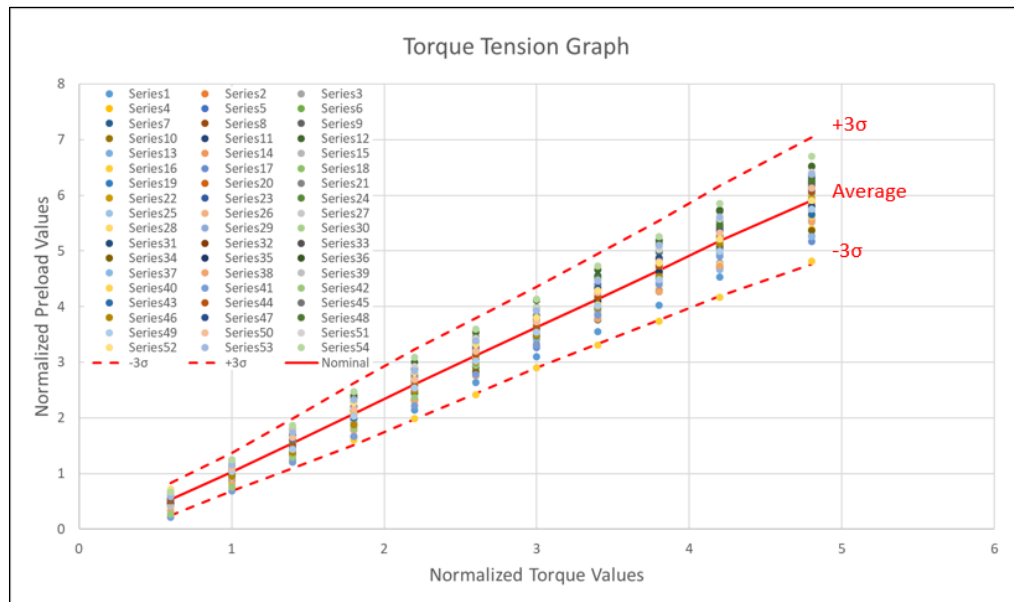


Figure 3.10 Torque-tension test results.

Table 3.1 Torque tension results for selected points.

Torque	Preload	Standard Deviation	Preload -3σ	Preload +3σ	Variation Percentage
0.6	0.535	0.097	0.243	0.827	55%
1	1	0.113	0.684	1.365	33%
1.4	1.541	0.148	1.096	1.985	29%
1.8	2.070	0.184	1.518	2.623	27%
2.2	2.602	0.209	1.975	3.230	24%
2.6	3.115	0.224	2.442	3.789	22%
3	3.628	0.241	2.904	4.353	20%
3.4	4.138	0.271	3.325	4.951	20%
3.8	4.651	0.297	3.759	5.543	19%
4.2	5.174	0.330	4.183	6.166	19%
4.8	5.903	0.379	4.764	7.041	19%

Since the torque and preload values are normalized, the K factor value to be calculated will not coincide with the actual one. However, if the torque/preload ratio behaviour during loading is examined, it will parallel the behaviour of the K factor. As the torque value increases, the torque to preload ratio decreases and converges to a final value, as shown in Figure 3.11.

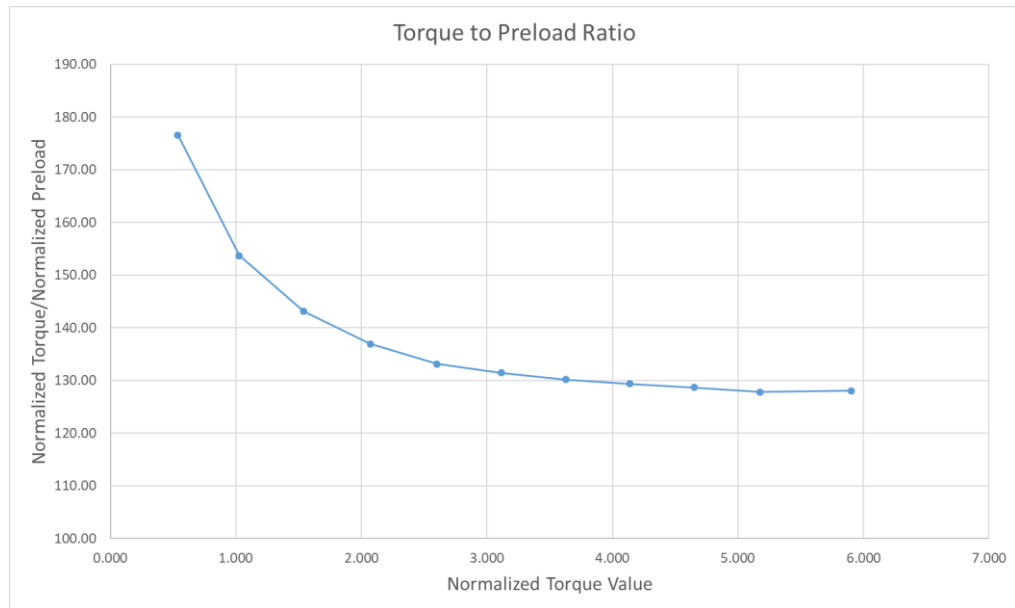


Figure 3.11 Torque to preload ratio.

