A Study on Comparison of Bolt Tensioning Applied by Conventional Torque Method and Hydraulic Tension Method

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Bilal Mohammed QASIM

ABSTRACT

A STUDY ON COMPARISON OF BOLT TENSIONING APPLIED BY CONVENTIONAL TORQUE METHOD AND HYDRAULIC TENSION METHOD

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Bolted assemblies are the most commonly used joints in mechanics and bolted joint requires preloading to perform its function successfully. Preloading is usually generated immediately after tightening the bolt by conventional torque wrench method. Other than torque wrench method, hydraulic tensioning a new method of bolt tightening has been used in industry recently. In this study, an experimental investigation on two types of tightening methods (conventional torque wrench and hydraulic tensioning) is performed. Bolt's behavior during tightening, and advantages-disadvantages of both tightening methods are investigated. The study comprises two tightening varieties of experiment; static bolt behavior and dynamic bolt behavior in a bolted joint.

A total of 24 bolts, consisting of 8.8 grades and three different diameters (M14, M16 and M18) with almost the same (150 mm) lengths, were tested by employing both tightening methods. Static tests/investigations were performed in laboratory site of Mechanical Engineering of Gaziantep University of Turkey while dynamic tests/investigations were performed by applying cyclic load in laboratories of Karabuk University of Turkey.

Keywords: Bolts, Preload, torque tightening method, hydraulic tightening method, cyclic load

ÖZ

Cıvatalı bağlantılarda ön gerilmenin konvensiyonel tork metodu ve hidrolik ön gerdirme metodu ile uygulanmasının karşılaştırılması

Bilal Mohammed QASIM
Yüksek Lisans Tezi, Makina Mühendisliği Bölümü
TezYöneticisi:Prof. Dr. Nihat YILDIRIM

Haziran 2014, 96sayfa

Cıvatalı bağlantılar mühendislik tasarım, imalat ve montajlarının en çok kullanılan birleştirme yöntemlerinden birisidir. Cıvatalı bağlantıların fonksiyonlarını başarılı şekilde yerine getirmesi gerekli ön yüklemenin uygulanmasına bağlıdır. Ön yükleme genellikle cıvatanın tork anahtarı ile gerekli olan tork değerine kadar sıkılması ile gerçekleştirilir. Tork anahtarından başka sanayide son yıllarda çokça kullanılmaya başlanan hidrolik gerdirme diğer yeni bir ön yükleme metodudur. Bu çalışmada iki farklı ön yükleme metodları olan tork anahtarı ve hidrolk gerdirme yöntemleri üzerinde deneysel çalışmalar yürütülmüştür. Her iki metod ile ön yükleme yapılan cıvataların ön yükleme sırasındaki ve sonrasındaki davranışları, avantaj ve dezavantajları incelenmiştir. Yürütülen çalışma cıvatalı bağlantıların statik ve dinamik davranışlarını kapsamaktadır. Her iki ön yükleme metodu kullanılarak 8.8 kalite değerine sahip yaklaşık 150mm boyundaki üç ayrı cıvata ölçüsündeki (M14, M16 ve M18) 24 adet cıvata ile ön yükleme testleri gerçekleştirilmiştir. Statik testler Gaziantep Üniversitesi Makina mühendisliği bölümü laboratuarında dinamik testler ise Karabük Üniversitesi, Demir Çelik Enstitüsü laboratuarında gerçekleştirilmiştir.

Anahtar kelimeler: Cıvata, önyükleme, tork anahtarı yöntemi, hidrolik gerdirme yöntemi,

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LIST OF SYMBOLS

Fi	Bolt tension(preload)
Fp	Clamp load in the clamped pieces
MA	Applied torque or input torque.
MG	Thread torque
At	Tensile stress area of the bolt
Sp	Proof strength of the bolt
Sy	Yield strength of the bolt
T	Torque
K	Nut factor
d	Nominal diameter
P	Thread pitch
μ_t	Coefficient of friction between male and female threads
μ_b	Coefficient of friction between the bearing surfaces under the turning fastener and member surface
r_t	Effective contact radius between threads
r_b	Effective bearing radius of the bearing contact area under the turning head or nut
$\mathrm{T_p}$	Pitch torque
$\mathrm{T_t}$	Torque component that overcomes the friction between male and female threads.
$\mathrm{T_b}$	Torque overcomes friction between the turning fastener head or nut and the clamped joint surface.
σ	Equivalent total stress
σt	Tensile stress
τ	Torsion stress
Ph	Hydraulic pressure

ε	Strain
$\Delta\ R$	Resistance change due to strain
R	Gauge resistance
FLA	Strain gauge series used for axial strain
FCT	Series used for torsion strain
P total	Total external tensile load applied to the joint
Pb	Portion of external tensile load taken by bolt
Pm	Portion of tensile load taken by members
Fb	Resultant bolt load
Fm	Resultant load on members
C	Fraction of external load P carried by bolt
Kb	Bolt stiffnesss
Km	Member stiffness
γ	Shear strain
G	Modulus of shear strain
J	Polar moment of inertia

CHAPTER 1

INTRODUCTION

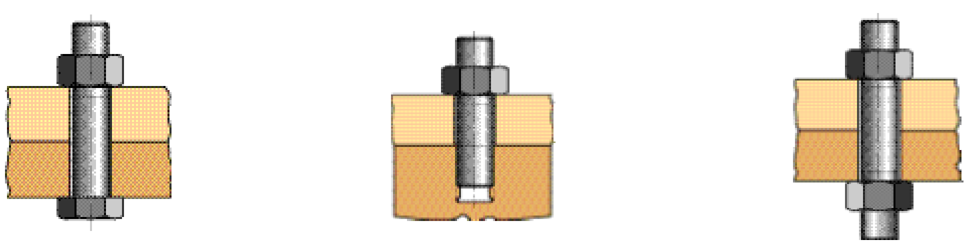
1.1 General

Threaded connection is a popular joining method in modern industries making up nearly 70% of all mechanical connections assemblies in industries worldwide. A screw thread can be defined as a ridge of uniform section in the form of a helix on either the external or internal surface of a cylinder. Internal threads refer to nuts and tapped holes, while external threads refer to bolts, studs, or screws. The effect preload in threaded connections is a very important factor affecting the joint performance and reliability. Improper preload magnitude can degrade the behavior and life span of the joint and lead to joint problems & failure.

Although bolted assemblies at first appear very simple in industry word, they cause several problems for designer engineers, assemblers, and maintenance departments. In reality, the design of a bolted assembly in ever where requires a methodical and rigorous approach, since mistakes can lead to failures, often costly and sometimes disastrous consequences.

Generally there are three kinds of threaded fasteners component as seen in figure 1.1:

- bolts and nuts,
- Studs with nuts on one end,
- Studs with nuts on both ends.



Bolt and nut　　Stud with nut on one end　　Stud with nuts on both ends

Figure 1.1 General kinds of threaded fasteners

1.2 Purpose of Tensioning of Fastener

Threaded fasteners are tightened for the obvious reason as seen in figure 1.2; the general purpose of the tightening load is multiple:

- Ensure the rigidity of the whole assembly and make it capable of sustaining external loads due to traction, compression, and bending moments and shear forces.
- Prevent leakage at the seals.
- Prevent shear stresses on the bolts.
- Resist spontaneous loosening effect.
- Reduce effect of dynamic loads on the fatigue life of the bolts.

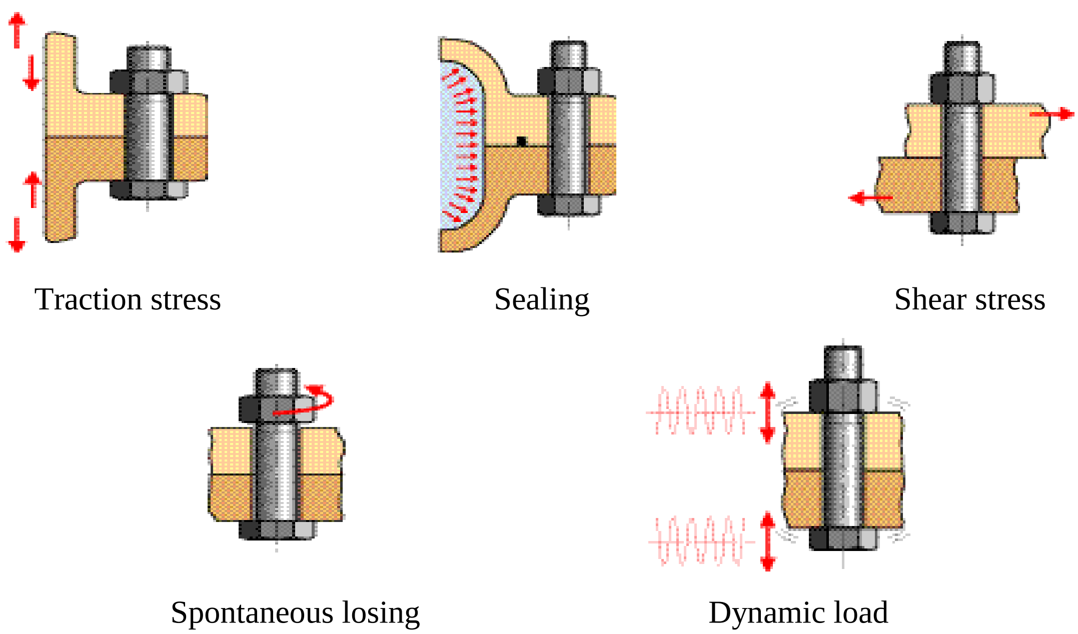


Figure 1.2 Purpose of the tightening

Correct tightening of a bolt means that making the best use of the bolt's elastic properties. The tightening process exerts an axial pre-load tension on the bolt. This tension load is equal and opposite to the compression force applied to the assembled components. It can be referred to as tension load.

1.3 Preload Magnitude

The term preload refers to the loading in a bolt immediately after it has been tightened. Investigations have shown that up to 90% of bolted joint failures are caused mainly by incorrect initial bolt tightening [1]. When accuracy assembly of parts required with multiple bolts, performance degradation can in many cases be attributed to non-uniform stress on the bolts, which is caused by the scatter of bolt preload. Thus, it is important to achieve an accurate magnitude of preload during the assembly process for a specific bolt joint.

The proper amount of tightening (pre-load) is very important. If the fasteners are too tight they may break off bolt - either during the tightening itself or when the working load is added to the pre-load in applications. If too loose, the fastener will shake loose in vibration. Often overlooked, but equally important, is the tendency of fasteners subjected to cyclic loading to fail from fatigue load effect if not sufficiently tightened.

Generally in threaded connection designs, most designers are often interested and focus with only the thread root area, i.e., The size of the screw thread and material of bolts, while the other factor, which is tightening method to reach the target preload, is often neglected by the designer, the other important parameter in the design is external load effects on structure, it is well known that preload magnitude in the bolted joints is reduced when external loads are applied to the joint after they are tightened. The external loads may be static or dynamic. The preload magnitude reduction of the bolts is observed in any bolted joints, which results in risks of failure of any machinery or equipment.

1.4 Why Bolts

Practically every engineering product with any degree of complexity usually uses one of the types of threaded fasteners. The significant advantage of threaded fasteners over the majority of other joining methods is easily and quickly assembly and disassembly with high rigidity results, in the same time it can be re- used. This important feature is often the reason why threaded fasteners are used in preference to other kind of joining methods. Advantages of threaded fasteners make it applicable comparing with other connection (screws, rivets, etc.), the important factors which help prefer bolts are following;

- Strong joints.
- Long lasting.
- Relatively more resistant to corrosion (depending on the coating and material used).

1.5 Behavior of Bolts and Nuts

Bolts are most often made of steel. Like most metals, steel is elastic, at least as long as the strain does not exceed the "elastic limit" beyond which permanent deformation occurs in metal. Within the "elastic limit", a metal part such as a bolt follows Hooke's law, for that reason strain (elongation) is proportional to the stress (load), as shown in the figure 1.3.

Any tightening method of bolts must ensure that the stress in the bolt never exceeds the point "A" (the elastic limit or "yield point" of material), both during the tightening operation and when the assembly is later exposed to efforts or loads during operation.

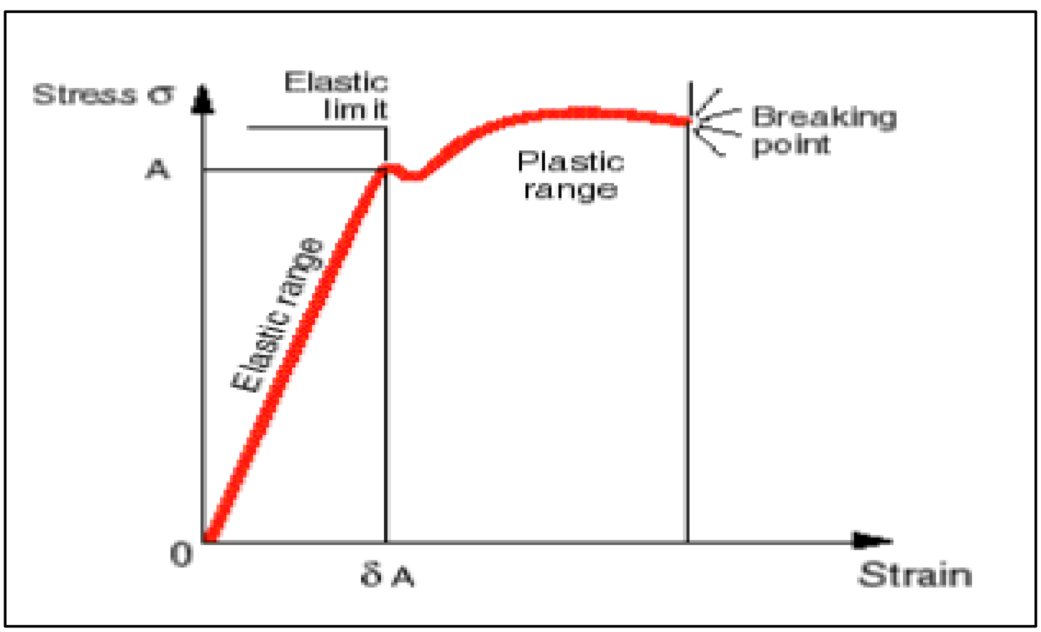


Figure 1.3 Stress strain curve

1.6 Methods for Controlling Tightens

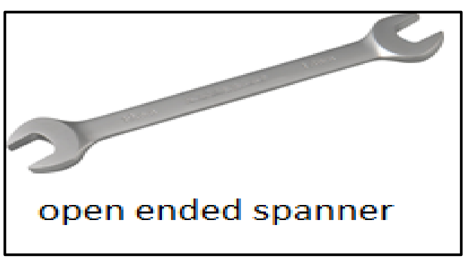
Six methods can be applied to control tightness of a threaded fastener. The order of increasing accuracy (an increasing cost), the tightening methods are:-

1. Feel.
2. Torque wrench.
3. Turn-of-the-nut.
4. PLI washers (preload indicator washer).
5. Bolt elongation.
6. Strain gages.

The decision as to which tightening method, select depends primarily on the criticality of the joint. Generally, the more critical the joint means more need for higher tightening accuracy and the greater the cost for obtaining the right preload magnitude. It should always be noted that the method of tightening selected will almost always lie between the two extremes,

1.6.1 Feel

For few applications will allow the high inaccuracy of the "feel" method tools Figure 1.4, while the high cost of tightening, highly accurate strain gages are used almost entirely in the laboratory and experimental tests.



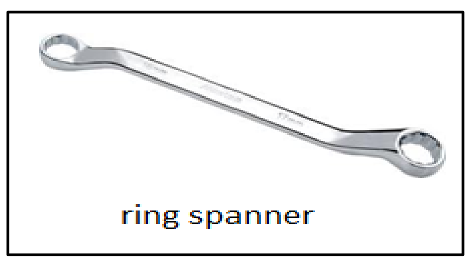


Figure 1.4 General types of spanners

1.6.2 Torque Wrench

This is the least expensive of the accepted methods of controlling fastener preload but it is the least accurate, chiefly because the reading of torque on dial is

Figure 1.5 Torque wrench

1.6.3 Turn – of – The Nut

It can be consider the “turn of the nut” method is common method. Briefly the nut is tightened to a “snug tight” position and then tighten further fraction of a nut turn depending on the assembled joint geometry. However, the standard does not specifically define the “snug tight” position. The “snug tight” position is where a step change in the gradient of the torque vs angle curve occurs. In order to carry out this method process with any accuracy preload magnitude a torque sensor and an angle encoder shall be used. Then by calibrating on the desired joint bolts with a direct tension measuring device which is fixed before, the required nut rotation after the snug tight position can be determined. This will then provide a method of tightening with some degree of accuracy. Its accuracy is affected by the workman care in measuring the angle the nut or bolt is turned.

1.6.4 Preload Indicating Washers (PLI washers)

A Preload indicator washer is sensitive only to axial load (Preload), it is interposed between the head of the bolt or nut and the surface of the joint. As the nut is tightened, the bumps (several tip on one site of washer) on the preload indicator washer yield plastically, reducing the gap between the head of the bolt and the washer. A feeler is used to measure this gap. When the gap has been reduced below a preselected maximum value, the tightening process is stopped. Figure 1.6 finally the inner ring has been flattened to this point; the correct preload force has been reached.

Figure 1.6 Load indicator washers

1.6.5 Bolt Elongation

In this method, a bolt from the lot is loaded in a tensile machine with the same nut as used in the application. Measuring distance from the nut face to the underside of the bolt head, and a plot by mathematical calculations using formulas is made of bolt elongation in relation to induced load. In the application, the fastener is tightened as a result of tightening the bolt elongation is measured until the required preload as determined from the plot has been achieved. However, this requires every bolt – nut pair to have almost exactly the same calibration curve as shown in figure 1.7, produced from the sample bolt – nut tensile test.

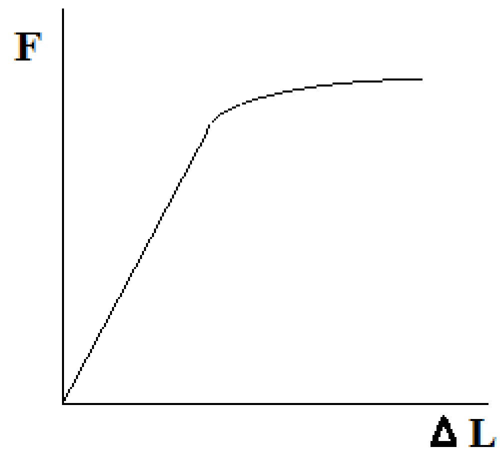


Figure 1.7 Calibration curve of a bolt – nut pair

1.6.6 Control Preload by Strain Gauge

A strain gage detects a minute dimensional change (strain) as an electric signal. By measuring strain with the gage bonded to the bolt structure as shown in figure 1.8, direct strain gauging of the bolt shank will produce accurate preload measurements. However, external instrumentation required to measure strain gage resistance change output during tightening and after tightening; method is very fragile and contains external wiring and use of cumbersome measuring equipment. After the strain gage fixed with bolts, bolts handled carefully due to the fragile nature of the strain gauges. These bolts resemble technology of load cells, in general, tightening by controlling preload with strain gage very expensive and mostly used for experimental purposes. The accuracy of this method may be 2 - 5%. [2]

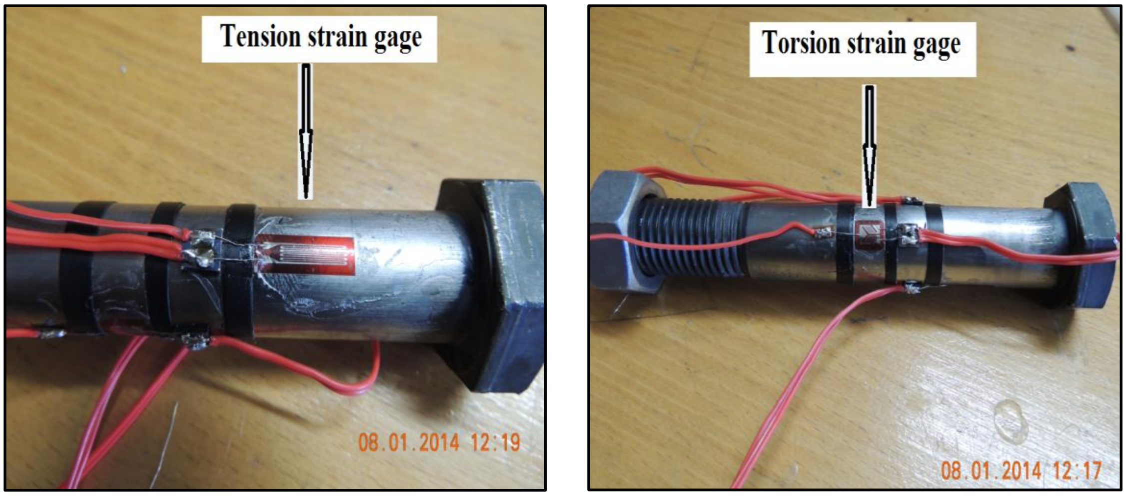


Figure 1.8 Two types strain gages fixed on bolt

1.7 Effect of Cyclic Loading on Preload Magnitude

Most mechanical connections by bolts are subject to dynamic loads. Rotating and reciprocating machinery generates significant cyclic loads. Two of the main failure modes of mechanical connections associated with cyclic loading is fatigue and preload loss.