3.3.4 Determining the Snug Torque

As a result of the experiments, the relationship between normalized torque and angle values are given in Figure 3.12 and the relationship between preload and angle is given in Figure 3.13. The main reason why the rundown region is observed in different widths in the test results is due to the difference in the initial positions of the bolts. Bolts are manually inserted into the test system ensuring that no torque is generated. Although there is a significant amount of scattering among the experimental results due to the uncontrolled rundown region, the fit can be greatly improved by shifting the reference points on the angle measurements.

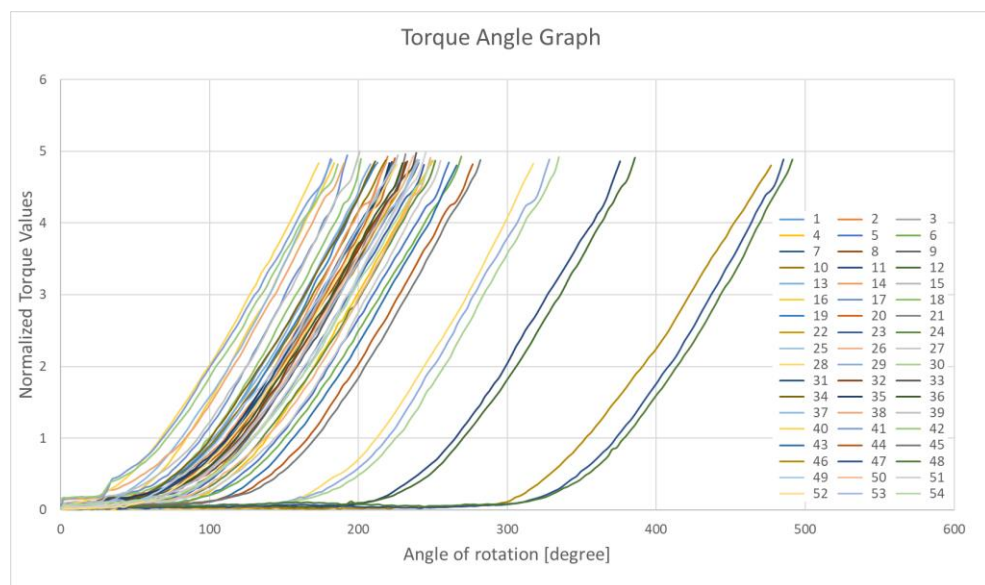


Figure 3.12 Test results – torque-angle relationship.

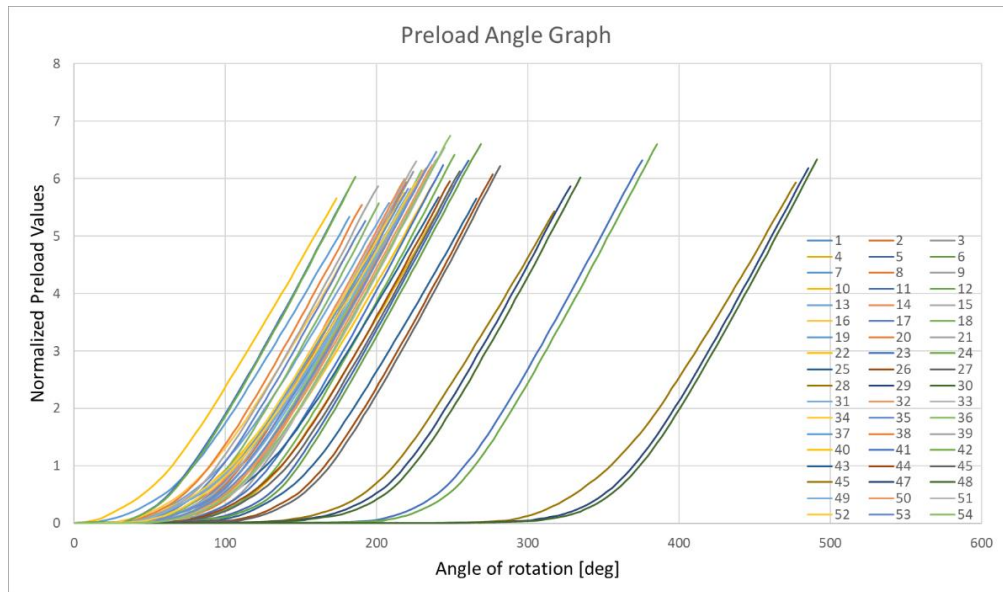


Figure 3.13 Test results – preload-angle relationship.