Threaded fasteners are used in the member's joint for centuries, even in systems with high responsibility. However, fatigue failure effect of bolted joints remains a concern, the failure of only one bolt can promotes the instability and the consequent loss of an entire system that using bolts as fastener. Bolt fatigue strength depends on the magnitude of cyclic stresses it experiences and stress amplitude of the bolt depends on the magnitude external axial load, and also depends on the stiffness of the joint which is depend on preload, i.e., the bolt and the tightened members.

Most preloaded joints experience some amount of preload loss, due to two reasons; first plastic deformation (permanent set) on the assembly parts second the vibration effect. This criterion does not address preload loss due to effect of vibration, as some method of preventing the nut and/or bolt from vibrating loose should be part of the basic design of the joint in order to keep the design in safe mode. Preload loss can vary between about 2% and 15% of the actual preload level in the bolt.

However, preload loss must be considered in the analysis of preloaded joint design, figure 1.9 shows different types of external vibrations on a structure which lead to self-losing.

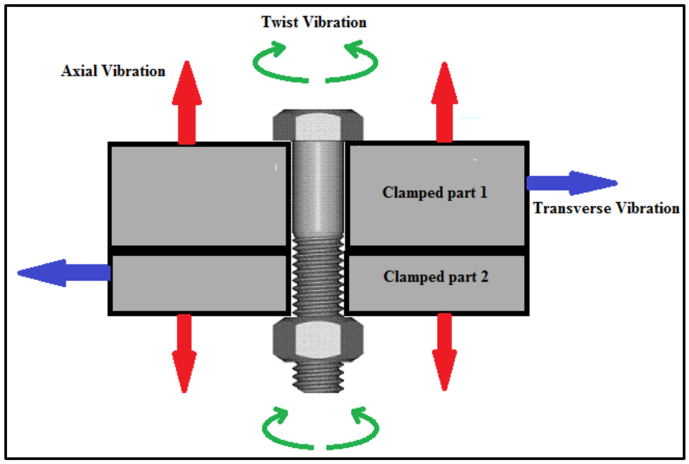


Figure 1.9 Principal external loading cases for clamping part of a bolted joint

1.8 A New Method of Bolt Tightening – Hydraulic Tension Method

A target bolt preload, can easily be achieved by applying a predetermined hydraulic force to the bolt body in axial direction without any torque applied to the bolt. This hydraulic force can be achieved by creating a controllable hydraulic pressure applied to a pre-calculated surface area of a hydraulic jack/cylinder.

This hydraulic force/load, by means of a suitable mechanism, can be transferred to the snug-tight bolt body stretching it along its axis "with no parasite torsion". The resulting preload on the bolt body is then directly proportional to the hydraulic pressure applied by the jack. Section of an experimental setup is designed and constructed to apply hydraulic load to the bolted connection as seen in figure 1.10.

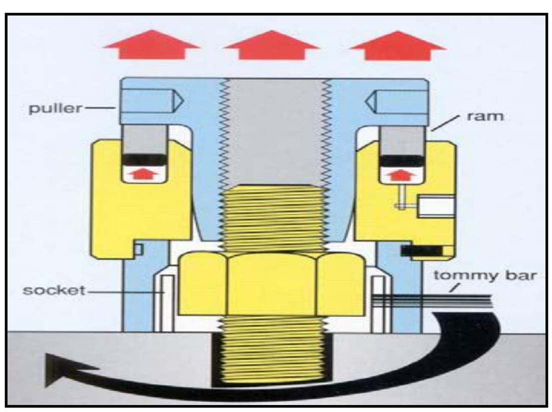


Figure 1.10 Hydraulic tension systems [3]

1.9 Object of Study

Although there are different tightening techniques, methods in the industry, the conventional torque wrench method is used widely, with lots of inaccuracies in both torque applied and the resultant pre-load created within the bolt body. Not only the axial pre-load, but also the torsional stresses created in the bolt body are inaccurate. Whereas new methods like direct bolt tension does not create any torsional stresses in the bolt, and the bolt pre-load can be set more accurately than the torque wrench method. In this study, a comparison of the two methods will be made (conventional torque tightening method and hydraulic tension method) by using both torque wrench device and hydraulic bolt tensioning device in tightening the same site bolts (M14, M16 and M18), and measuring the stresses created within the bolt in each method. Static preload magnitude measure after tightening complete in each method will be discus and compare. Also study the behavior of preload under dynamic loads effect is very important in order to show that a decrease in preload magnitude using the conventional torque method is greater than bolt hydraulic.

1.10 Layout of Thesis

This thesis consists of six chapters which are organized in such a way that; in the following chapter two, comprehensive literature survey of research on the object is summarized, the historical background on the threaded fasteners, also effects of friction between surfaced during tightening on preload magnitude in conventional torque tightening, and investigations in hydraulic tension tightening, finally self-losing and cyclic load effect on preload magnitude.

In the third chapter, present basic information about conventional torque tightening method and hydraulic tension tightening method varies. The parameters that use in calculating the correct torque magnitude and additional parasite torsion stresses are discussed for the conventional torque tightening method. For hydraulic tensioning method, principal of tensionening procedure as well as calculations of hydraulic pressure are discussed. Finally prload bolt separation criteria and cyclic loading effect on preload magnitude discussed.

In the chapter four, the bolts geometries, the property class of the bolts that used in this study and the single bolt cylinder which is used as structure in both tightening methods in static and dynamic test. In the second part of this chapter is giving the methodologies for calculating target preload for M14, M16 and M18. Torque magnitudes and hydraulic pressure magnitude calculations in both tightening methods.

Chapter five presents the experementally measured result data that have been recorded during both tightening methods in static and dynamic parts. The results calculated are turned into graghics and tabels in order to compare two tightening methods(conventional torque tightening and hydraulic tension method).

Conculusions of the experemental invistigation are presented in chapter six.

CHAPTER 2
BACKROUND AND LITERATURE SURVEY

2.1 Introduction

Although crude fasteners had been around since early civilizations, being employed in some fields for example carts and agricultural equipment, the level of technological improvement was slow and fairly static for hundreds of years until the Industrial Revolution in Europe. As with so many other things, this new era meant that large numbers of screws and bolts could be produced in a shorter amount of time to cover manufacturing requirements, and with more kinds and precision. By the mid 1700's, the Wyatt brothers in England succeed in manufacturing 150,000 wooden screws a week.

In other site of word in America, several companies were making fasteners by the late 1700's. However, there was no standard for the size or thread density from business to business. The Rugg & Barnes Company and the A.P. Plant Company, both companies established and integrate and start manufacturing bolts and nuts in the early 1840's, this step can be considered as the first large manufacturers to focus solely on making fasteners. In a matter of a few years, another new historical event loomed that would begin to bring and push the industry into the modern age: the Civil War in America brought with it a huge demand for machinery, and by extension, screws, nuts, and bolts.

It was during this period that the need for developing an American thread standard became apparent, and William Sellers enter the picture. In 1864, he proposed a uniform system of screw threads which differed from the British (Whitworth) standard in that the tops and bottoms of the threads are more rounded rather than flattened. The twentieth century saw steady growth for the industry in the U.S., and by 1969 there were 450 companies, 600 plants, and more than 50,000 people employed in the manufacture of fasteners.

Today, in a slightly exaggerated sense, bolts hold together our entire civilization. Billions of bolts are manufactured throughout the year for the widest variety of uses.

Trayer (1932) had focus on the root of threaded fasteners, the roots of modern bolted timber connection design by Trayer, methods for computing safe design loads varied extensively due to a lack of physical test data that quantified connection strengths. As a result, Trayer runs several hundred tests on specimens of various configurations in an attempt to provide an understanding of recommendations for the design of bolted connections. From test data, Trayer produced empirically based design formulae for bolted, double-shear joints and made recommendations about proper bolt spacing, end distance, alignment, and choice of bolt diameter. It was within this work that the term proportional limit stress was first introduced "as the average stress under the bolt when the slip in the joint ceases to be proportional to the load.

In an effort to determine what is known and not known about bolted connection performance, Soltis and Wilkinson (1987) did an extensive review of all previous single and multiple-bolt connection studies. Their findings suggest that, among other things, moisture content, spacing, end and edge distances, fabrication tolerances and fastener aspect ratio have a direct effect on the performance of bolted connections. The Yield Limit Model also requires information on the fastener bending strength and dowel embedment strength for a determination of overall joint performance. Other factors that may effects the strength characteristics of a bolted timber connection include friction and bolt tensioning effects.

2.2 Studies on Preload Magnitude

2.2.1 Axial Type Loading

E. Hemmati Vand, R. H. Oskouei, and T. N. Chakherlou (2008) performed an experimental method to measure the clamping force value "preload" at bolted connections due to application of tightening with wrenching torque to tighten the nut have been Presented. This study was designed based on Hooke's law by measuring compressive in the axial strain of a steel bush placed between the nut and the plate.

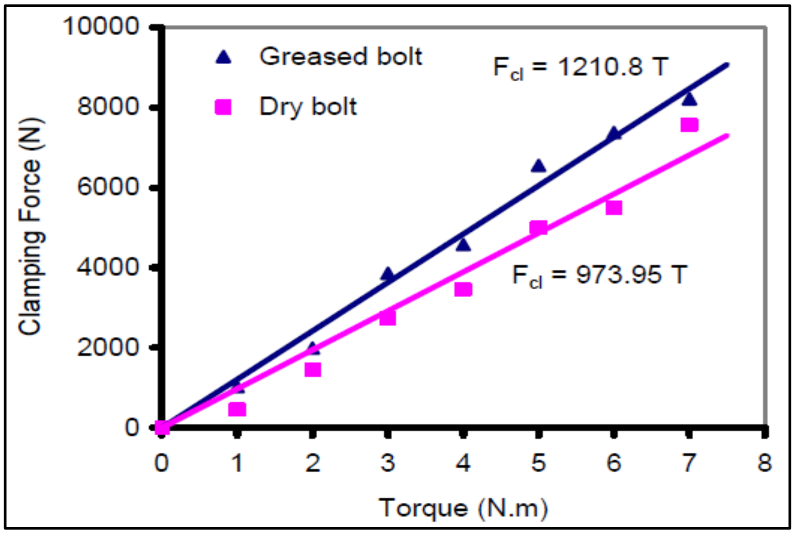


Figure 2.1 Clamping force and torque curve [6]

It is found in the study that the lubrication of bolt threads provides a higher preload in the joint comparing with dry conditions, because of a decrease in friction coefficient between bolt and nut, and consequently, decreases in torque coefficient from 0.205 to 0.165, for the dry and lubricated conditions, respectively. To conclude, applying torque to the lubricated bolt creates higher preload that is desirable and safer for the design of mechanically fastened joints connecting main parts of the structures.

J.G. Williams, R.E. Anley, D.H. Nash, T.G.F. Gray (2009) The behavior of a simple single-bolted-joint under tensile separating loads is analyzed using conventional analytical methods, a finite element approach and experimental techniques. The variation in bolt force with external load predicted by the finite element analysis conforms well to the experimental results. For experimental analyses AS 21523 bolts and nuts, featuring integral washer contact were used. Several observations can be made from study; firstly, the transfer of external load to the bolt is small in all cases. And the second result is that increasing the pre-load level in the experimental case reduces the transfer of load to the bolt, finally discrepancies between the finite element and experimental results appear during study.

Ralph S. Shoberg (2010) studied method of estimating the residual tension (preload) achieved by a fastener in a bolted assembly without controlling torque to reach target preload. Ralph S. Shoberg technologies developed and patented an auditing method known as the “M-Alpha Torque-Tension Audit” to estimate the residual clamping load ”preload” without having to rely on strain gaging a bolt, using ultrasonic methods, or using force washers. The ability in this method to estimate both residual torque and residual tension on a bolted joint is the reason that the “M-Alpha” method auditing process is considered to be a significant technological breakthrough for fastener tightness technology.

The angular displacement (turn) of the fastener lead to stretches the bolt and compresses the structures being clamped. The elastic clamping region of the torque-angle signature can be considered is the region where clamping force is directly proportional to the angle of turn from the “elastic origin.” So tightening bolt in elastic origin is located by the projection of the straight-line portion of the signature curve backwards to original zero torque, or the prevailing torque level in the case of the thread locking interference type of applications.

This principal can be observed when viewing a clamp load-angle curve figure 2.2. Once the curve becomes linear relation, the clamp load will increase at a consistent rate with each angle of rotation. This relation remains true until the curve stops being linear due to yielding of the fastener or the joint.

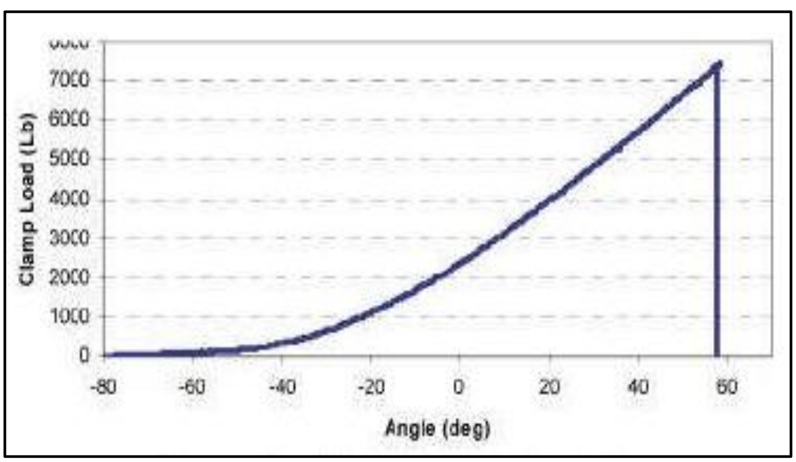


Figure 2.2 Clamp load vs angle relation [8]

2.2.2 Shear Type Loading

Fouad Hilmy Fouad (1978) worked on experimental study for slip behavior of bolted friction – type joints with coated contact surface, the study covered about 600 friction – type bolted joints were tested to evaluate the slip characteristics of five different coating systems on the contact surfaces and to study the influence of several variables on their slip behavior. The coating system were (organic zinc – rich primer, organic zinc – rich primer with an epoxy top coat, inorganic zinc – rich primer with a vinyl top coated, powder epoxy and a vinyl system). The variable considered where the hole size (15/16, 1 and 1-1/8 in. diameter holes).

Based on the test results, allowable bolt shear stresses for friction type with coated contact surfaces are provided. Comparisons are made with current bolt specifications.

Saman Fernando (2001) carried out a study on engineering insight to the fundamental behavior of tensile bolted joints. In order to shed some light onto the behavior of a generic tensile bolted joint a comprehensive 3D Non-linear Elasto--Plastic Finite Element model analysis has been conducted. For simplified analysis purposes, two types of tension, joint load configurations considered, the first type is an external load is applied at the surface adjacent to the nut and the head, while the second type is an external load is applied at the jointed interface. The study results were computed that applied load on the bolt load in a preload tensile joint depends on the stiffness ratio of the bolt and the joint, longer bolts provide better properties of a dynamic joint than shorter bolts and correct preload is paramount in achieving high--performance dynamic tensile joints, finely calibrated torque wrench is not a reliable method of achieving a desired bolt tension.

Christine Heistermann (2011) had investigated by behavior of pretension bolts in friction connections. Four different bolt types have been investigated with respect to the ease of installation and maintenance on the one hand and structural applicability on the other hand the latter one is mainly defined by the behavior of the pretension force in the bolts. Various influences on the reduction of clamping force are experimentally checked, such as the type and thickness of coating, the thickness of the clamping package and external loading, it was found that the longer lock bolts perform better than the short ones. Including the extensional sleeve between clamped plates and the nut, from the test results with a variety of cyclic load can be concluded that the additional losses due to external loading are small compared to the losses resulting from pure relaxation figure 2.3. Finite element used in calculations, consider a perfect roughness of the coating and are represented by a constant friction coefficient. Therefore, a simple, direct correlation between the losses of pretension in the two bolts and the slip resistance can be observed. It is therefore expected that for higher losses of pretension in the bolts if there is less slip resistance between joint parts.

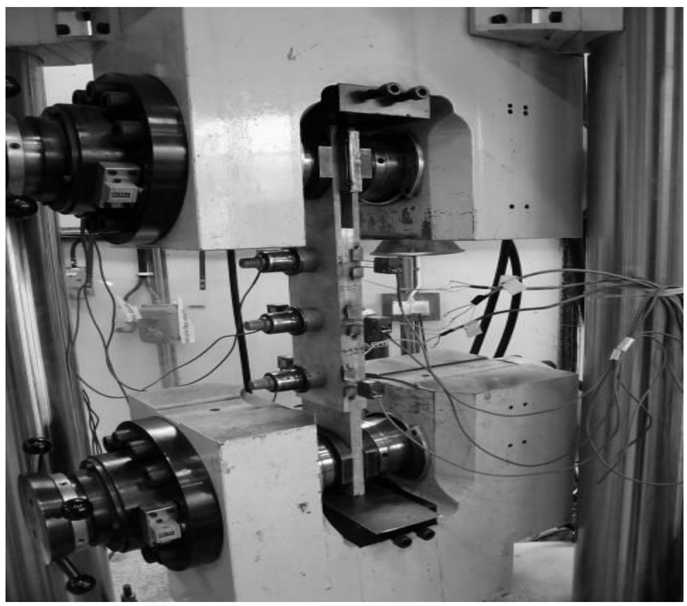


Figure 2.3 Picture of the test rig for the performed static and fatigue test [10]

2.3 Friction Effect on Torque

Sayed A. Nassar, G. C. Barber, Dajun Zuo (2004) Carried out a study on bearing friction torque in bolted joints, this analysis focuses on the bearing friction torque component under the turning head of a threaded fastener figure 2.4. Further, it analyzes the error contained in the current practice when an approximate value, equal to the mean contact surface radius, is used instead of the actual bearing radius. The new formulas for the bearing friction radius are developed from a mathematical model of a bolted joint using four different scenarios of the contact pressure distribution under the rotating fastener head or nut. New formulas developed in this paper provide a more accurate calculation of the fastener under head bearing friction torque component. Similarly, the effect of the variable sliding speed of the bearing friction torque component increases with increasing the bearing radii ratio. Finally, the numerical results show that the modeling of the fastener under head pressure has a significant effect on the bearing friction torque component and on the overall fastener torque-tension relationship.

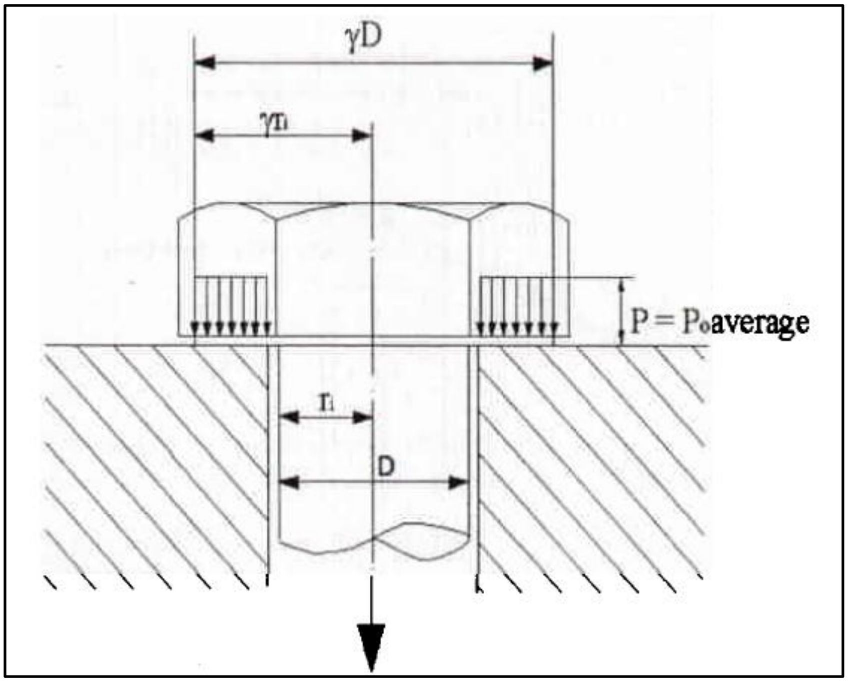


Figure 2.4 Constant Under head Pressure [11]

2.4 Hydraulic Tightening

E. Giacomelli , F. Graziani and P. Battagli (1990) carried out a study on advantages of the hydraulic tightening of threaded connections, For the tightening method virtually exclusive characteristic of hydraulic tightening is that supply great force, practically unobtainable with other systems, a hydraulic tightening thread becomes indispensable when screw threads are large in size. The study discussed hydraulic tightening provides significant advantages for the users of the machine. The operating restraints for tightening can be reduced to a minimum. Especially those linked to the amount of preloading, furthermore opportunities provided are systematically and rationally utilized up through final formulation of the design. The experimental results proved that using the hydraulic tightening method the reliability and performance of connections can be improved, so dimensions and weights of bolt and structure can be reduced. Furthermore, if all the advantages and opportunities provided are systematically and rationally utilized up through final formulation of the design, the adoption of hydraulic tightening method results in machines that are more compact, lighter and often less expensive cost.

Linear Motion & Precision Technologies company engineers (SKF) (2001) made study on hydraulic bolt tightening method, the investigation covers the advantage and disadvantage of conventional torque tightening method and also it contain methods and devices for measuring tightening torque. Hydraulic bolt tensioning procedures and advantage of this method cleared, the study contained the technical analyses of bolt tensioning method. Finally the study summarized results of hydraulic bolt tensioning offers the following advantages:

- No “parasite” torsion stress in the bolts,
- High preload precision simply by controlling the hydraulic load,
- Fast, safe and easy operation,
- The integrity and the reliability of threads and bearing surfaces are maintained,
- Readily compatible with simultaneous tensioning of all the bolts in an assembly (or even several assemblies),
- Use with a wide range of bolt sizes: 5 to 500 mm,
- Easy adaptation due to modular design,
- Greater homogeneity due to excellent efficiency.
- Easy implementation to control preload.

2.5 Investigations on Self Loosing of Threaded Fasteners

A well-known problem of bolted connections is that they open by themselves after assembly also after applied external load. This self-loosening effect leads to the necessity of regular control and maintenance of bolted connections. Loosening issues tend to occur where dynamic loads act on joints secured with threaded fasteners.

Loosening of the single – tooth implant screws is an ongoing problem. An American paper reports that 43% structural screws came loose in the first year (Aboyoussef et al. 2000). A Japanese paper reports that 26% of screws needed re-tightening in the first year (Khraisat et al.2004).

William Eccles (2010) carried out a study on tribological aspects of the self – loosening of threaded fasteners. The study had investigated a number of issues with the tightening and self – loosing of threaded fasteners, it was found that upon repeated tightening of electro – zinc plated fasteners significant wear of the contact surfaces of the bolts / nuts thread and face occurred. The other result that William Eccles reached during investigation, if an external axial load is acting whilst the joint is experiencing transverse slip, under the appropriate conditions, the loosening process will continue until nut detachment occurs. The study found that the friction coefficient in the circumferential direction in the threads is greater than that on the face bearing surface during conditions of transvers of slip.

Saman Fernando studied on mechanisms and prevention of vibration loosening in bolted joints, the total loss of the fastener or subsequent fatigue failure due to loss of bolt pre-tension are the predominant failure modes of vibration loosening discussed. The present paper identifies critical parameters in preventing loosening and analyses two possible mechanisms of vibration loosening. The mathematical models developed shed light on the effect of various bolted–joint related parameters on vibration loosening and joint integrity. It also develops simple rules-of-thumb for the prevention of vibration loosening. Finally, the paper discusses the different types of nuts to prevent losing in preload magnitude in bolted joints.

Anirban Bhattacharya , Avijit Sen , and Sntanu Das (2010) present results on the anti-loosening characteristics of threaded fasteners under vibratory conditions. This investigation an attempt has been made different test the anti-loosening ability of various locking screw fasteners methods, such as nylock nut, aerotight nut, cleveloc nut, chemical lock, flat washer, nylon washer, spring washer and serrated washer with bolts of different materials, sizes and types with different initial clamping forces “preload” under the accelerated vibrating conditions. On the basis of the test results, chemical locking has been found to show best anti-loosening nut characteristics followed by nylock and aerotight nut as included in figure 2.5.

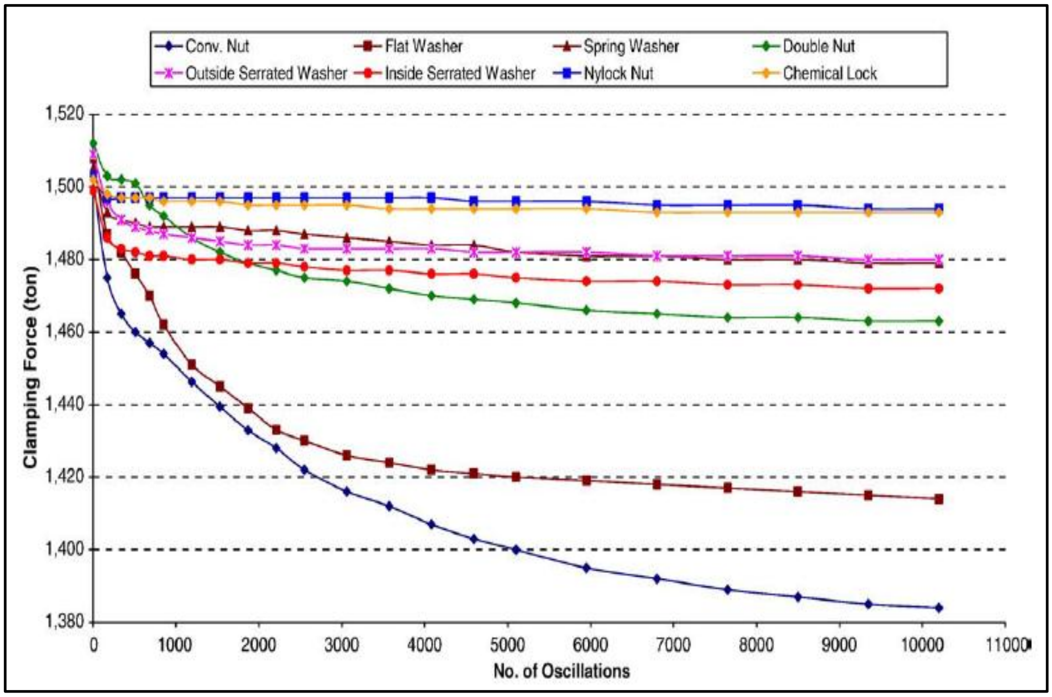


Figure 2.5 Comparison of loosening for Metric High Tension Steel Bolt (M16) with different nuts [16]

Also from the results obtained that not only the bolt material, but other fastening elements such as washers or nuts also play a key role behind the anti- loosening property of the fasteners in the same time initial tightening torque also plays a significant role behind the self-locking property of fasteners. The initial loosening rate is quite high and depends on clamping force magnitude.

2.6 Fluctuating Load Effects on Preload

Bolts and screws play an important role in the performance of machinery. The majority of these fasteners are subjected to fluctuating loads, leading to the well-known phenomenon of fatigue which is responsible for most of the premature failure in bolts. There are many parameters such as preload magnitude, tightening type, thread pitch.

Eugene I. Radzimovsky (1952) carried out a study on bolt designer for repeated loading, a new approach to strength calculations for bolts subjected to periodically change loads. The study satisfactory for assemblies under steady load conditions, however, they become unsatisfactory if the load acting on the bolted assemblies periodically changes. The study developed a rational method for calculating the strength of the bolts for repeated loading conditions based on the actual forces acting on the bolt, the range ratio of these forces, stress concentration, local stresses, and fatigue properties. Also included is a study of how the size and form of individual bolt parts influence the reliability of bolts. The tightened bolted assembly can be represented schematically as a system which consists of two springs; figure 2.6 one of these springs represents the bolt and is extended; the other spring represents the connected parts and is compressed.

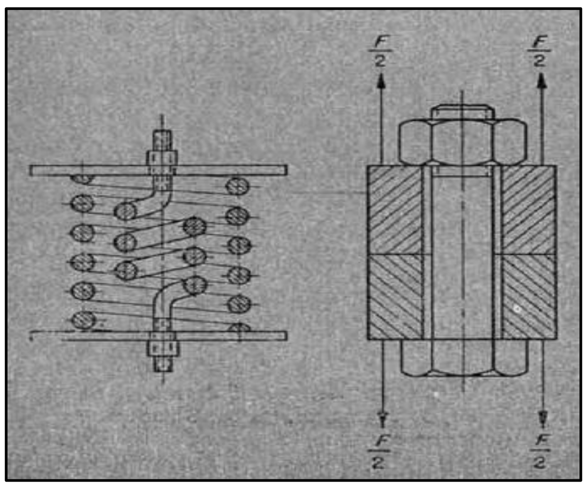


Figure 2.6 Spring analogy for a bolted assembly [17]

G. W. Skochko and T. P. Herrmann (1992) investigated factors which affect fatigue strength of fasteners, by using appropriate design factors between the failure and design fatigue strengths, such tests are used to establish fatigue failure and design parameters fasteners for axial and bending cyclic load conditions. The study reviews the factors which influence the fatigue strength of low Alloy steel threaded fasteners, identifies those most significant to fatigue strength, and provides design guidelines based on the direct evaluation of fatigue tests of threaded fasteners. Influences on fatigue strength of thread manufacturing process (machining and rolling of threads), effect of fastener membrane and bending stresses, thread root radii, fastener sizes, fastener tensile strength, stress relaxation, mean stress, and test temperature are discussed.

Feargal Peter Brennan(1992) studied on fatigue and fracture mechanical analysis of threaded connections, the thesis aims to develop a comprehensive usable engineering design approach to the fatigue analysis of threaded connections. Fatigue crack initiation is discussed with reference to the specific setting of a critical thread root. A crack initiation model is adapted for employment in thread root design. A method has been developed for predicting fatigue life in large threaded connections under random loading, It was concluded that the greatest discrepancies between experimental results and the fracture mechanics predictions. One of study results that the stress analysis is generally divided into two phases: (i) evaluation of the load distribution within the entire connection, and (ii) evaluation of the stress state in the region of a loaded projection (bolt tensile area).

K. Din and M. T. H. Ghazali (2004) fatigue life of bolt subjected to fatigue loading conditions, this experimental study was conducted. Two sizes of High Strength Friction Grip (HSGF) bolt were chosen and they were subjected to low and high cycle constant amplitude loading condition figure 2.7. Each of them undergone 4 different stress ranges and at 3 means stress level. Three bolts from each size were tested under static loading in order to obtain their mechanical properties. Unlike for high cycle fatigue, the low cycle fatigue where Smax/σY is between 55 - 95%, the fatigue life of bolt is not significantly influenced by the mean stress and the stress range.

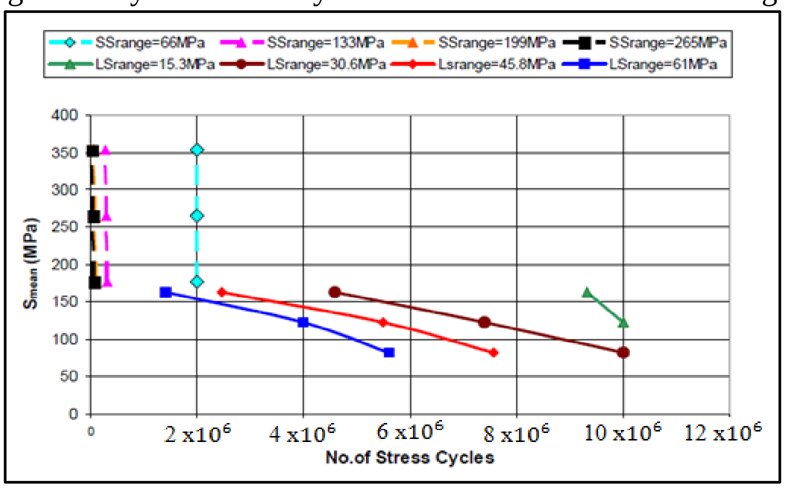


Figure 2.7 Linear plots of the mean stress and number of stress cycles [20]

CHAPTER 3

THEORY OF BOLTS AND CONNECTIONS

3.1 Introduction

Prior to understanding the “whys” of this study, it is necessary to know some of the “hows” and “whats” of basic tightening of threaded fasteners. The aim of this chapter is to present basic information about conventional torque tightening method and hydraulic tension tightening method varies. The parameters that use in calculating the correct torque magnitude and additional parasite torsion stresses are discussed for the conventional torque tightening method. For hydraulic tensioning method, principal of tensionening procedure as well as calculations of hydraulic pressure are discussed.

The strain gauge types and configuration attached to bolts and System 7000 from Micromeasurment which is used to measure the preload and turn magnitude during tightening and after tightening are discussed. Finally prload bolt separation criteria and cyclic loading effect on preload magnitude discussed.

3.2 Conventional Torque Tightening

This is the least expensive of the accepted methods of controlling fastener preload magnitude, but is one of the least accurate method, chiefly because the reading on dial is affected by the friction that generate on threads and under head of bolt and nut to be overcome.

The literatures show that the uncertainty associated with use of torque to establish a target preload is typically between 10 and 35%. These studies have been performed under controlled, laboratory conditions.[21].

3.2.1 Arise of Preload during Torque Tightening

When a bolt is tightened, by applying a torque to the bolt head or nut, the nut advances with the pitch of the thread and compress the parts between the bolt head and nut, the force of compression created by the advance of nut create on equal and opposite force on the bolt body, this force stretching the bolt is called the tightening load or preload. The bolt gets stretched during tightening causing the clamping plates to be compressed together as shown in figure 3.1

Fi: bolt tension or preload prior to any joint settling.

Fm: clamp load in the clamped pieces

MA: applied torque or input torque.

MG: thread torque or the reaction torque to the input torque.

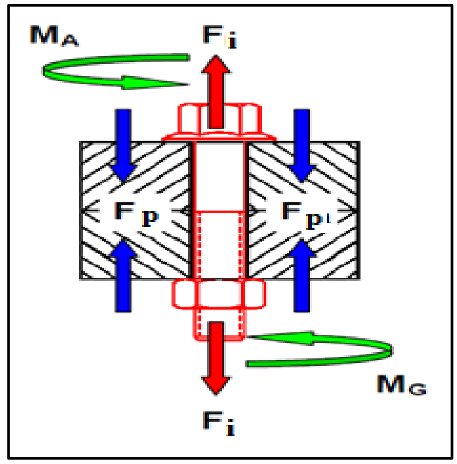


Figure 3.1 Loads during torque tightening

The thread torque can be described as the torque it takes to keep the nut from turning and loosing when an input torque is applied to the bolt head. During the tightening process the bolt tension preload is equal to the clamp load and this value are often interchanged and depends on final torque magnitude. This is perfectly correct until the joint is in service and external loads are applied.

High preload keeps bolts tight, increases joint strength after tightening, creates friction between parts to resist shear and other external loads effect on member, and improves the fatigue resistance of bolted connections. The recommended preload value Fi, which can be used for either static (stationary) or fatigue (alternating) applications, Fi can be determined from formula :

$$Fi = 0.75 \times At \times Sp \quad \text{(reusable connections)} \tag{3.1}$$

$$Fi = 0.9 \times At \times Sp. \quad \text{(permanent connections)} \tag{3.2}$$

In these formulas, (Fi) is the bolt preload, (At) is the tensile stress area of the bolt, and (Sp) is the proof strength of the bolt. Value of proof strength can be obtained from:

$$Sp = 0.85 \times Sy \tag{3.3}$$

where Sy is the yield strength of the material. Soft materials should not be used for threaded fasteners.

3.2.2 Calculation of Torque Tightening Method

In most applications of threaded fasteners, a very high percentage of the input torqueing power is consumed in overcoming the combined effect of two kind frictional torque components. The torque applied to a fastener is absorbed in three main areas. First, there is under head of bolt or nut friction, which may absorb 50 percent or more of the total torque as shown in figure 3.2

M 14 nut after applying torque 173 N.m   M 14 new nut

Figure 3.2 Friction effects on nut surface

While thread friction absorbs as much as 40 percent of the applied torque, the final 10 percent of the applied torque only develops the clamping force that holds the components together as shown in figure 3.3. The torque-tension relationship in threaded fastener applications is very sensitive to friction variations.

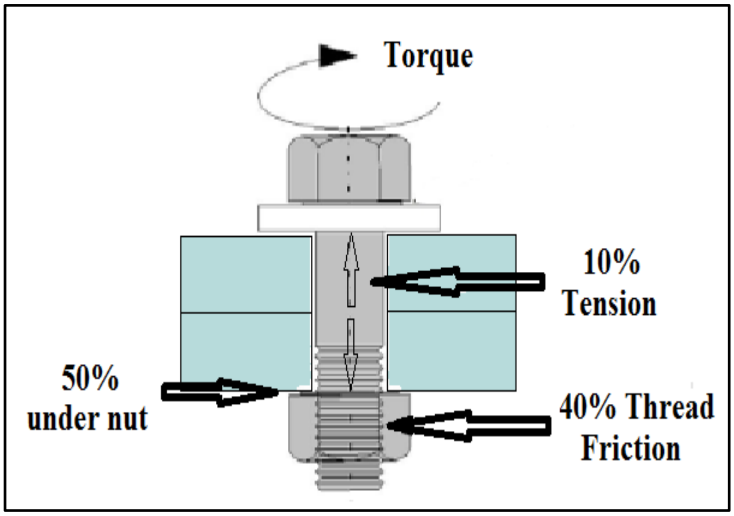


Figure 3.3 Torque absorbed on bolt and structure

A basic equation to estimate the torque-tension relationship is:

$$T = K * d * Fi \tag{3.4}$$

Where

$T =$ Torque

$K =$ Nut factor

$d =$ Nominal diameter

$Fi =$ Preload

In the prevailing torque tightening method, a more accurate torque-tension relationship for a threaded fastener as follows;

$$T = \left[\frac{P}{2\pi} + \frac{(\mu_t)(r_t)}{\cos\beta} + (\mu_b)(r_b)\right] Fi \tag{3.5}$$

Where, T is the input tightening torque applied to the fastener , Fi is the fastener preload, p is the thread pitch, (μ_t) is the coefficient of friction between male and female threads, (μ_b) is the coefficient of friction between the bearing surfaces under the turning fastener and member surface, (r_t) is an effective contact radius between threads, (r_b) is an effective bearing radius of the bearing contact area under the turning head or nut, and (β) is half of the thread profile angle which is 30° for standard threads.