The snug point can be estimated by examining the torque-angle and preload-angle graphs. If the angle values corresponding to the snug point are used as a reference for shifting, the quality of the snug point estimation can be examined from the resulting shifted graph. For torque-angle plots, the graphs are shifted according to the predicted snug torque value, while the preload-angle graphs are shifted with reference to the preload value corresponding to the snug torque value. Thus, after a few iterations, the value of snug torque is determined as 1. Since snug point is the starting point of the elastic behaviour, the angle value corresponding to this point is set to zero. Shifted torque-angle and preload-angle graphs with reference to the snug torque value are given with Figure 3.14 and Figure 3.15.

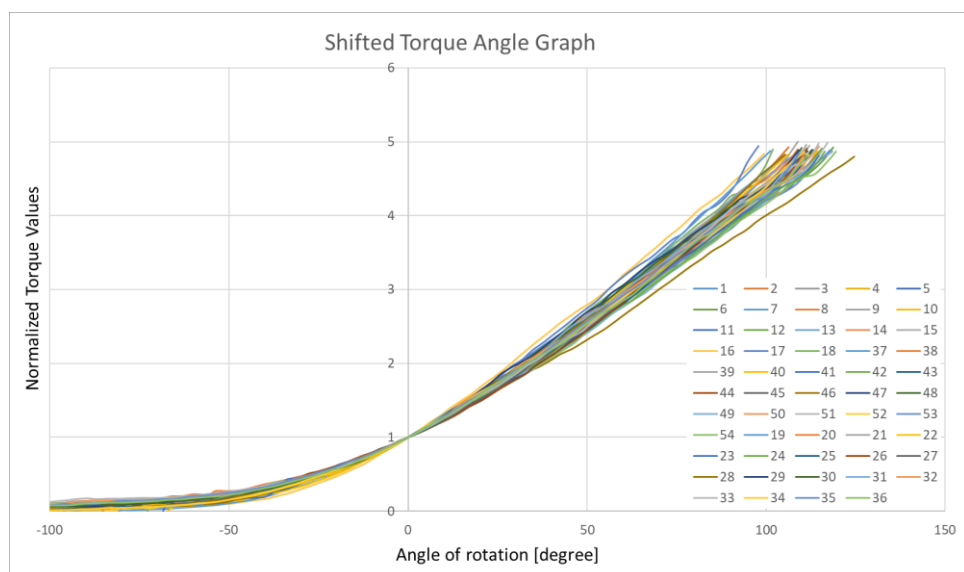


Figure 3.14 Shifted torque-angle graph.

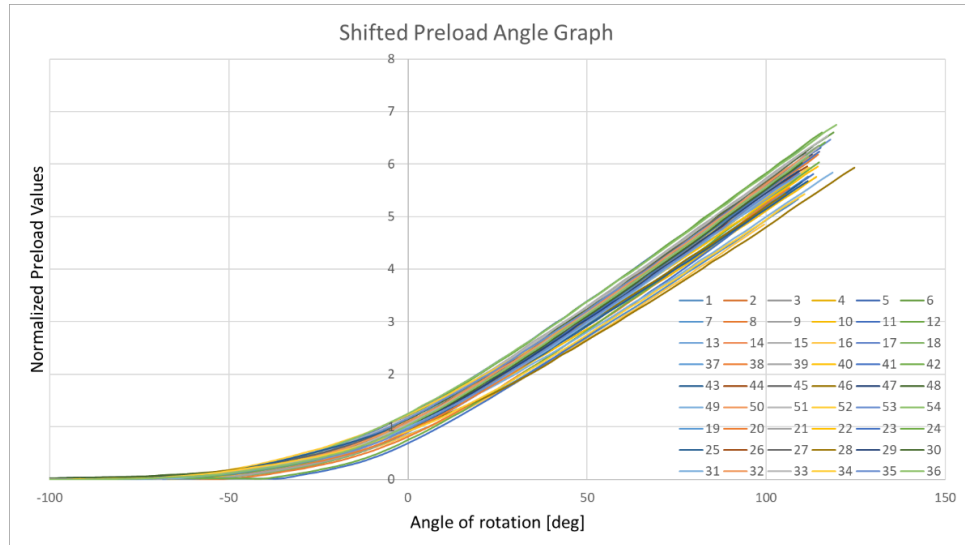


Figure 3.15 Shifted preload-angle graph.

3.3.5 Determining the Tightening Angle

The tightening angle value is determined by considering the preload-angle relationship in the elastic region by accepting the snug point as the origin. Since there is a scatter between the shifted preload-angle relationship curves given in Figure 3.15, an average curve is extracted from these curves as given in Figure 3.16 to decide the value of the tightening angle.