Equation (3.5) may be expressed as

$$T = T_p + T_t + T_b \qquad (3.6)$$

Where (T_p) is the pitch torque component that creates the fastener tension and joint clamping force Fi,(T_t)is the torque component that overcomes the friction between male and female threads, and(T_b)is the bearing friction torque component that overcomes friction between the turning fastener head or nut and the clamped joint surface.

3.2.3 Typical Nut Factors

The typical nut factor must be typical of the type of bolt-to-joint interfaces used in the specific application. In addition, the nut factor must also be typical of the type and size of nut (or bolt head) used in the specific application, table 3.1 shows common nut factors. The parameters that must be considered in selecting a typical nut factor are the following:

1- Material at the bolt-to-joint interfaces.

2- Surface finish at the bolt-to-joint interfaces.

3- Lubricants at the bolt-to-joint interfaces.

4- Nut (or bolt head) type and size.

Table 3.1 Common nut K factor [22]

Nut Factors (K)			
Lube	Min. Reported	Mean	Max. Reported
As-Received	0.158	0.2	0.267
Alloy or Mild Steel Fasteners As-Received	-	0.2	
Stainless Steel Fasteners			
Cadmium Plate (Dry)	0.106	0.2	0.328
Copper Based Anti-Seize	0.08	0.132	0.23
Cadmium Plate (Waxed)	0.17	0.187	0.198
Fel-Pro CS4	0.08	0.132	0.23
Fel-Pro C70	0.08	0.095	0.15
Fel-Pro N 5000 (paste)	0.13	0.15	0.27
Machine Oil	0.10	0.21	0.225
Moly Plate or Grease	0.10	0.13	0.18
Never-Seeze (Paste)	0.11	0.17	0.21
Neolube	0.14	0.18	0.20
Phos-Oil	0.15	0.19	0.23
Solid Film PTFE	0.09	0.12	0.16
Zinc Plate (Waxed)	0.071	0.288	0.52
Zinc Plate (Dry)	0.075	0.295	0.53

3.2.4 Incorporation of Additional Parasite Torque Stress

In addition to the desired axial tension stress, torque tightening method introduces additional “parasite” torsion stress in the bolt during tightening which can reach over 30% of the tension stress. [13]

The resulting equivalent stress in the bolt tightening can be by using Von Mises stress formula ,is greatly increased and can exceed the critical point (yield point) of the material, whereas the tension stress itself which is generated from preload remains within admissible limits as shown in figure 3.4. Furthermore, the residual torsion stress increases the risk of spontaneous loosening at a later stage after tightening complete.

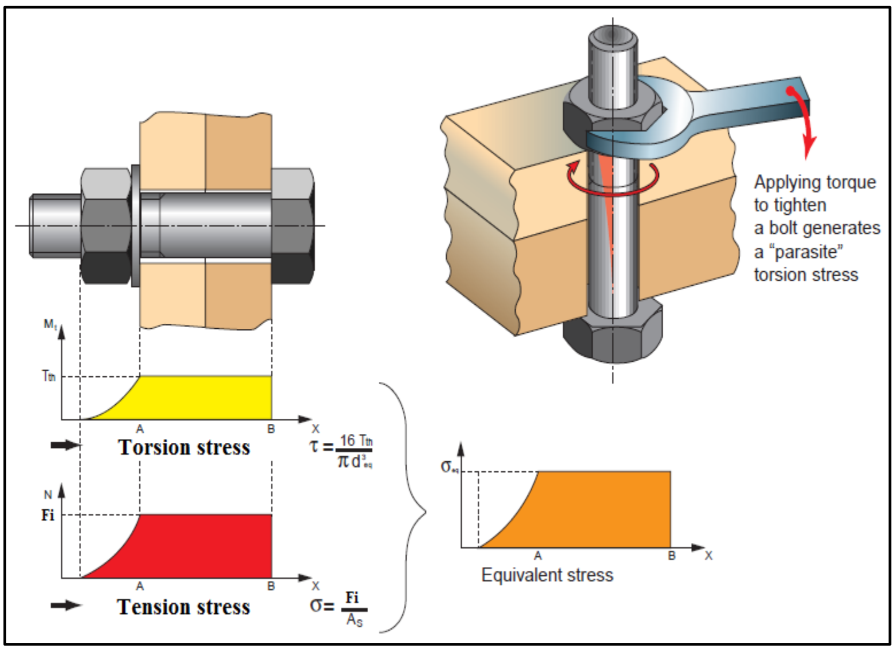


Figure 3.4 total equivalent stress in bolt [13]

For this adjustment, the combined tensile stress (von Mises stress) $\sigma\ total$ in (MPa) can be calculated from the following:

$$\sigma\ equivalent = \sqrt[2]{\sigma t^2 + 3\,\tau^2} \qquad (3.7)$$

$$\tau = \frac{16\ T_t}{\pi d^3} \qquad (3.8)$$

$$\sigma t = \frac{Fi}{A} \qquad (3.9)$$

Where

σt = tensile stress

τ = torsion stress

d= equivalent diameter

Fi = preload

A = area

T_t = torque bolt/ nut threads, it can be calculte from below formula

$$T_t = Fi\left(\frac{P}{2\pi}\right) \tag{3.10}$$

Where

Fi = preload

P = thread pitch

.

3.2.5 Bolt Yield

The torque tightening method is commonly used in automotive engine assembly rehabilitation for connecting rod bolts, engine head bolts, and crankshaft bearing cap bolts. Also in the civil engineering field, when bolts first replaced rivets in the construction of buildings and bridges, tightening beyond the yield point quickly proved to be a reliable method of fastener assembly. The preload obtained by tightening beyond the yield point is related to many factors, one of them the material yield strength, hence threads is the weakest part of the bolt. A section of the threaded portion of the bolt suffers a reduction of area and will neck out when tightening over yield strength. The change in stress area makes the bolt considerably weaker and as the bolt is stretched even further the clamping load decreases.

The other factor inversely proportional to the thread friction coefficient. The thread friction coefficient is very important since the yield point during tightening results from two components, first tensile loads, second the torsional load due to the thread friction and pitch torque. Table 3.1shows the nut factor coefficient varies from application to application.

3.2.6 Modeling the Tightening Process

Achieving proper control of the tightening process is possible by understanding the relationship between torque applied and development of tension. It is necessary to become familiar with what actually happens when a fastener is tightened. The general process of tightening a fastener involves turning nut, advance of the lead screw, and torque, turning moment, so that preload, tension, is produced in the fastener as result. The desired result is a clamping preload to hold components together.

The most general model of the torque turn signature for the threaded fastener tightening process has four distinct zones, as shown in figure 3.5

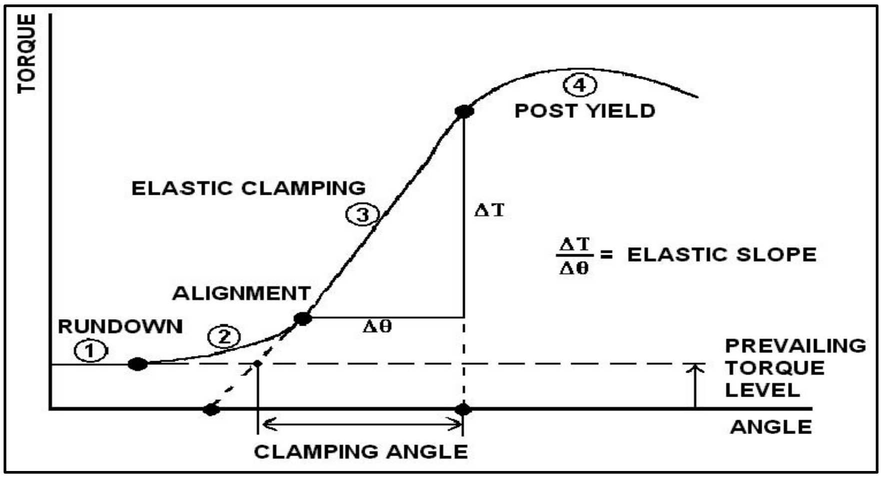


Figure 3.5 Four distinct tightening zones[23]

The first zone in tightening is the rundown or prevailing torque zone that occurs before the threaded fastener head or nut contacts the bearing surface of member.

The second zone in tightening is the alignment or snugging zone wherein the fastener and joint mating member surfaces are drawn into alignment to achieve a “snug” condition.

The third zone in tightening is the elastic clamping range, wherein the slope of the torque-angle curve is essentially constant.

The fourth zone in tightening is the post-yield zone, which begins with an inflection point at the end of the elastic range this zone is critical zone. Occasionally, this fourth zone can be due to yielding in the joint parts or gasket, or due to yield of the threads in the nut or of the fastener or clamped components, or the bolt/ fastener body itself

3. 3 Hydraulic Bolt Tensioning Method

A hydraulic bolt tensioning, however, can achieve accurate and predetermined bolt loading (preload) in a single simultaneous operation, providing the uniform gasket compression essential for the integrity of critical bolted fastener connections. Particularly on larger flanges and bid bolt sizes, also hydraulic bolt tensioning can also be significantly quicker than a conventional torque tightening.

Hydraulic tensioning methods allow controlled tightening up to the recommended tightening magnitude. The force is produced and applied without any parasite stress torsion, acting on the axial direction of the bolt. Since this is also done without friction and the nut can be turn without any friction, so friction under the nut and the surface as well as thread friction will not affect. Since materials can be used optimally, and it is possible to use bolts with a smaller diameter or else to apply a greater preload to achieve a higher degree of safety.

By using several such devices in sequence or in parallel produces a considerable saving on time, as well as achieving advantage of simultaneous apply exactly equal forces when applying the hydraulic pressure to stretch the bolts.

3.3.1 Tensioning Procedure

Hydraulic bolt tensioning essentially provides a hydraulic load that acts directly upon the stud bolt to retain its specified residual preload when the hydraulic pressure in the tensioners is released.

The tightening by hydraulic bolt tensioning can be summarized in figure 3.6 and in seven steps as below;

1. Placing nut driver on nut.
2. Assemble bridge over the structure; position the bridge window so that access to either the nut runner or hexagon nut is comfortably achieved in order to tighten with tommy bar.
3. Assemble the hollow hydraulic jack over the bridge; position the hydraulic connections so that access is comfortably achieved.
4. Assembling the stud pullers into bolt tensioner screwing onto the threads protruding above the hexagon nuts. Ensure the stud pullers are threaded to the same diameter, thread form and pitch as the bolts to be tensioned. Use the spanner supplied with the tensioning equipment to fully screw down the inserts, until contact is made with the top face of the hydraulic jack. Connect a link hose from the pump unit to the hydraulic jack.

5. Close pressure let down valve on pump unit, then using hand pump, pressurize the system to the required pressure. When the required pressure is reached in system, stop the pump (hold pressure). At this stage the bolts will be initially loaded with the load being held by the tensioners system.
6. Using the tommy bar, rotate the hexagon nuts, through the bridge access windows
7. Slowly release the pressure from the system by open the valve on pump.

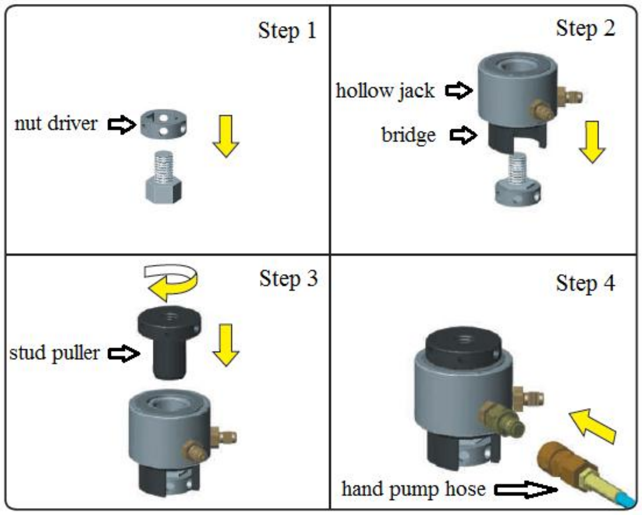


Figure 3.6 General hydraulic tightening steps

The load is now transferred from the tensioners to the nuts, it is recommended to repeat steps six more than one time, if the desired preload has not yet been reached as shown in figure 3.7 Show the total arrangement of hydraulic bolt tensioning system that used in this study.

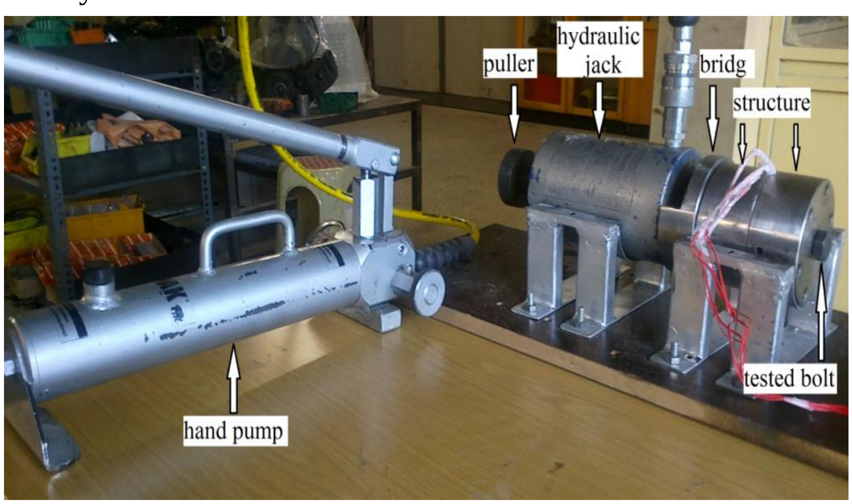


Figure 3.7 Arrangement of hydraulic bolt tensioning system

3.3.2 Hydraulic Pressure Calculation

Designed bolt preload are easily achieved by applying a predetermined hydraulic force to a calculated surface area. This load is transferred to the bolt which will stretch along its axis and the resulting preload will be directly proportional to the hydraulic pressure applied by the jack and therefore known. As a result of the bolt stretching under the hydraulic load and dependent on the type of our structure, a gap will appear under either the hex nut, As the hydraulic load is released the hex nut will maintain the applied load.

The same magnitude of preload that used in torque tightening method is going to be calculating in hydraulic tensioning method.

$$Fi = 0.85 \times At \times Sp \qquad (3.11)$$

In above formula, calculating bolt residual preload, residual preload can be calculated from known stress or bolt stretch requirement, depending on relation between force and pressure in formula 3.12 the magnitude of hydraulic oil pressure can be get from Appendix B. depending on preload as reference.

$$P = \frac{Fi}{A} \qquad (3.12)$$

Where

P = hydraulic pressure

Fi = preload

A = ram area of jack

The load loss allowance ratio of tensioning force to bolt preload should be calculate as below formula

$$P\,h = P * load\ loss\ ratio \qquad (3.13)$$

Where

Ph = final hydraulic pressure

P = theoretical pressure

. 3.3.3 The Load Loss Allowance

It is clear that the greater the difference in stiffness between the tightening bolt and the structure, the smaller the fraction of the effect external load of the assembly which affects the bolt. Therefore, it is better to use bolts with a high length to diameter ratio (L/d), for stiffness is lowest in this case. Furthermore, the load loss allowance factor is also the main parameter determining the ratio between the theoretical applied hydraulic pressure P during tightening, and the final hydraulic pressure Ph that lead to stretch the bolt and obtain the target preload, as shown in figure 3.8

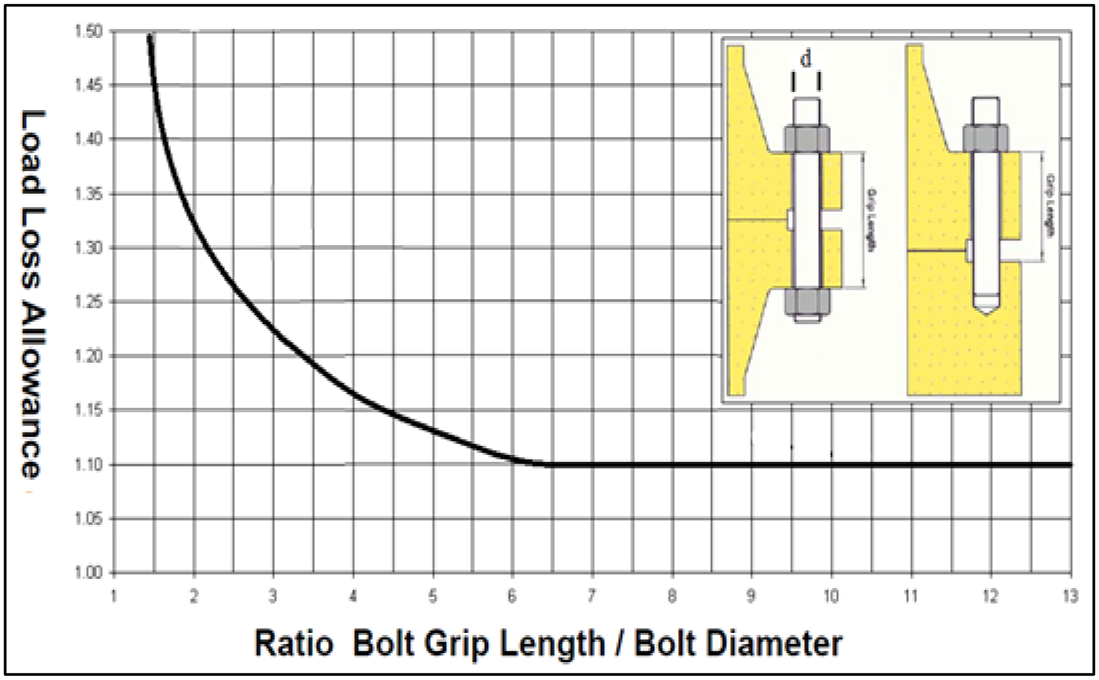


Figure 3.8 Load loss allowance graph [24]

3.4 Measuring Preload and Torsion Magnitude by Strain Gauge

A useful method of finding surface stress in a bolt during tightening and after tightening is to measure the local strain or deformation. There are various methods whereby these can be found. Often foil strain gauge models are made of a scaled wiring grid member so that deformations in attached body can be measured by strain gauge grid length change. The most common type of strain gauge used is based on electrical resistance.