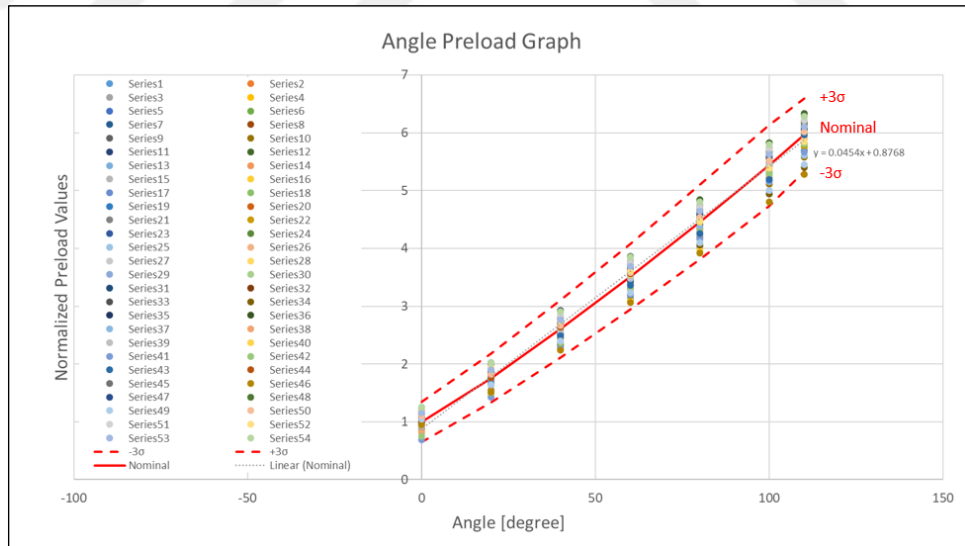


Figure 3.16 Nominal angle-preload graph.

For the bolted joint design, the tightening angle is determined considering that the -3σ value corresponding to the selected tightening angle must provide the minimum preload value required for the design. At the same time, the highest preload value, $+3\sigma$, of the selected tightening angle must meet the strength criteria of the design.

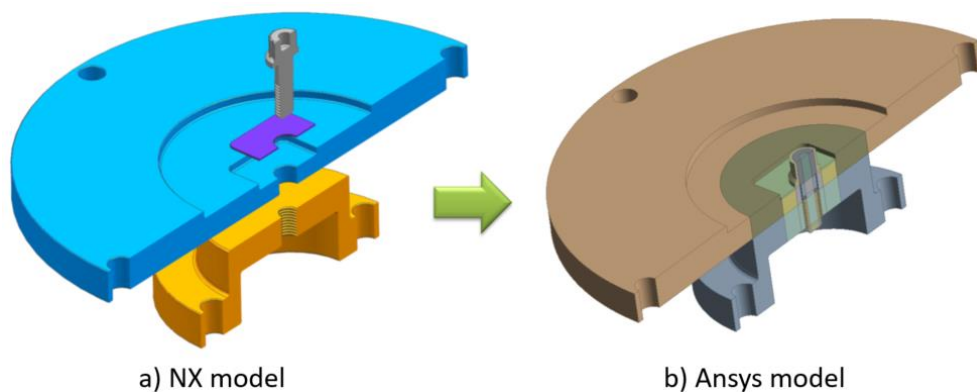
Since the required preload value of the T800-LHT-800 engine is not known, the average preload value of 5.9, which is obtained against the 4.8 maximum torque value, is accepted as the target. The tightening angle value required to give a preload of 5.9 is determined as 110° as a result of the torque-tension tests.

3.3.6 Comparison of Experimental and Numerical Results

In order to determine the relationship seen in torque-tension tests by a numerical study, the test configuration is analysed with ANSYS 2021R2, one of the most preferred finite element method program. Since the torque-tension tests are performed in the elastic region, the analysis model is built with elastic materials, but the analysis cannot be classified as linear because it includes large deflection effects. It should also be noted that although elastic analysis is taken as the main analysis, a comparison with elastoplastic analysis is also made to determine the rightness of the claim.

3.3.6.1 Geometry preparations

The nominal geometries created in Siemens NX 12, a Computer Aided Design program, are imported to Ansys2021R2 as Parasolid format. Afterwards, geometric modifications are made in Ansys Designmodeler and the model is made ready for the analysis. As given in Figure 3.17, the chamfers and fillets that are predicted to have no effect on the analysis results have been removed. In addition, it is aimed to increase the performance of the contacts with the cut geometries considering the relevant contact surfaces. Moreover, the thread areas of the bolt and bushing are cut, allowing local mesh improvements to be achieved.



a) NX model

b) Ansys model

Figure 3.17 Geometric modifications of the test model.

3.3.6.2 Meshing

Thanks to the cut mutual geometries, it has been tried to obtain node to node mesh elements in the contacts as much as possible. The model is meshed with

quadratic elements consist of Ansys' Solid186 (20-Node Structural Solid) and Solid187 (10-Node Tetrahedral Structural Solid) elements [49]. As can be seen in the mesh views given in Figure 3.18, fine mesh is only used on the contact surfaces and thread area in order not to increase the model cost.