3.4.1 Strain Gauge Principal

When strain is generated in a bolt and strain gage attached, the strain is relayed via the gauge base to resistance wire or foil in the gage base. As a result, the fine wire experiences a variation in electrical resistance change, the variation is exactly proportional to the strain, equation 3.12. Is relation between resistance change and strain using data acquisition to convert the signal output from change in resistance directly to strain.

$$K = \frac{\frac{\Delta R}{R}}{\varepsilon} \tag{3.14}$$

Where

$K =$ gauge factor as shown on package

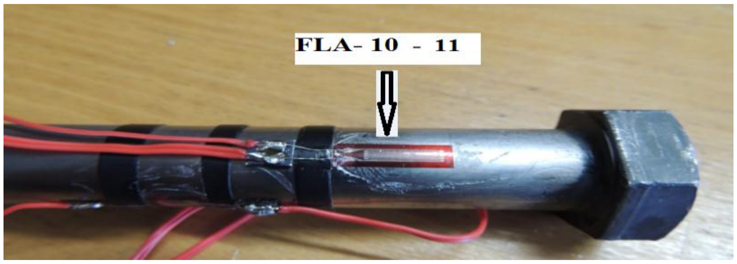
$\varepsilon =$ strain

$\Delta\, R =$ Resistance change due to strain

$R =$ Gauge resistance

Two kind of strain gages used in the study to measure axial strain and torsion strain. FLA strain gauge series used for axial strain and FCT series used for torsion strain, two kind of strain gauge as shown in figure 3.9

Axial strain gauge



Torsion strain gauge

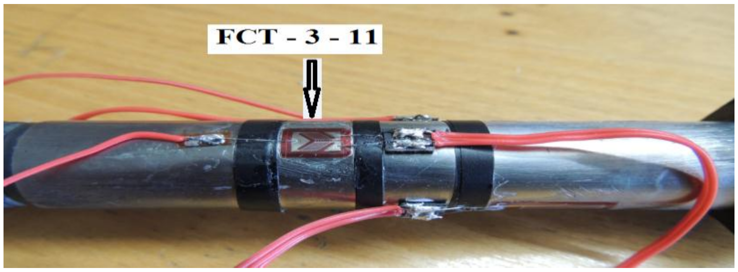


Figure 3.9 FLA and FCT types strain gauges

System 7000 from Micro-Measurements used in study as shown in figure 3.10 , precision data acquisition instrument system intended for static and dynamic tests and measurement applications, system includes a scanner with 8 channels of data acquisition. Each channel can be configured, via software, to accept output signals from strain gages. The Wheatstone bridge balanced and calibrate in software mathematically.

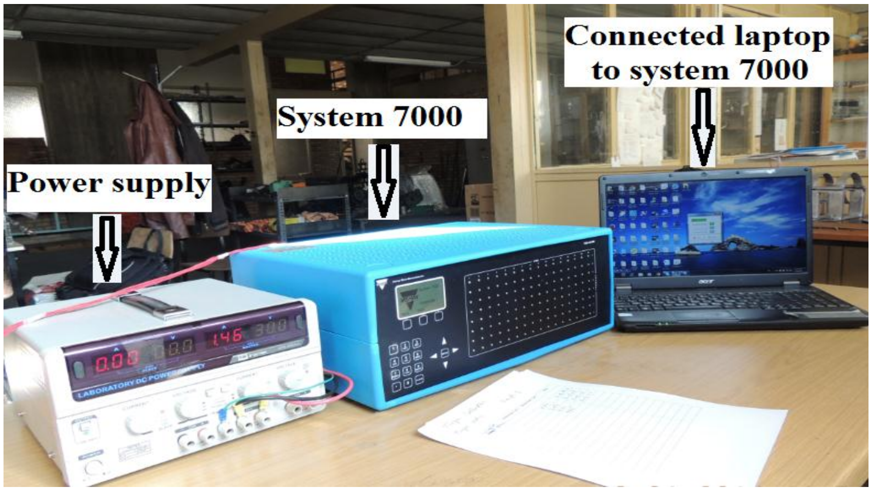


Figure 3.10 Complete systems 7000

3.4.2 Strain Gauge Installation

The test bolt surface must be prepared before the strain gauges install and bondable terminal pads can be installed. Cleanliness is important for successful strain gage installation. The object of preparation surface of bolt is to create a smooth surface which can be wetted so it can receive the adhesive. Figure 3.11 shows the installation requirements of strain gauges.

Bellow general steps of installation strain gauge on bolt;

1- Strain gauge surface preparation.
    - Degreasing
    - Abrading
    - Burnishing of layout lines
    - Conditioning
    - Neutralizing
2- Strain gauge bonding
3- Soldering and coating

Figure 3.11 Installation requirements of strain gauges.

3.4.3 Configuration of Strain Gauge on Bolt

In each bolt three strain gauge arranged to measure axial strain and torsion strain, two gauges, R1 and R2, (FLA types) have their axes parallel to the axes of the bolt, as shown in figure 3.12 These are sensing bolt preload and would produce a satisfactory electrical output with load that generate during tightening, the gauges R1 and R2 are placed diametrically opposite each other on the surface of the bolt, both R1 and R2 connected quarter bridge and connected to data acquisition system by RJ45 connector. Also because of arrangement R1 and R2 diametrically opposite, the effect of slight bending stress will not effect, if there is bending, one of the gauges will be under compression and other one will be under tension, so the result in each strain gauge will cancel each other.

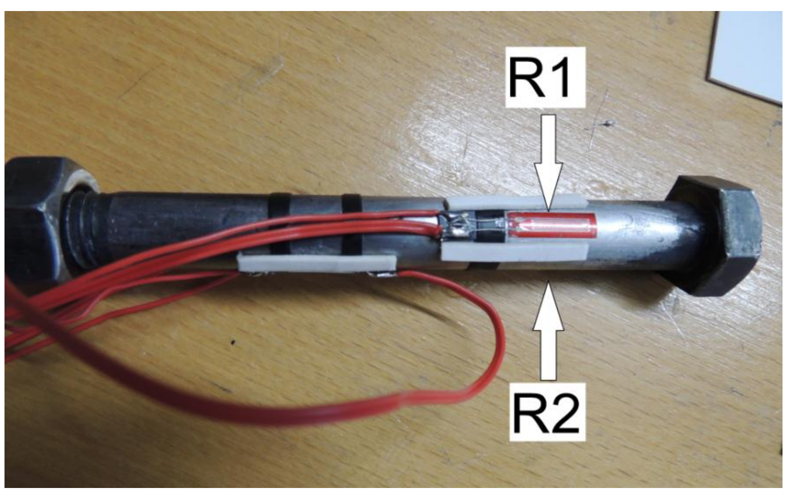


Figure 3.12 Axial strain gauges R1 and R2

To measure torsional magnitude during tightening and after tightening, combine strain gauge R3 (FCT series) used as shown in figures 3.13, the combine strain gauge consist of two strain gauges, the first strain gauge perpendicular to other one in arrangement, so when strain gauge R3 mount parallel to the axes of the bolt, as result of this mount, the two strain gauges finally mounted at the axes 45°with bolt axis, which lead to read maximum strain on bolt surface. The two strain gauges connected in half bridge with data acquisition system, so the effect of bending cancellation by this connection, also in the same time during calibration of torsion strain gauges bending force applied to bolt and there were no effects of this bending force to torsion strain gauge readings.

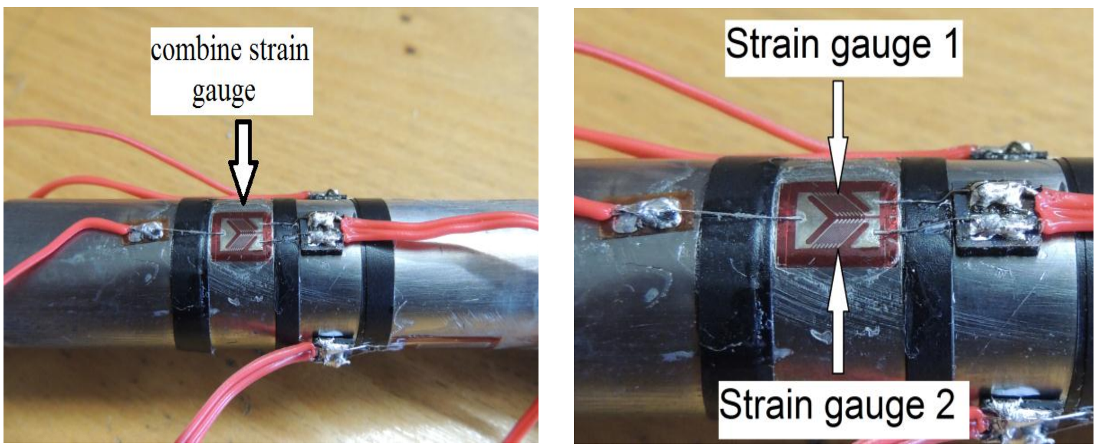


Figure3.13 Combine strain gauge

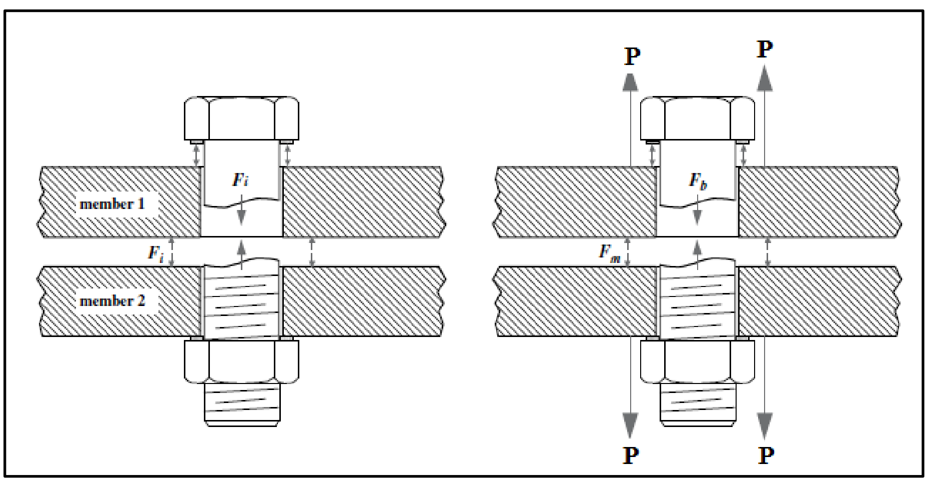
3.4.4 Temperature Compensation in the Strain Gauge

Adequate temperature compensation is necessary for accurate measurement of strain gauge in static test, the effect of temperature compensation identified when test be done in laboratories, the effect of temperature change on strain gauge reading appear in tests over indefinitely long period of time when proper installation are made, for FLA strain gauge series, which used in this study to measure axial strain, all our measurements of stains are made in short time as well as it held in constant temperature laboratories.

The same above temperature effect conditions in measuring torsion strain with FCT strain gauge series achieved, as well as one of strain gauges that used in measuring torsion strain gauge are going to be as a dummy gauge in the half bridge, so the temperature effect compensates obtain naturally.

3.5 Preload Bolt Saperation Criteria

Separation of a preloaded in joint must not occur below the joint separation load. The equations presented in this section are based on simple linear preloaded joint theory. For more accurate, detailed analysis may be used instead of the simple linear equations to determine if joint separation occurs in structure. The analysis, however, must use the proper preload magnitude (Fi) and maximum expected joint separation load (P total). If the bolt is loaded above its yield allowable at the joint separation load, the analysis must also be nonlinear. Based on this analysis, to be in safe site, joint separation must not occur below the joint separation load. The criteria states that joint separation must not occur below the joint separation load. This designer requirement may not always be realistic. For hardware such as that which is part of maintains hazardous material and/or pressure system, complex preloaded structure joint/seal designs are generally used. To prevent leakage in such a design, every structural element should be treated as critical situation. Therefore, the joint/seal/fastener system must be analyzed and the interaction of the individual components accounted. For these kind cases the system must demonstrate a separation safety factor, while using the minimum magnitude of preload (Fi) on all fasteners, figure 3.14 shows simple single bolted joint in tension.



Preload stage External load stage

Figure 3.14 Simple single bolted joint in tension

From the basic definition of stiffness and the principles of equilibrium and compatibility in bolted assembly structure, the following relations for the bolt force, Fb, and member force Fm, as shown in figure 3.15 , can be obtained:

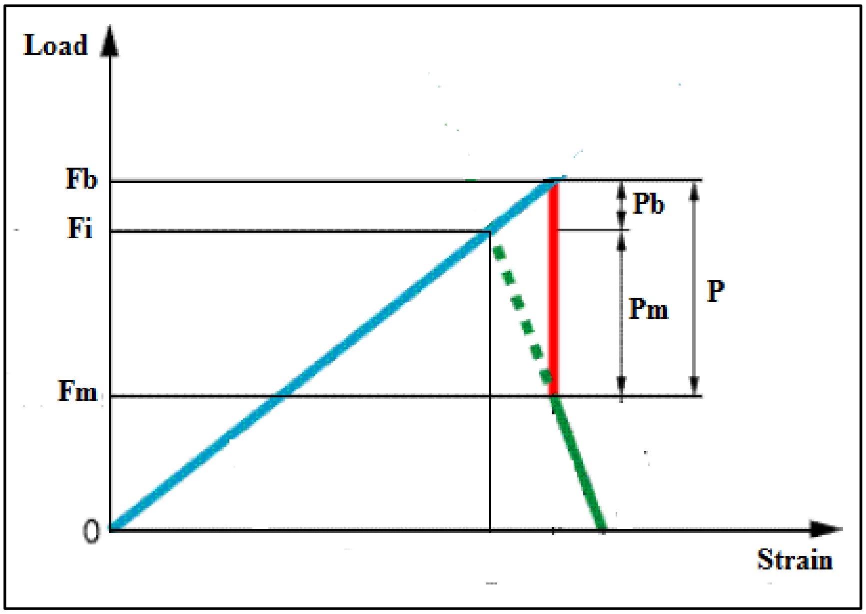


Figure 3.15 Diagram showing the effect of an external load P on a bolted assembly with preload Fi

$$Fb = CP + Fi = Pb + Fi \qquad (3.15)$$

$$Fm = (1 - C)P - Fi = Pm - Fi \qquad (3.16)$$

$$C = \frac{Kb}{Kb+Km} \qquad (3.17)$$

Where:

Fi = preload

P total = Total external tensile load applied to the joint

P = external tensile load per bolt

Pb = portion of P taken by bolt

Pm = portion of P taken by members

Fb = resultant bolt load

Fm = resultant load on members

C = fraction of external load P carried by bolt

1 − C = fraction of external load P carried by members

Kb = bolt stiffnesss

Km = member stiffness

3.6 Preload Relaxation

When preload are first applied to bolts in a joint by any kind of tightening, local yielding takes place due to excess bearing under nut and bolt heads, rough surface finish, local high spots, and lack of perfect, squareness of bolt and nut bearing surface. Also, applied loads are not distributed evenly on each thread in both nut and bolt. Thread deformation may, therefore, occur to redistribute load more evenly on the threaded parts, all this factors replace to redistribute load more evenly on the threaded parts, so all this factors results in a loss of preload within period of time (minutes, hours to days). As a general practice, an allowance for more than 10% loss of preload should be made when designing a joint. An alternative to require retightening several minutes to several days from initial bolt load applied. For critical threaded joints, simulations may be used to determine the extent of preload loss and remedial action required.

For longer period of time, preload may be reduced or completely lost due to external effects, vibration, creep, temperature cycling (including ambient temperature changes), joint load, etc. An increasing in joint bolt preload or, for transvers vibration, it is recommended to use of thread locking method which prevents relative motion within the joint, also may avoid preload relaxation problems caused by vibration and temperature cycling, creeps is generally a high temperature change problem, although some loss in preload can be expected even at normal temperature. Harder materials in joint and material resistance to creep at the temperature developed within the joint must be considered if creep causes a problem. In any case, good joint design and tightening method will minimize preload relaxation.

A comparison of various bolt tightening methods available today in terms of their preload accuracy and relative cost is shown in the figure 3.16. It is obvious that the cost of the tightening system goes up with the increasing accuracy of tightening.

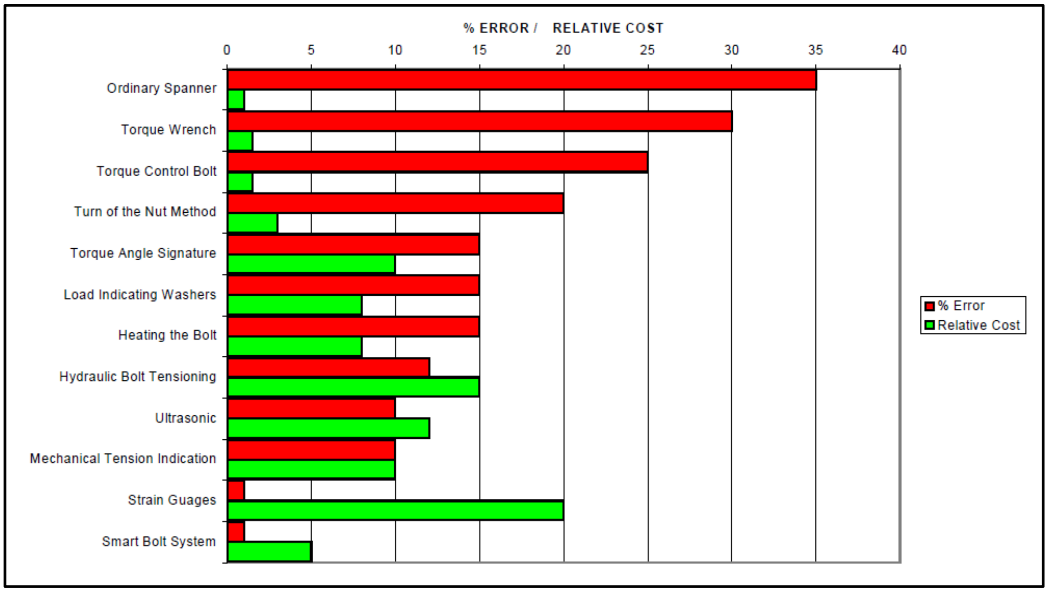


Figure 3.16 Comparison of error and relative cost between tightening methods [2]

3.7 Cyclic / Fatigue Loading Effect on Preload

Quite a while ago, engineers discovered that repeatedly applied and then removed a nominal load to and from a metal part (known as cyclic load), as shown in figure 3.17 the part would break after a certain number of load-unload cycles; even when the magnitude of maximum cyclic stress level applied was much bellow than the ultimate tensile stress, and in fact, much lower than the yield stress. By researches, the engineers discovered as they reduced the magnitude of cyclic stress, the part would sustain more before breaking. This behavior of part became known as "Fatigue" because it was originally thought that the metal got "tired"

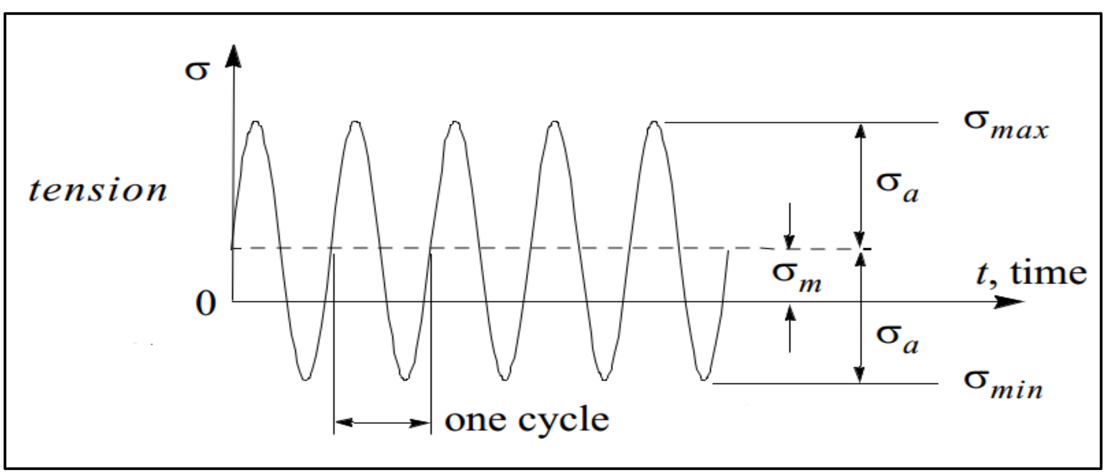


Figure 3.17 Cyclic stressing

In summary, when a ductile metal is loaded so that the load magnitude is gradually increased from zero to a max, final rupture of the material is produced by very large strains effect. However, if the same material is subjected to external repeated loads, failure in metal may occur as a result of stresses much lower than the elastic limit and there will be no plastic deformation in the region of the fractures. Most often in the metal behavior, theirs is usually no prior indication of impending failure. Both tensile and compressive effect stresses can lead to fatigue damage.

The process of fatigue failure consists of three (3) stages:

1. Initial fatigue damage loading to crack nucleation and crack initiation.
2. Progressive cyclic growth of a crack propagation until the remaining un-cracked cross-section of the part becomes too weak to sustain the external loads imposed.
3. Finally, in part sudden fracture of the remaining cross-section.

3.7.1 Different Types of Fatigue / Cyclic Loading

There are three (3) different types of fatigue loading:

1. Zero-to-max-to-zero: Where a part which is carrying no load is then subjected to an external load, and later, the external load is removed, so the first part goes back to no-load condition.
2. Varying loads superimposed on a constant load: the part initially loaded then external cyclic load added to a part
3. Fully-reversing load: Once cycle of this type of loading occurs in part when a tensile stress of some value is applied to an unloaded part and then released, then a compressive stress of the same value is applied and released.

The second type of fatigue loading is used to supply external load to the bolt, so the final stress of bolt will be fluctuating between yield point and pure preload magnitudes by using Vibrophores machine manufactured by ZWICK Testing Machine Ltd Company, as shown in figure 3.18 used to determine preload behavior under tension cyclic load.

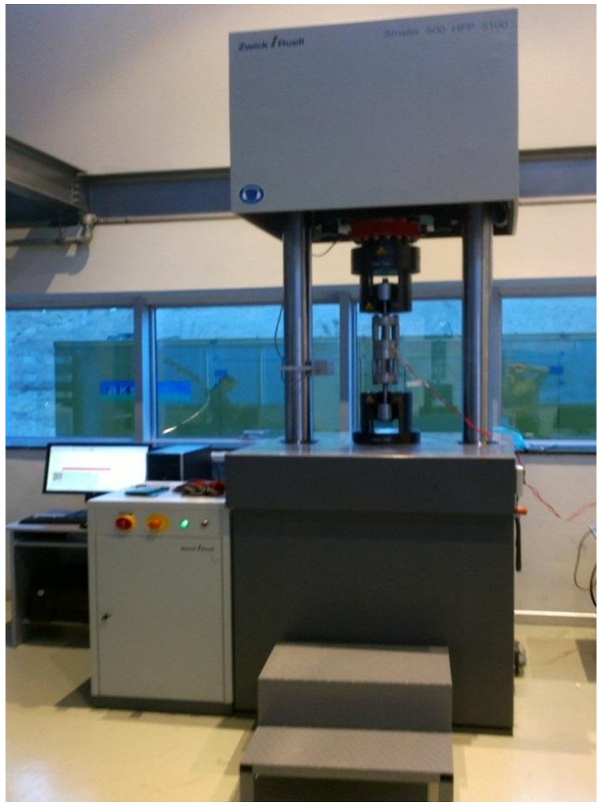


Figure 3.18 Fatigue load machine from ZWICK Company

CHAPTER 4

EXPERIMENTAL WORKS ON PRELOADING OF BOLTED CONNECTIONS

4.1 Introduction

The objective of this chapter is to clarify the bolts geometries and the property class of bolts that used in this study and the single bolt cylinder which is used as structure in both tightening methods in static and dynamic test. Description of tightening tools in both tightening methods was explained.

In the second part of this chapter is giving the methodologies for calculating target preload for M14, M16 and M18. Torque magnitudes and hydraulic pressure magnitude calculations in both tightening methods cleared to reach target preload, also the chapter cover the external dynamic load magnitude in axial direction parallel to the bolt axis for M14 and the stress range fluctuation during cyclic load effect.

4.2 Parameters of Tested Bolts

The behavior of any structural component is highly dependent on the properties of the material which it is produced; bolts as product are no exception. A threaded fastener's grade indicates its tensile strength and hardness, the higher the clamping load required, the higher the grade need to be use.

During the study and comparison between the two types of tightening, A total of 24 bolts, consisting of 8.8 grades and three different diameters (M14, M16 and M18) as shown in figure 4.1, (150 mm) in lengths as shown in figure 4.2, were tested in both static and dynamic investigations.

Figure 4.1 diffrent diameter of tested bolts

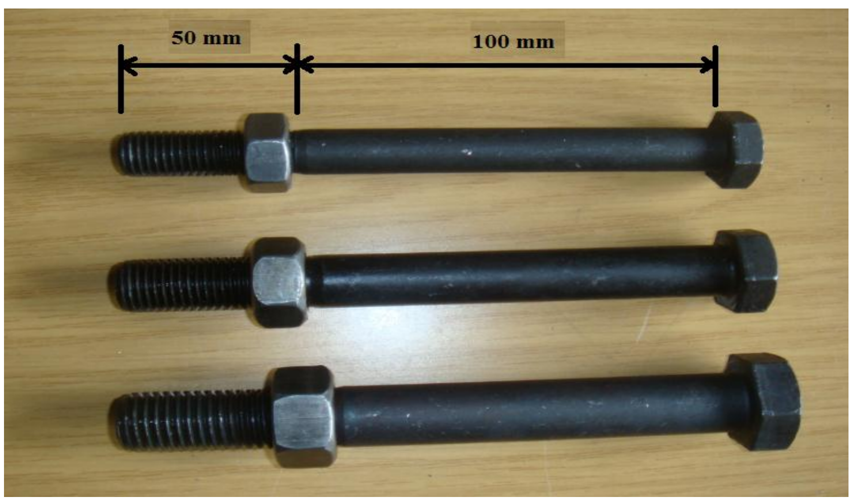


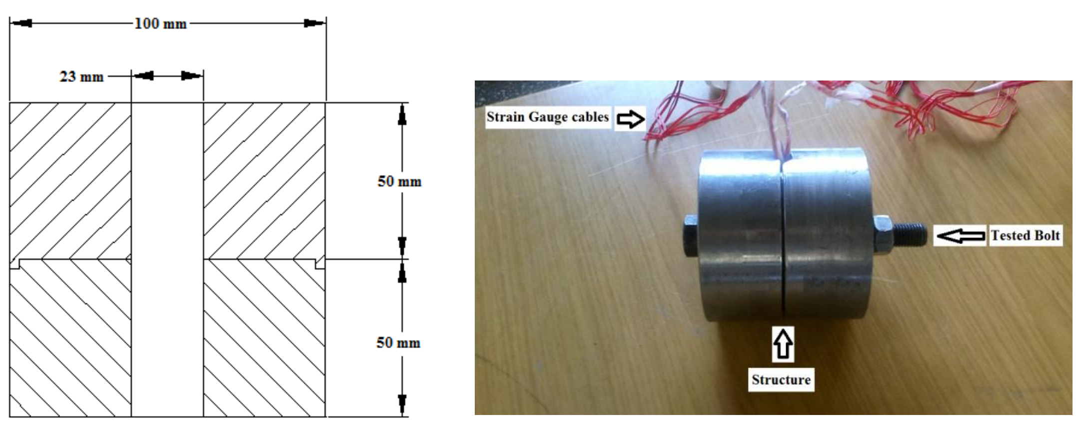
Figure 4.2 Length of tested bolts

Grade 8.8 bolts are usually from the manufacturer as hexagon head bolts, and often galvanized, and sometimes zinc plated, the universal color of grade 8.8 is black. Generally, in the head of the bolts grade 8.8 there are three radial lines or the numerals 8.8 stamped on the head, these two signs used to distinguish. Table 4.1 shows details and mechanical properties of grade 8.8.

Table 4.1 Description and mechanical properties of grade 8.8

Material	Class 8.8 bolts can be made from carbon steel which conforms to the following chemical composition-- Carbon: 0.25-0.55%; Phosphorus: 0.035% maximum; Sulfur: 0.035% maximum.
Heat Treatment	Class 8.8 bolts shall be heat treated by quenching in a liquid medium from above the transformation temperature and reheating to a tempering temperature of 425°C
Core Hardness	For diameters less than or equal to 16mm: Rockwell C22 - 32 (Vickers HV 250 - 320) for diameters greater than 16mm: Rockwell C23 - 34 (Vickers HV 255 - 335)
Surface Hardness	Shall not be more than 30 Vickers points above the measured core hardness of the product
Tensile Strength	For diameters less than or equal to 16mm: 800 N/mm2 minimum For diameters greater than 16mm: 830 N/mm2 minimum
Elongation	12% minimum

All tested bolts in both tightening methods, tightened to assembly structure consist of two cylinder parts with a hole in the center as shown in figure 4.3. The material of major structure is steel grade ST 52 and manufactured by turning on lath machine.



(a) (b)

Figure 4.3 Major structure (a) technical drawing, (b) structure photo

4.3 Description of Tightening Tools in Study

Using a torque wrench a known amount of torque will be applied to the bolts and as a result the target preload will achieve. Because a torque wrench is considered a measuring tool, it must be properly calibrated and maintained on a preventative maintenance and calibration schedule, during torque tightening method in this study a new LICOTA torque tightening tool as shown in figure 4.4 used for tightening the bolts , Appendix A is the torque wrench certificate calibration provided by the manufacturer company.

Figure 4.4 LICOTA torque tightening tool

For Hydraulic tensioning method, target preload is easily achieved by applying a predetermined hydraulic force, this hydraulic force is transferred to the bolt which will stretch along its axis and the resulting preload will be directly proportional to the hydraulic pressure applied and therefore known. In order to supply optimum stretch bolt preload, hydraulic hollow jack, 30 ton CFTMAK type, 700 bar used, as shown in figure 4.5, the system consists of a hollow jack (108 mm out diameter, 40 mm inner hollow diameter and 215 mm height), and the second part is hand pump.

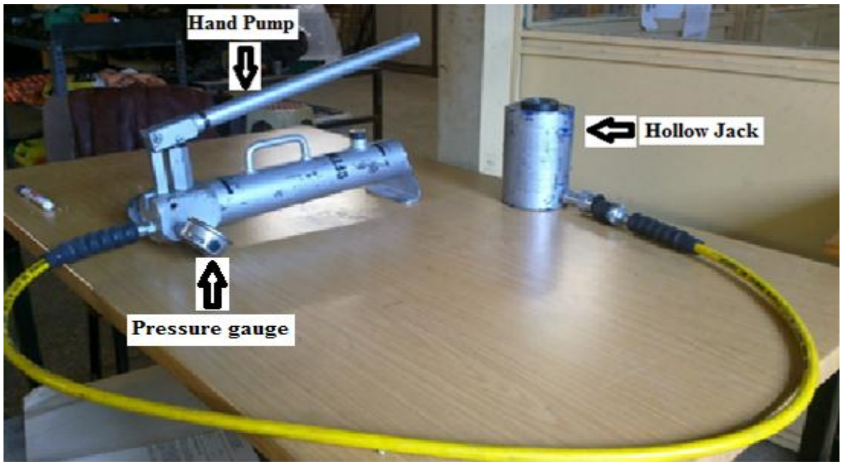


Figure 4.5 Hydraulic pump-jack arrangement used in the bolt tensioning system

The major factor of preload accuracy in the hydraulic tightening system depends on the hydraulic pressure that generate by using a hand pump, digital pressure gauge as shown in figure 4.6 used to measure accurate pressure in the system during tightening and after tightening.

Figure 4.6 Image of digital pressure gauge

4.4 Hydraulic Jack Calibration

Before a hydraulic hollow jack system can be used, calibration parameters need to be accurately determined. These calibration parameters can only be determined by simulating the hydraulic system under load.

The calibration procedure performed in mechanical engineering laboratories by using universal testing machine as shown in figure 4.7 to supply static load.

Figure 4.7 universal testing machine

During calibration method hydraulic hollow jack placed in the working area of the testing machine, and force is then generated by forcing hydraulic fluid using hand pump into the jack cavity. The calibration procedure yields data points between hydraulic pressures and load shown in table 4.2. Table 4.2 Hydraulic jack calibration results

Pressure (bar)	Weight (kg)
50	2202
75	3306
100	4409
120	5295
140	6178
160	7061
180	7945
200	8829
220	9712
240	10593
260	11478
280	12365
300	13244

4.5 Torsion Strain Gauges Calibration

This calibration provides information regarding the torque measurements using FCT type strain gages, and theoretical torque measurements. The intent is to provide a better understanding of how to obtain dependable torque measurements by measuring torque via strain gages, also to check the effect of bolt bending if appear on strain gauge reading.

In theoretical torque calculations, the torque T is caused by the weight of the mass hanging at the lever of the length1 m, a nut welded to the end of the lever and tightened to the M14 bolt, second nut tightened to fixed bolt on the lever and prevent rotation between bolt and lever, the head of bolt clamped by using a vise. Figure 4.8 is schematic of torque calibration, and figure 4.9 shows the all components of calibration test.

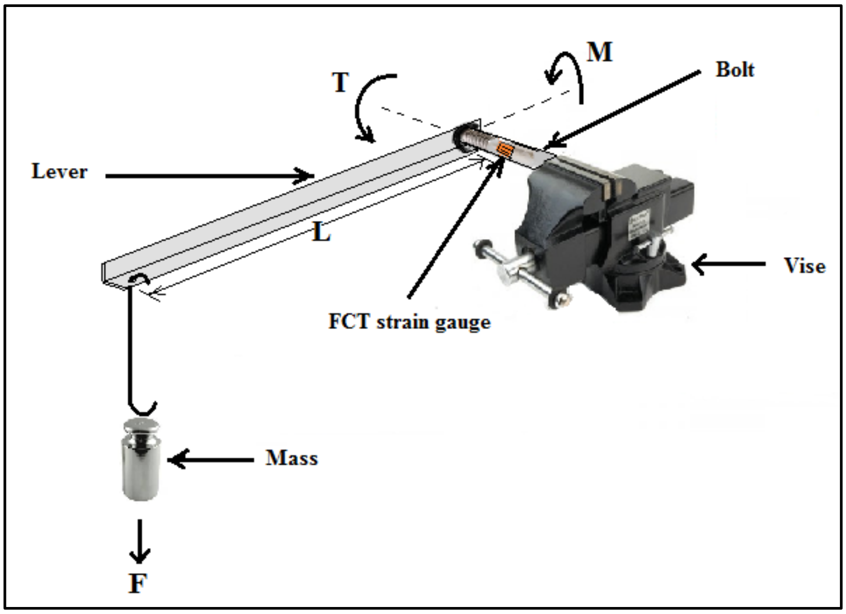


Figure 4.8 schematic of torque calibration.

Known value of the masses hangs on the end of the lever, so the theoretical torque is computed from mass values according to the below formula.

$$T = F * L \qquad (4.1)$$

Where

T = Torque

F = force (weight of mass)

L = arm distance

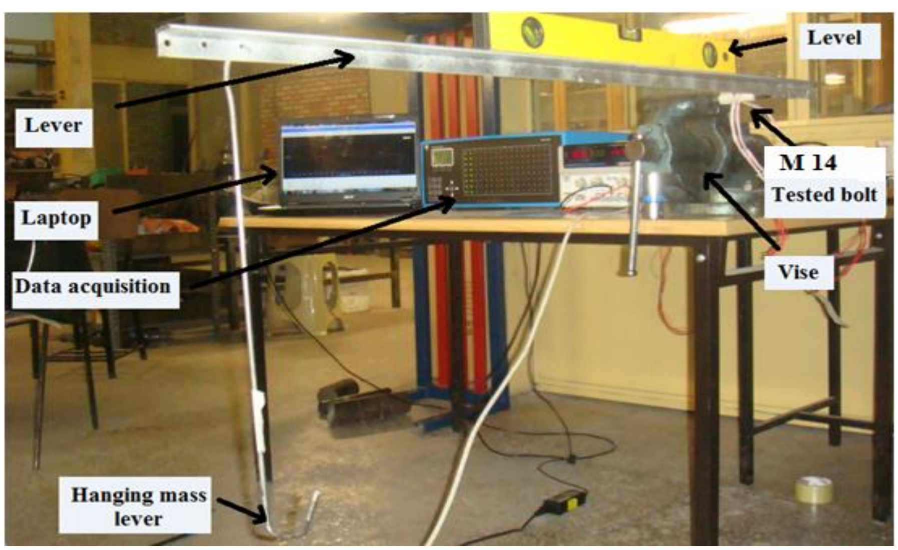


Figure 4.9 Torsion calibration test components

The output of FCT series torsion stain gauges used to measure the torque magnitudes while masses hanged on the lever according to below equations.

$$\gamma = 2 * \in \tag{4.2}$$

Where

γ = shear strain

$\in$ = strain reading of strain gauge at 45° angle of bolt axis

$$G = \frac{\tau}{\gamma} \tag{4.3}$$

Where

$G =$ modulus of shear strain

$\tau =$ shear stress

$$T = \frac{2\tau J}{d} \tag{4.4}$$

Where

$T =$ torque

$J =$ polar moment of inertia

d = bolt diameter

Comparison between torque results in both methods, theoretical calculations and torque obtained from strain gauge reading, did not exceed 1%. Likewise during the calibration test, table 4.3 shows torque magnitudes as a results of hanging different weights on end of arm for M14 bolt,the torque calculated by two methods, first calculated by theoretical formula (4.1) and second calculated from strain gauge reading using formulas (4.2,4.3 and 4.4).

Table 4.3 torque magnitudes calculated by different methods

Weight (Kg)	Torque magnitude calculated by theoreticaly (N.m)	Torque magnitude calculated by strain gauges (N.m)
6.230	42.40	42.46
8.561	55.32	55.31
9.984	63.20	63.11

Effect of bolt bending on torsion strain gauge readings studied. To demonstrate that half bridge arrangement of torsion FCT strain gauges compensates bending. A bending force applied perpendicular to the end of the bolt (head of bolt clamped by vise), because both strain gauges will sense the bending effect with equal strain, but opposite sign, one of the strain gauges will be under compression and the other will be under tension effect, because of opposite sign the compensate in half bridge obtained. This fact was clear when applying bending moment on bolt structure, no changes in strain gauge reading during moment effect.

4.6 Target Preload

A bolt used in any situation where the bolt is under tension to hold two or more components together must be tensioned correctly, so the bolt in this situation must have the proper preload applied magnitude, after tightening complete, if the bolt preload magnitude below target preload value with wide variation the nut may come loose, also if preload magnitude tightened above target preload with wide variation, equivalent stress will become closer to yield point, any additional external load stress will lead to bolt failure.

In the first part of this study, in static tests of bolts with no external loading, however, two levels of static preload (85% and 69% of the yield load) are applied to force the limits near yield and proof loads. For M14 and M16 bolts a preload level of 85% of yield strength was targeted. Whereas, approximatly 69% of yield strength was used as the preload level for M18 bolts. As an example, for M14 size bolts, with a yield strength of 660MPa, bolt tensile area of 122.65 mm2 and a target preloading stress reaching nearly 85% of yield strength, table 4.2 below shows target preload magnitudes.

Table 4.4 Static test preload magnitude for grade 8.8

Bolt Size	Bolt Grade	Tensile Stress Area (mm²)	Yield strength (Mpa)	Fy (N)	Target Preload (N)	Fi / Fy
M14	8.8	122.65	660	80916	68810	85%
M16	8.8	165.04	660	108926	92591	85%
M18	8.8	213.71	660	141048	97222	69%

4.7 Torques to Preload Calculations

The use of torque has long been associated with methods of tightening and inspecting bolts. The object of tightening bolts is to achieve a minimum tension (preload), not torque, generally in the bolts. There is a relationship between torque and tension (preload), but it is highly variable and must be used with care and caution. The equation to estimate the torque (T) from a required preload (Fi) is given in equation (3.4) as:

$$T = K * d * Fi \tag{3.4}$$

The variables in the torque-preload relationship (3.4) include lubrication, thread fit, the nut / bolt surface contacting with the washer or structure surface makes properties the nut factor K and the target preload in the bolt, as well as bolt diameter. Here it is the nut factor which causes the most uncertain in torque magnitude and nut factor varies from application to application with average between 0.16 ~ 0.26 usually.