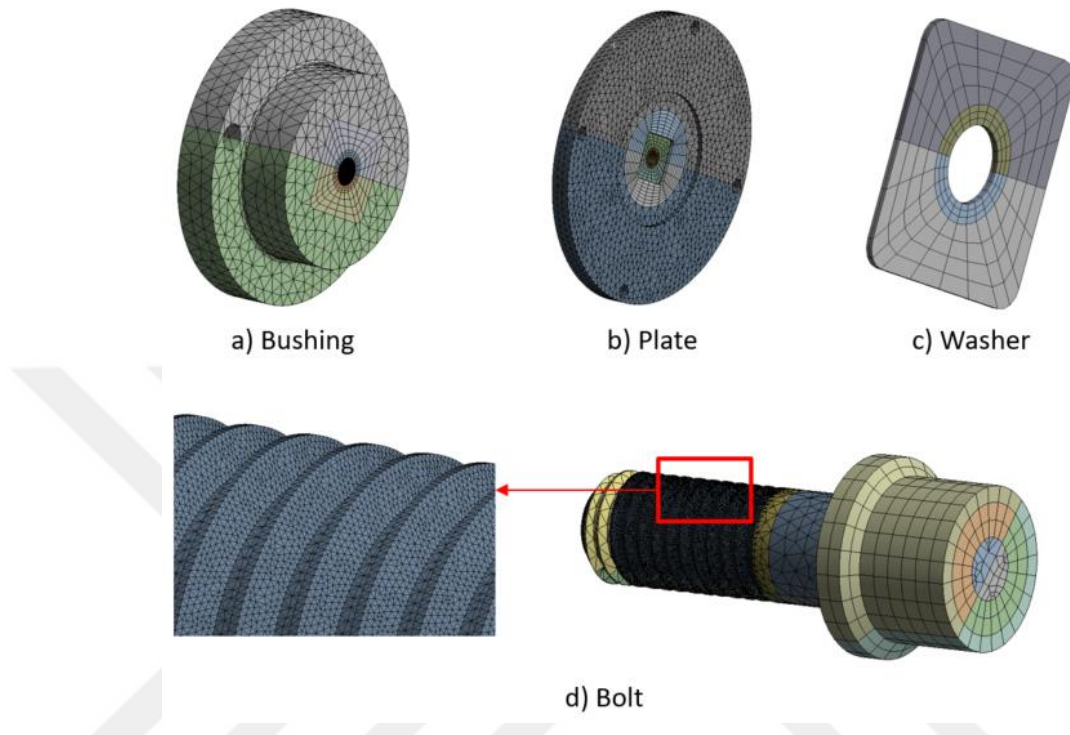


Figure 3.18 Meshed views of the test components.

3.3.6.3 Loads and boundary conditions

Since the angle input versus the preload output will be examined in the analysis results, the tightening angle value is given as the angle of rotation from the bolt head. The elastic origin was calculated from the equation in the graph given in Figure 3.16. The bolt is rotated a total of 130° in the analysis, 20° to reach the snug point and 110° to achieve the targeted preload. As an application method, a remote point defined to the center of the bolt head which is connected to the bolt head surface with RBE3 elements. At the same time, the movement of bolt the head is restricted in rot-y and rot-z degrees of freedom in order to ensure that the bolt moves linearly by rotating only around the X axis with the angle input. In addition, in order to prevent the washer from turning, frictionless support has been defined on its side surfaces as shown in Figure 3.19. As an alternative to prevent the rotation of the washer, the nodes selected from the two opposite corners of the washer but did not have a remarkable effect on the results when the Y degrees of freedom from one and the Z degrees of freedom from the other were restricted.

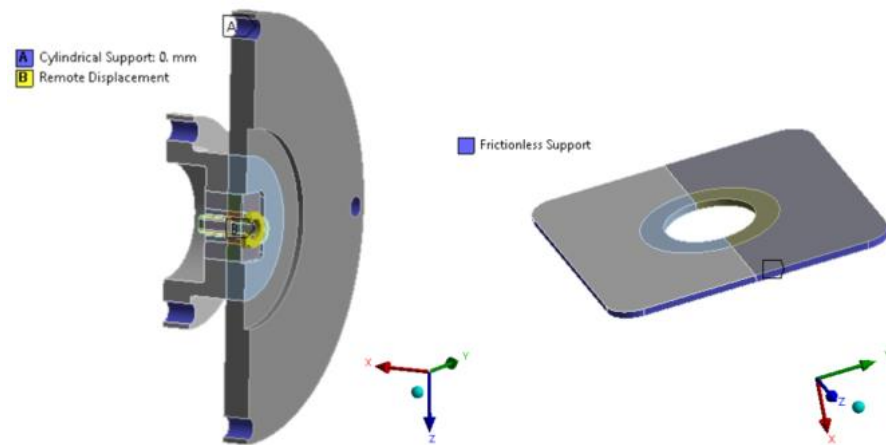


Figure 3.19 Boundary conditions of the finite element model.

The inner surfaces of the bolt holes, which interface with the torque-tension device, are fixed with the help of the cylindrical supports to immobilize the model. As an alternative to this boundary condition, fixing the front and back surfaces of the model did not make a significant difference.

3.3.6.4 Contact modelling

Contact modelling is the part that has the most impact on the analysis results. All contacts are defined as frictional contacts and Conta174 (8-node surface-to-surface contact) is used for contact elements and Targe170 (8-node contact 3-D target segment) is used for target elements [49]. In the iterations using different friction coefficients, it is observed that the friction coefficient did not affect the results. However, the data obtained from the test results are used for the friction coefficient in the final analysis. Nodal-projected normal from contact is preferred as detection method for better contact performance [50].

Due to the complex geometry, especially for the contact between the threads, a correctly selected interface treatment technique is needed. "Offset only, ignore initial status, no ramping" feature that comes with Ansys2021 is preferred for the contact, the improvement of this technique compared to other techniques is shown in Figure 3.20. The results in the graph are the initial contact results for three different interface treatment technique, in which the contact quality is evaluated before the analysis.

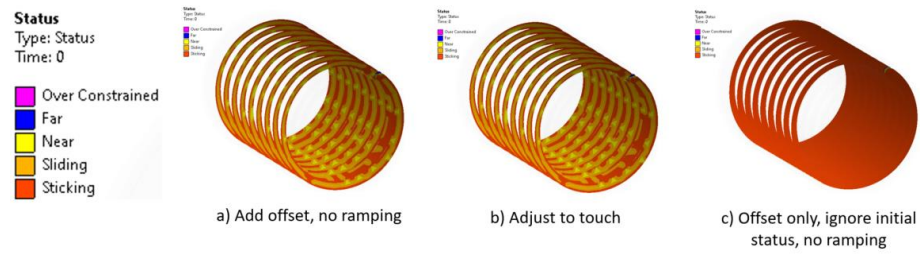


Figure 3.20 Initial contact status of different interface treatment techniques.

Pure penalty method is chosen for contact formulations and after iteration studies with different normal stiffness factors, 0.1 is chosen for the contact between the threads and 0.035 is chosen for the contact between the bolt head and the bearing surface. In addition, since the normal stiffness factor does not have a certain value and considering the large rotation of the contacts, the update stiffness feature is activated and the "each iteration, aggressive" option is selected. Furthermore, finite sliding formulation is used for the contacts since the analyses are carried out with large deformation effects.

3.3.6.5 Finite element model results and comparison

Although there is a significant difference in solution time between the elastoplastic analysis created with multilinear isotropic hardening material data and the elastic analysis, there is no remarkable difference in terms of analysis results. The main reason for this situation is that the analysis results remain in the elastic region, as claimed. As given in Figure 3.21, contact pressures of elastic and elastoplastic analyses give close results.

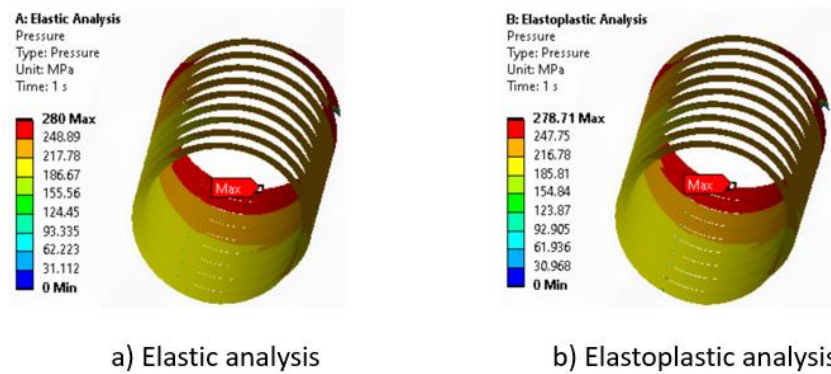


Figure 3.21 Comparison of contact pressure results.

While stress results are examined in elastic analysis results, plastic strain results should be examined in elastoplastic analyses. Equivalent stress results normalized with yield strength are given in Figure 3.22 for the elastic analysis. It can be seen in the figure that the regions exceeding the yield strength are the same singular regions for both analyses. Even if the yield strength is exceeded, when the plastic strain results are

examined, it is seen that plastic strain values are so small that they can be neglected. In addition, the main reason for the stress distribution to appear at the minor diameter as well as in the thread root is that, unlike the usual bolted joint analyses, in this analysis, besides the preload, there is also additional stresses due to the torque.