In static part, M14, M16 and M18 bolts preload magnitudes calculated according to equation (3.4), for example the target preload Fi for M14 is 68810N and the required torque (T) with an average nut factor of 0.2, is nearly 173.0N.m, and for M16 target preload Fi is 92591N and required torque is nearly 270 N.m. While for M18 target preload Fi is 97222 N and required torque is 350N.m.

4.8 Hydraulic Pressures to Preload Calculations

In this study hydraulic bolt tensioners use a hollow hydraulic jack placed around the screw, stretching it axially. This hydraulic force/load, by means of a suitable mechanism, can be transferred to the snug-tight bolt body stretching it along its axis “with no parasite torsion”. The resulting preload on the bolt body is then directly proportional to the hydraulic pressure applied by the jack. An experimental setup constructed to apply hydraulic load to the bolted connection as seen in figure 4.10.and figure 4.11 is a general view of a hydraulic tensioning test system.

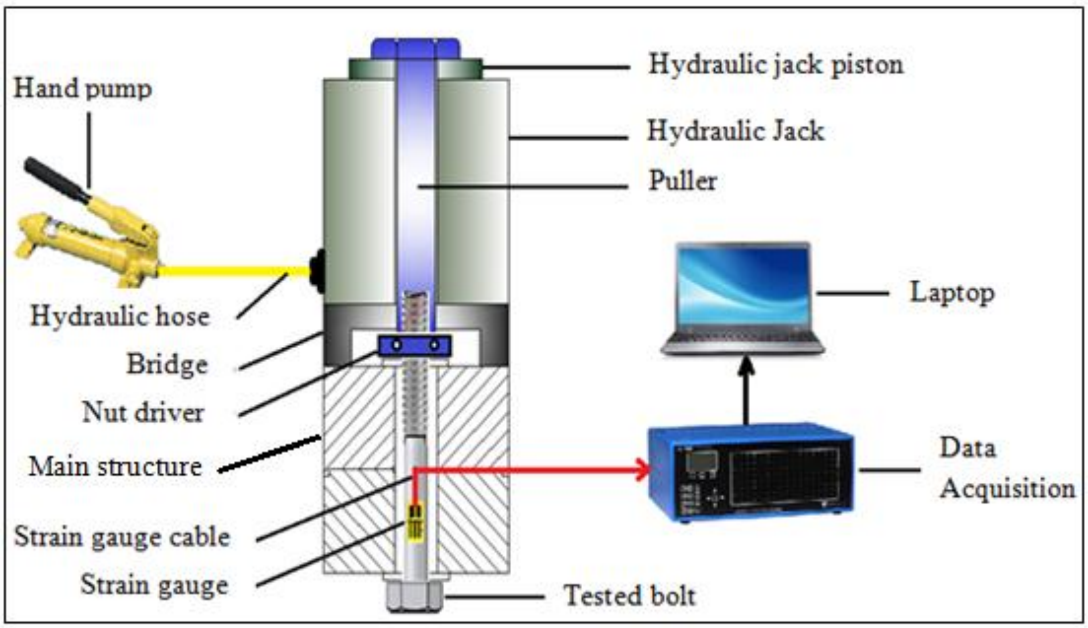


Figure 4.10 schamatic drawing of the test set up

Figure 4.11 General view of all components of hydraulic tensioning test system

The same magnitude of preloads given in table 4.2 is used to calculate the required hydraulic pressures for three different bolt sizes. A CFTMAK type hydraulic hollow jack, having a pressure area of 42.857cm2 and providing 30 tons at 700 bar is used during tests.

Table 4.4 below shows preload and hydraulic pressure values used for M14, M16 and M18 bolts for static preloading only. For example the target preload Fi for M14 is 68810N and the required hydraulic pressure after multiplying it by load loss allowance factor, according to Appendix B, is nearly 171 bars.

Table 4.5 Preload vs Hydraulic pressures with (load loss allowance factor =1.1)

Bolt Size	**Target Prelaod (N)**	**Hydraulic Pressure (bar)**
M14	68810	171
M16	92591	230
M18	97222	242

4.9 Cyclic / Fatigue Loading Calculations

One of the most common failure mechanisms for bolts is fatigue. Fatigue is the phenomenon that occurs in the bolts as result of cyclic variation of the applied stress. Fatigue failure of the bolts is often found in the first engaged nut and bolt threads, which have the highest stress or at the head to shank fillet radius as shown in figure 4.12.

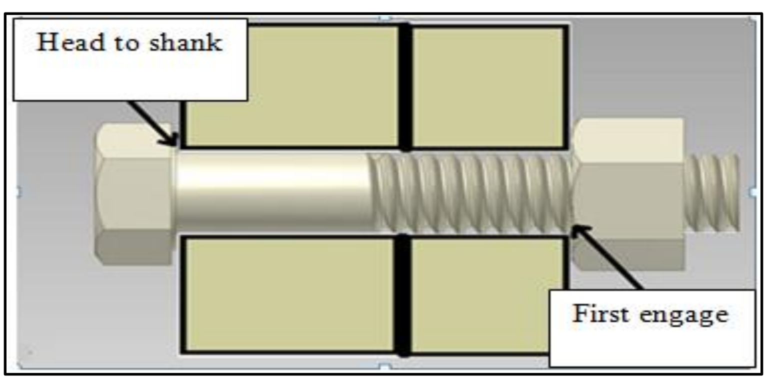


Figure 4.12 Common locations of fatigue crack initiation in a bolt

A cyclic fatigue loading (in axial direction parallel to the bolt axis) applies to bolted connection (via clamped parts) for both methods of torque wrench and hydraulic tension. Fatigue testing machine of ZWICK-Vibrophores seen in figure 4.13 is used to apply cyclic fatigue loading. During cyclic load application on the bolted connection it is so arranged that the final likely load on the bolt creates a stress not more than the yield strength of the bolt. So the final stress of the bolt, at worst, will be fluctuating between pure preload stress and the yield point as seen in figure 4.14

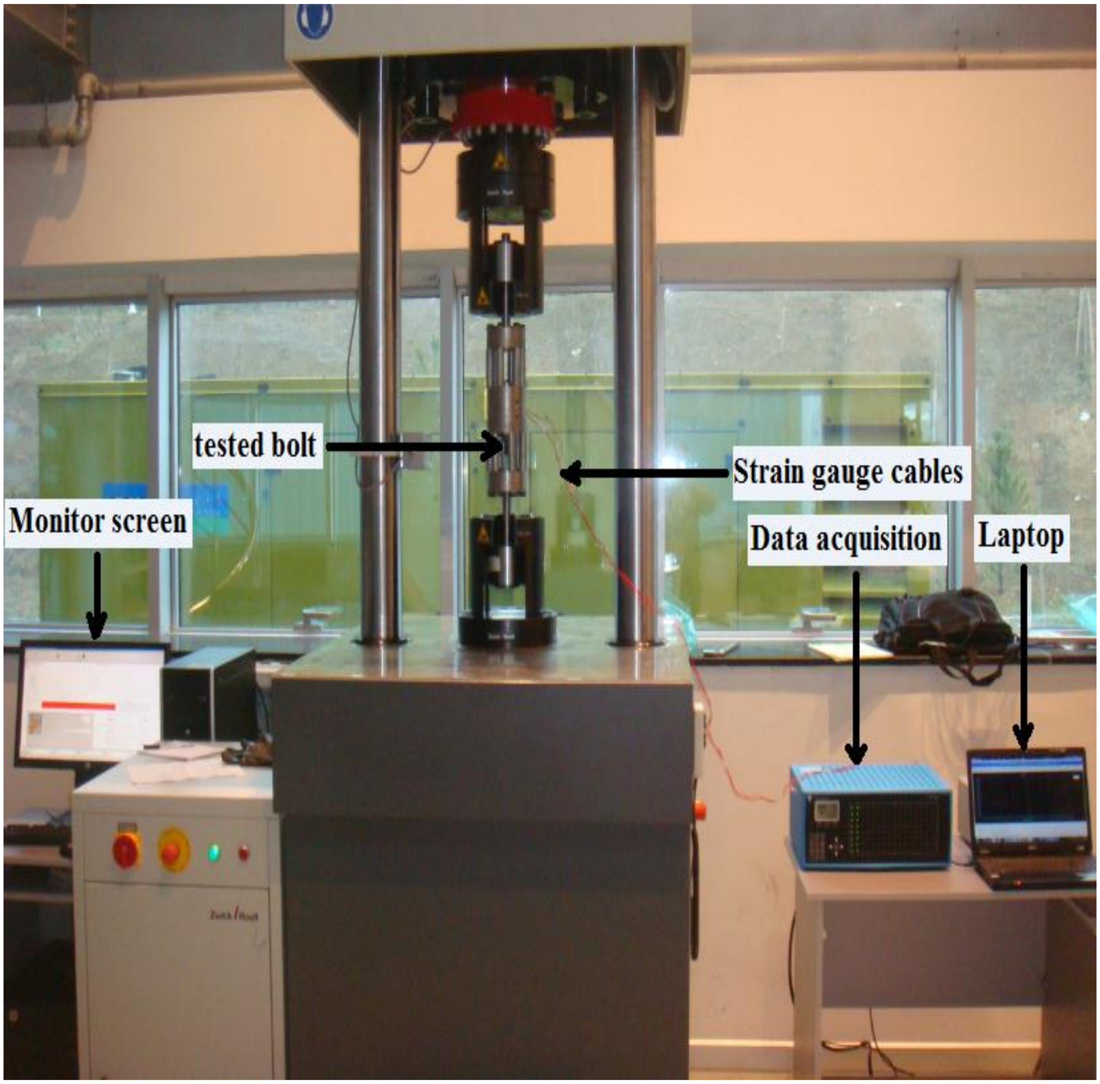


Figure 4.13 Fatigue loading machine (ZWICK / Roell Model)

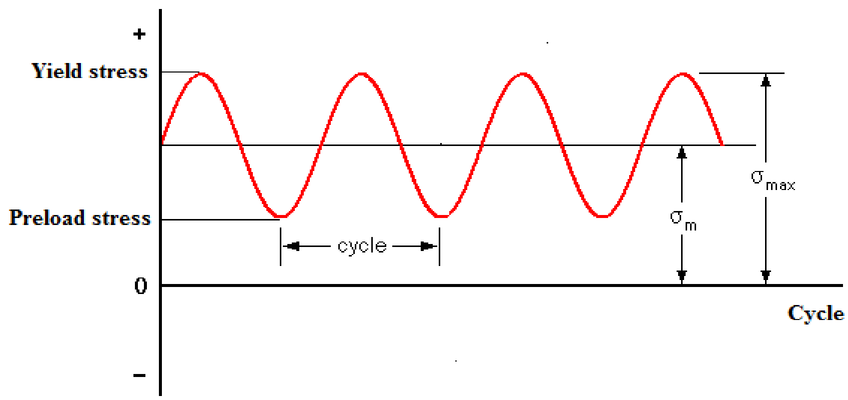


Figure4.14 Estimated bolt stress under dynamic loads

A structure of a bolted connection as seen in figure 4.15 is used during dynamic loading of the system. Major structure, in figure 4.15, is the parts and bolt being tested, whereas additional structures are the extra parts connecting the major structure to fatigue testing machine.

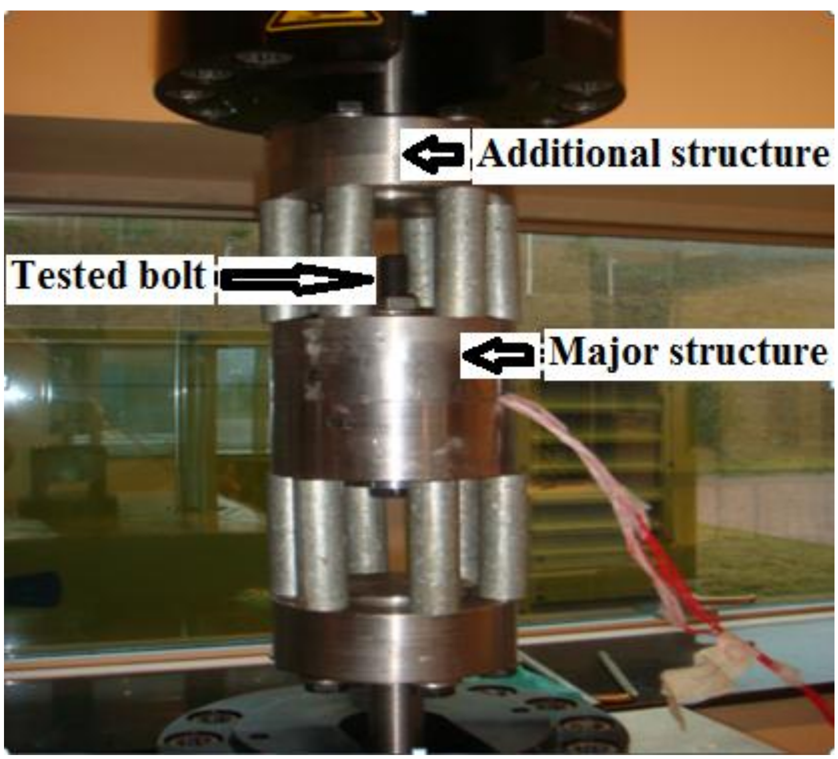


Figure 4.15 Bolt Structures

Six samples of M14 bolts are used in dynamic load tests, three tightened by conventional torque wrench and other three bolts tightened by hydraulic tension method. All bolts are tightened to reach a preload magnitude of 58500N (corresponding to 85% of the proof load this time), then an external cyclic load magnitude of 67100N applied with frequency 60cycle/sec for total number of 300.000 cycles. During tests, after every 50.000cycles test is interrupted for a short duration and bolt preload is measured by using strain gauges present on the bolt body. Figure 4.16 shows external fatigue testing machine loads and figure 4.17 shows total cyclic loads on the bolt during dynamic test.

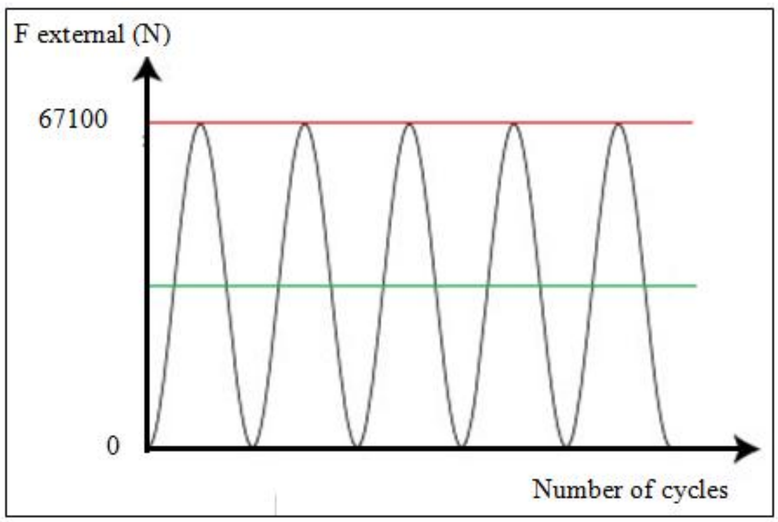


Figure 4.16 External fatigue testing machine loads

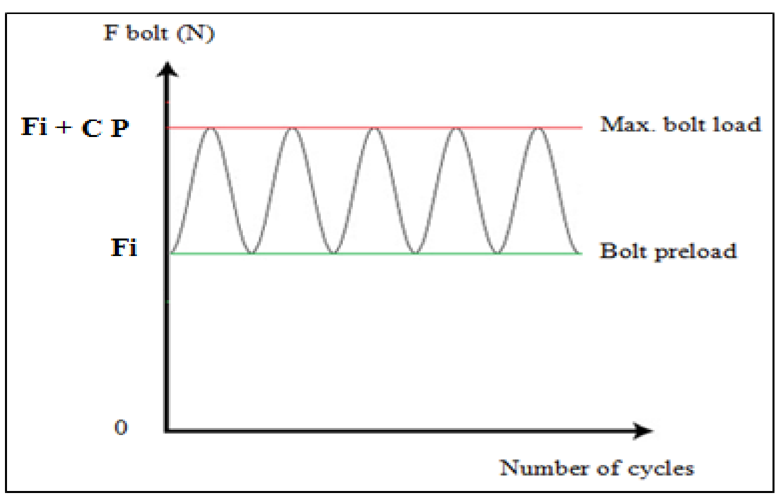


Figure 4.17 Total cyclic loads on the bolt during dynamic test.

CHAPTER 5

RESULTS AND DISCUSSION

5.1 Introduction

In this chapter, variation of major performance related to both bolt tensioning methods, conventional torque tightening method and hydraulic tensioning method are presented in graphical and table forms for preload results during tightening, and after tightening in static and dynamic loads. Both tightening methods measurements done in room temperature.

The experimental test results, obtained for three bolt sizes (M14, M16, and M18) were used to investigate the accuracy of tightening methods (conventional torque tightening and hydraulic tensioning) to reach the target preload values.

All bolts used in testing are instrumented with strain gauges for axial stress and torsion stress calculations and exact measurement of the preloading generated by two different tightening methods. In dynamic test part, cyclic constant load is applied to the structure to study the behavior of tightened bolts and discuss the effect of tightening method on preload relaxation during cyclic load. Torsional stress, which is built up in a fastener is also discussed and presented in this chapter for both tightening methods.

5.2 Static Test Results and Discussions

The problem with bolts is how to apply correct tension (preload). A bolt used in any situation, where the bolt is under tension to hold two or more components together must be tensioned correctly by tightening methods. A bolt after tightening must have the proper preload applied, if not it may come loose or cause over loading. Bolts and joint members, respond elastically as the bolt is tightened by any tightening method, so the joint members are compressed to a slight amount and the bolt is stretched by a larger amount.

Whatever the method used to tighten a bolt, the goal is in fact to apply a traction load to the bolt, and a compression load to the assembled components. In general, the bolt has a relatively low stiffness compared with that of the structural parts on which the compression stress is applied. Figure 5.1 shows that line S1, which corresponds to the bolt, has only a slight slope, whereas S2 corresponds to the structure. Naturally, the tension in the bolt has the same value with structure, but under tension and structure under compression.

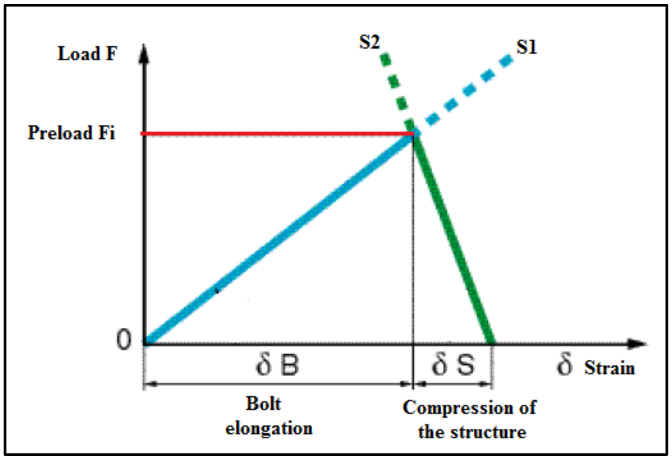


Figure 5.1 Load/deformation diagram of a bolted joint

Experiments were conducted subjecting bolts of M14, M16 and M18 to the same target preload in both tightening methods, the deviation between preload results from target preload studied and analyzed the accuracy of both tightening methods.