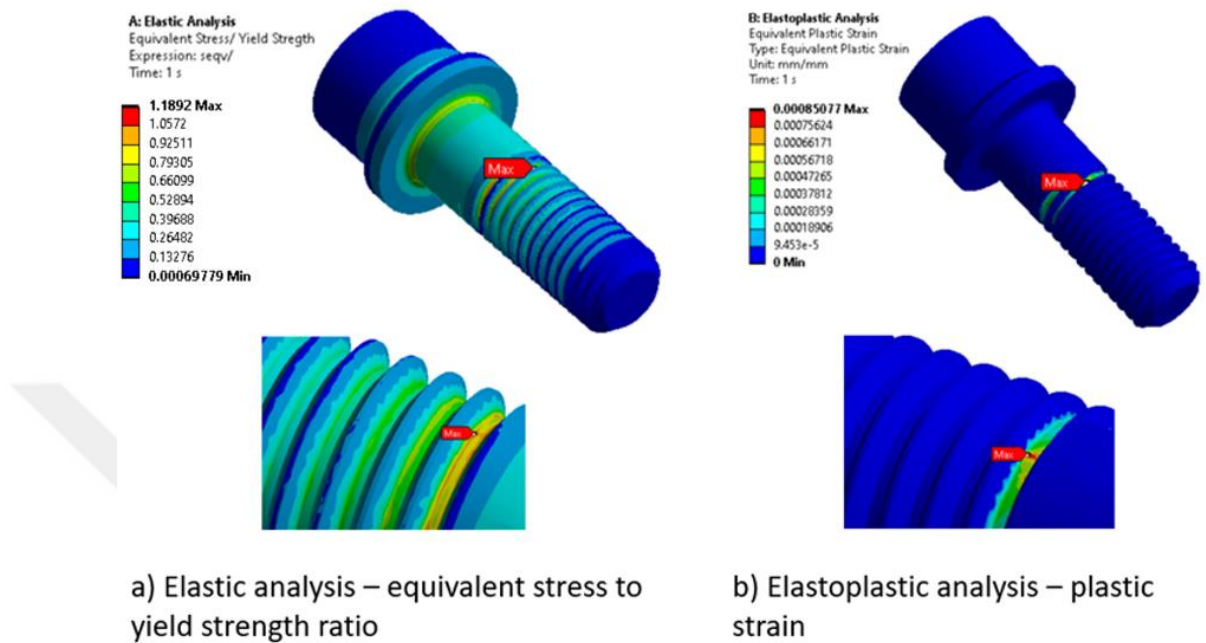


Figure 3.22 Finite element analyses results.

The main reason for the analyses performed is to determine whether results compatible with the torque-tension tests can be achieved. As seen in Figure 3.23 compatible results are obtained between the analysis results and the torque-tension tests for the torque-angle relationship.

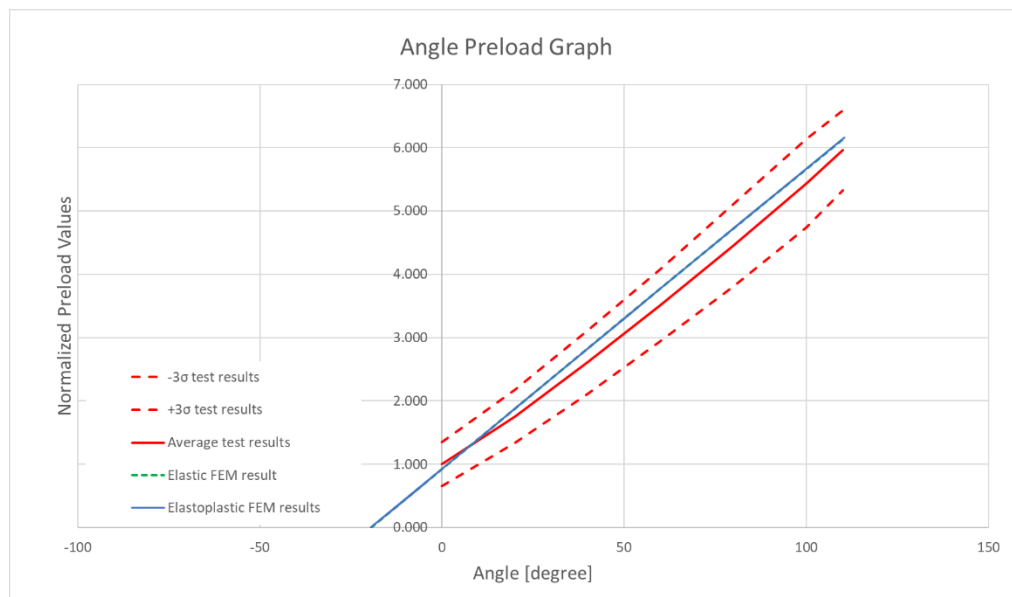


Figure 3.23 Tests and FEM results comparison graph.

3.4 Torque and Torque-Angle Tightening Methods Comparison

In order to examine the preload variations of each method in the actual assembly condition, new tests are conducted using a tightening wrench instead of a torque-tension device. Simulating the actual assembly condition using a tightening wrench includes the variations such as wrench accuracy, different tightening speeds, and operator skills. Snap-on's Techangle model a high-precision wrench that can provide angle or torque input, is used for the tests. As can be seen in Figure 3.24, the wrench turns on a green light when it reaches the determined value and interrupts the loading digitally. This control system drastically reduces variation from operator skill.



Figure 3.24 Snap-on TechAngle tightening wrench.

10 tests are carried out for each methods with the determined tightening values for the torque-controlled tightening method and the torque-angle tightening method. 4.8 torque value is used for torque control tightening. 1 torque value and 110° tightening angle are used for torque-angle tightening method, respectively. The results of the tests performed with the wrench are given in Figure 3.25.

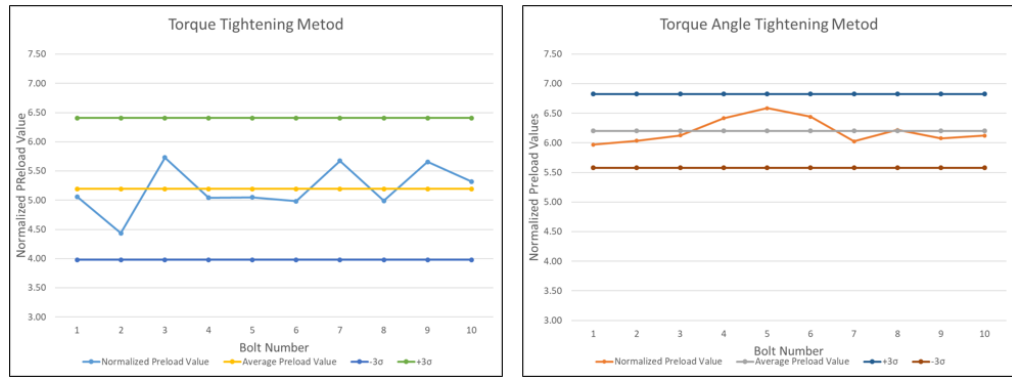


Figure 3.25 Preload variation for each method with wrench tests.

Since the scales are the same, the superiority of the torque-angle method can be easily determined in the graphs where the results of the two tightening methods are given in order of 10 tests. With the torque tightening method, an average preload value of 5.19 and 23% preload variation are obtained, while 10% preload variation is observed with an average of 6.20 with the torque angle tightening method. In assembly tests performed with inputs specified for both methods in order to obtain the same preload value according to the torque-tension test results. Torque-angle tightening method gave 19% higher average preload value and 40% higher minimum preload value compared to torque tightening method.

4. CONCLUSION

Gas turbine engines are modern machines whose components are exposed to high pressure and high temperatures and which require very precision work in terms of both design and production compared to other industrial machines due to fuel efficiency, availability, and safety requirements. Bolted joint connections are commonly preferred in this modern machine design due to its low cost, replaceability, and reliability. The most important parameter in the design of bolted joints is the preload value, if it is too much, it can cause loss of integrity and if it is insufficient, it can cause leakage or separation. For critical bolted joints, that could cause a catastrophic event if they fail, the accuracy of the preload value is crucial. Although there are many methods to provide preload in the literature, the most frequently used method in the industry is the torque-controlled tightening method due to its ease of application and low cost.

In this study, the torque-angle tightening method, which causes less preload variation compared to torque-controlled tightening, is investigated and a detailed methodology is presented on how to determine the snug torque and tightening angle parameters of this method.

Test components are created with reference to the critical bolted joint of the T800-LHT-800 engine. With the torque-tension tests performed, tightening parameters targeting the same preload value are determined for both tightening methods. As a result of the assembly tests performed with a tightening wrench using the specified tightening parameters, it has been demonstrated that the torque controlled tightening method causes %23 preload variation, while the torque angle tightening method causes %10 preload variation.

On the other hand, using the torque angle tightening method instead of the torque tightening method for the bolted joint examined increased the minimum preload value by 40% and the average preload value by 19%. Even if the torque angle tightening method is not preferred during the design phase, when considering that the only difference between the two methods is the input of the tightening wrench, costly design changes can be avoided by simply changing the tightening method after any field problem due to insufficient preload.

Furthermore, the FEM model is established to represent the torque-tension tests performed. In the analyses performed by turning the bolt with an angle, the results are

consistent with the tests. In addition to examining the effect of different boundary conditions for bushing, plate and washer components in the analyses, an elastoplastic analysis model is established and the effect of the material data is also examined. It has been determined that the greatest effect on the results comes from the contact behaviour and the contact behaviour is optimized by using advanced contact settings.

Although the torque angle tightening method is much better in terms of cost and ease of application compared to alternative tightening methods, the need for joint-specific tests makes this method more costly than the torque tightening method. As a future study, if a statistical method can be developed for snug torque with tests to be carried out with different configurations and bolts of different size, the tightening values for the torque angle tightening method can be estimated before the joint-specific tests. Even the need for FEM analysis can be eliminated with a new approach if the equations developed in the literature to estimate the stiffness of the bolted joint are combined with the statistical snug torque estimation.

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BIOGRAPHY

Yasin Okan Kösterit started his career in the automotive industry as a Calibration and Control Engineer. He is currently working as a Lead Structural Analysis Engineer in the aviation industry.