5.2.1 Conventional Torque Tightening

In general, fasteners in applications are designed for maximum utilization of materials and material strength safe preload magnitude is normally the target, but preload in bolted connections differs according to accuracy of tightening method.

Torques specified in Table 4.4 are applied by conventional torque wrenches with mechanical setting and the actual preloads induced in the bolt body are measured and calculated by using strain gauges installed in the bolts. Applied torque vs actual preload results are given in figures (5.2, 5.3 and 5.4).

Table 4.4 Static test preload magnitude for grade 8.8

Bolt Size	Bolt Grade	Tensile Stress Area (mm²)	Yield strength (Mpa)	Fy (N)	Target Preload Fi (N)	Fi / Fy
M14	8.8	122.65	660	80916	68810	85%
M16	8.8	165.04	660	108926	92591	85%
M18	8.8	213.71	660	141048	97222	69%

Reading strain magnitudes by FLA strain gauges used to measure the normal stress in the bolt (Hooks law), and preload magnitude calculated from normal stress as below formulas

$$E = \frac{\sigma}{\varepsilon} \quad \text{(Hooks law)} \tag{5.1}$$

$$\sigma = \frac{Fi}{A} \tag{5.2}$$

Where

E = Young's modulus

σ = Normal stress

ε = strain

A = bolt area

For M14 bolts, the target preload value adopted to reach the level of 85% of yield strength, table 5.1 shows the preload magnitudes during tightening and after tightening.

Table 5.1 conventional torque tightening preload magnitudes for M14 (with K = 0.2) bolts

Torque (N.m) applied by torque wrench	**Strain on the Bolt Body and Preload Magnitude**					
	Sample 1		**Sample 2**		**Sample 3**	
	$\mu \in$	**Preload Magnitude (N)**	$\mu \in$	**Preload Magnitude (N)**	$\mu \in$	**Preload Magnitude (N)**
0	0	0	0	0	0	0
25	272	2349	265	6969	176	4607
50	444	11655	463	12167	394	10343
75	635	16656	855	22444	685	17981
100	875	22956	1199	31474	955	25069
125	1095	28744	1433	37616	1282	33639
150	1378	36173	1661	43588	1603	42066
173	1825	47906	2008	52697	1878	49284

Figure 5.2 shows preloads values after tightening for three samples of M14, sample 1 preload magnitude after tightening comes out to be 47906 N, with a deviation of nearly 30.3 %. For the second sample of M14, the preload magnitude is 52697 N, with a deviation of 23.4%. The third sample of M14 reaches a preload magnitude of 49284 N, with a deviation of 28.3%. According to strain gauges results, all three samples need further in tightening in order to reach the target preload. In the practical application the technician depends on torque wrench tool as measuring device, and they don't know the exact value of preload and how to be far from the target preload value, thus is a rise in critical bolted connection application.

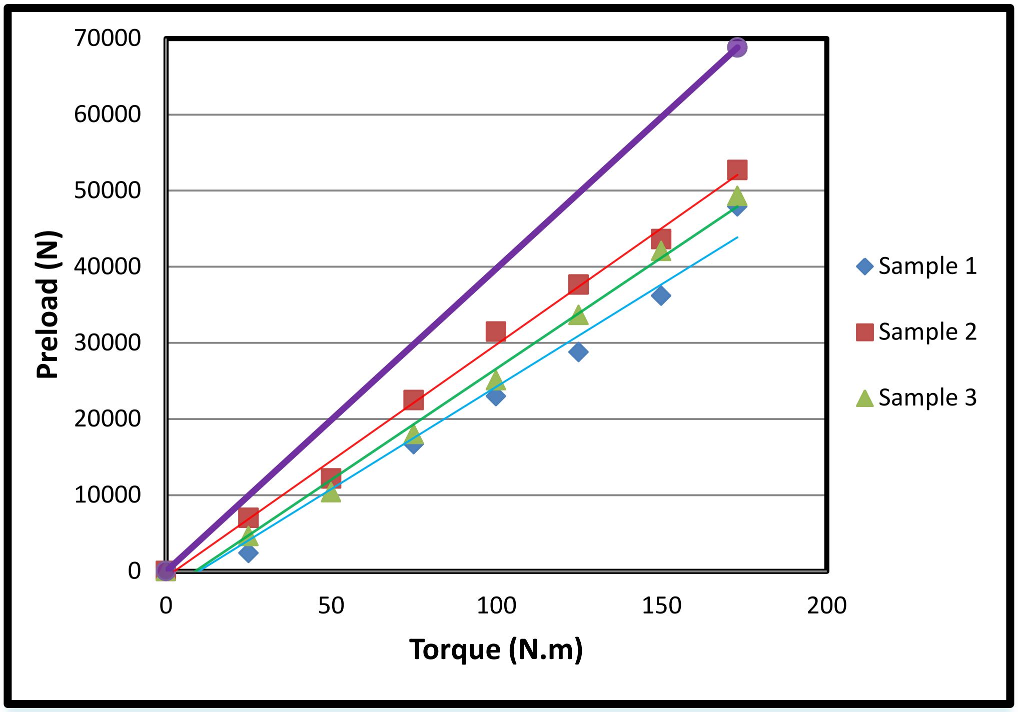


Figure5.2 M14 torque vs preload curves

For M16 bolts also the target preload value is to reach the level 85% of yield strength, table 5.2 shows the preload magnitudes during tightening and after tightening.

Table 5.2 conventional torque tightening preload for M16 (with K = 0.2) bolts

Torque (N.m) applied by torque wrench	Strain on the Bolt Body and Preload Magnitude					
	Sample 1		Sample 2		Sample 3	
	$\mu \in$	Preload Magnitude (N)	$\mu \in$	Preload Magnitude (N)	$\mu \in$	Preload Magnitude (N)
0	0	0	0	0		0
70	400	11690	561	19674	377	13247
100	424	16700	717	23869	464	16288
130	509	21710	825	25145	629	22063
160	590	26720	1000	35070	867	30436
190	683	31730	1228	43087	1120	39287
220	804	36740	1725	60515	1202	42156
250	974	41750	2255	79083	1514	53128
270	1689	59301	2740	96092	1585	55621

Figure 5.3 shows preloads magnitude after tightening for three samples of M16. The target value of preload is 92591 N. For the sample 1 preload magnitude after tightening comes out to be 59301 N, with a deviation of nearly 36.7 %. For the second sample of M16, the preload magnitude is 96092 N, with a deviation of + 2.5 %, the preload magnitude exceed the target preload for that reason plus sign in deviation percentage added. The third sample of M16 reaches a preload magnitude of 55621 N, with a deviation of 40.6 %.

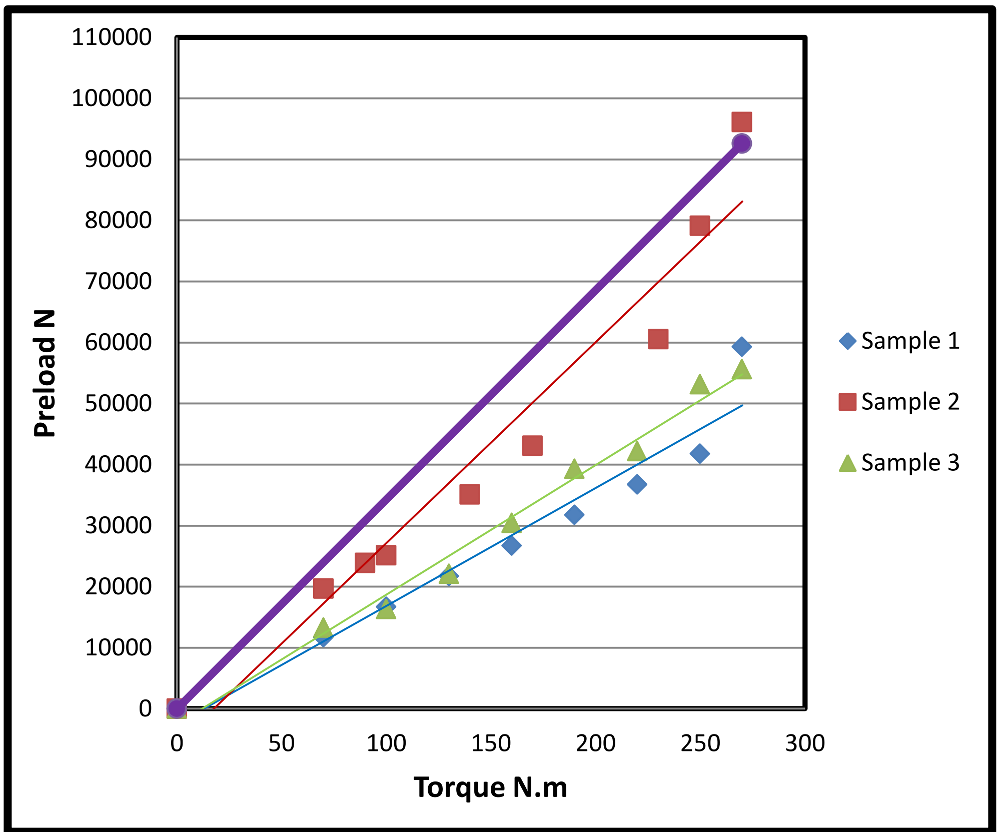


Figure5.3 M16 torque vs preload curves

The scenario of target preload value was changed, 69% of yield strength for M18 bolts. Table 5.3 shows the preload magnitudes during tightening and after tightening. The target preload of M18 bolts with torque 350 N.m comes out to be 97222 N.

Table 5.3 conventional torque tightening preload magnitudes for M18 (with K = 0.2) bolts

Torque (N.m) applied by torque wrench	Strain on the Bolt Body and Preload Magnitude					
	Sample 1		Sample 2		Sample 3	
	$\mu \in$	**Preload Magnitude (N)**	**$\mu \in$**	**Preload Magnitude (N)**	**$\mu \in$**	**Preload Magnitude (N)**
0	0	0	0	0	0	0
70	370	14959	320	18264	351	14235
80	445	17096	375	22365	423	18961
90	493	19233	475	26845	510	22865
100	551	21370	500	30586	531	23830
140	756	29918	815	36575	833	37383
180	967	38466	1100	49365	1093	49028
220	1175	47014	1500	67316	1363	61167
260	1385	55562	1950	87510	1648	73935
300	1646	64110	2250	114436	1926	86433
320	1741	68384	2650	118924	2154	96662
350	1885	84348	2879	129214	2381	106852

Figure 5.4 shows preloads magnitude after tightening for three samples of M18, the target value of preload is 97222 N, for the sample 1 preload magnitude after tightening comes out to be 84348 N, with a deviation of nearly 13.2 %. For the second sample of M18, the preload magnitude is 129214 N, with a deviation of +32.9 %, the preload magnitude exceeds the target preload level, so plus sign added. The third sample of M18 reaches a preload magnitude of 106852 N, with a deviation of + 9.9 %.

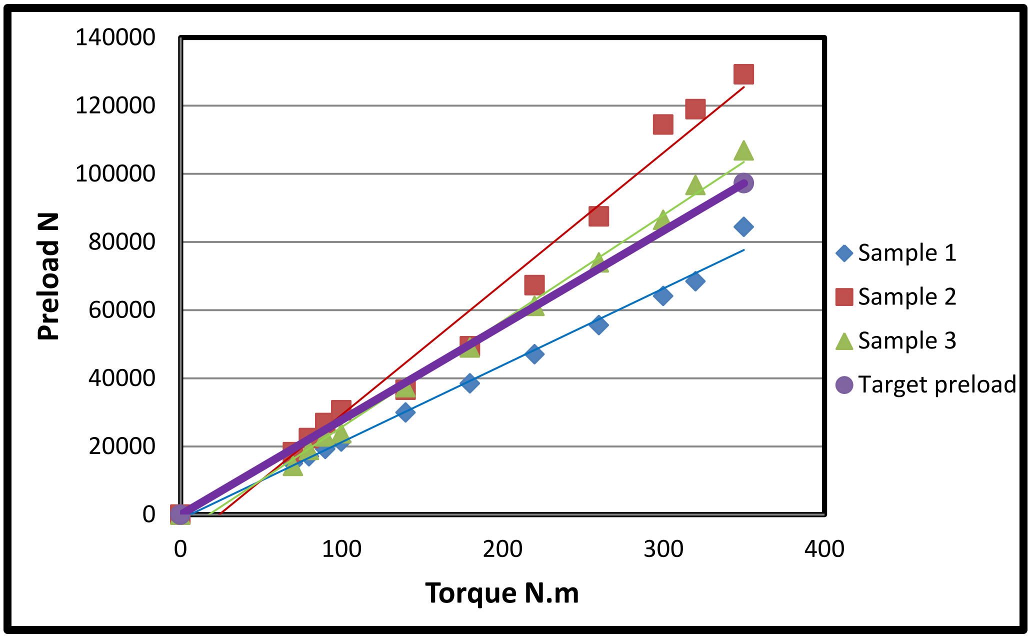


Figure5.4 M18 torque vs preload curves

According to the conventional torque tightening method results, generally the final preload magnitude deviation from target preload are not same, even for the same size of the bolt, the final tightening preload magnitude variation depends on some parameters, the main parameter is a nut factor in the torque equation (3.4), this factor depends on the friction coefficients in the threads of the bolt and the nut, and on the bearing contact surface condition between the nut and the structure, also the number of times a bolt has been installed.

Generally in torque tightening method applied, the deviation in the final tightening preload of the bolts can vary between +/- 20% when conditions are good, and +/- 60% when the condition is bad, this wide range in preload magnitude is due to the combination of the following three phenomena:

1. The accuracy and tolerance in the applied torque tool.
2. Geometric effects and surface roughness on the threads and the bearing surfaces of the fastened components.
3. Degree of lubrication of bearing surface.

5.2.2 Hydraulic Tensioning Tightening

During hydraulic tightening method, the hydraulic force can be achieved by creating a controllable hydraulic pressure applied to a pre-calculated surface area of a hydraulic jack/cylinder; table 4.5 shows the adopted hydraulic pressures to reach the target preload. The same magnitude of target preloads in torque tightening used to calculate the required hydraulic pressures for three different bolt sizes.

Table 4.5 Preload vs Hydraulic pressures with (load loss allowance factor =1.1)

Bolt Size	Target Prelaod (N)	Hydraulic Pressure (bar)
M14	68810	171
M16	92591	230
M18	97222	242

For M14 the target preload value adopted to reach the level of 85% of yield strength which is equal 68810 N, table 5.4 shows the preload magnitudes during tightening and after tightening by the hydraulic tensioning method.

Table 5.4 hydraulic tensioning pressures and preload magnitudes for M14

Sample 1			Sample 2			Sample 3		
Pressure (bar)	**$\mu \in$**	**Preload Magnitude (N)**	**Pressure (bar)**	**$\mu \in$**	**Preload Magnitude (N)**	**Pressure (bar)**	**$\mu \in$**	**Preload Magnitude (N)**
0	0	0	**0**	0	0	**0**	0	0
30	435	11406	**25**	464	12167	**36**	547	14372
55	817	21433	**48**	690	18099	**61**	938	24623
83	1266	33219	**74**	1060	27825	**95**	1409	36999
110	1663	43641	**102**	1519	39874	**108**	1575	41348
128	1935	50781	**121**	1815	47656	**125**	1893	49704
150	2268	59535	**133**	1984	52080	**143**	2179	57214
159	2420	63525	**159**	2370	62199	**159**	2440	64050

Figure 5.5 shows the target value of preload (68810N) and preloads magnitude after tightening for three samples of M14. For the sample 1 preload magnitude after tightening comes out to be 63525 N, with a deviation of nearly 7.6 %. In the second sample, the preload magnitude is 62199 N, with a deviation of 9.6 %. The third sample reaches a preload magnitude of 64050 N, with a deviation of 6.9 %.

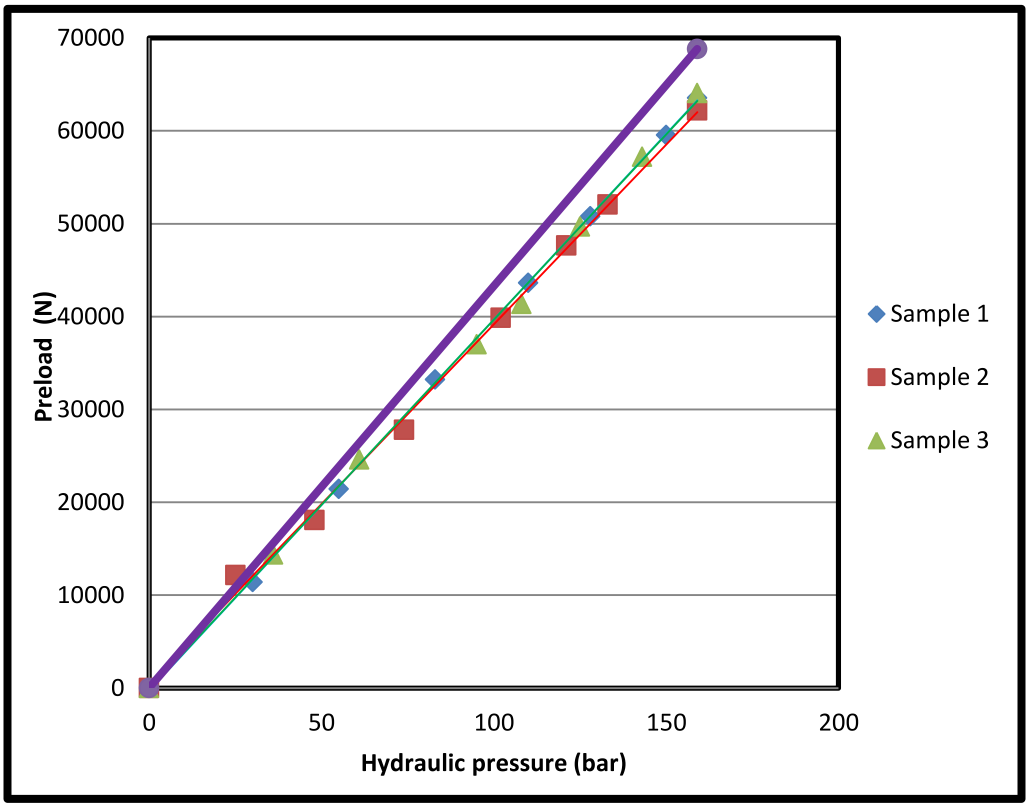


Figure5.5 M14 Hydraulic pressure vs preload curves

Also for M16 the target preload value adopted to reach the level of 85% of yield strength which is equal 92591 N, table 5.5 shows the preload magnitudes during tightening and after tightening by the hydraulic tensioning method.

Table 5.5 hydraulic tensioning pressures and preload magnitudes for M16

Sample 1			Sample 2			Sample 3		
Pressure (bar)	**$\mu \in$**	**Preload Magnitude (N)**	**Pressure (bar)**	**$\mu \in$**	**Preload Magnitude (N)**	**Pressure (bar)**	**$\mu \in$**	**Preload Magnitude (N)**
0	0	0	**0**	0	0	**0**	0	0
28	294	10311	**20**	212	7452	**20**	253	8169
51	569	19972	**52**	580	20358	**52**	522	21063
76	849	29774	**74**	840	29476	**74**	820	34285
97	1094	38384	**100**	1137	39892	**100**	1127	42850
127	1448	50781	**130**	1487	52149	**130**	1400	50681
153	1735	60864	**150**	1705	59794	**150**	1719	62894
167	1944	68176	**172**	1965	68913	**172**	1957	70511
207	2390	83835	**200**	2279	79925	**200**	2274	78112
220	2549	89411	**219**	2491	87377	**219**	2497	85250
230	2708	94970	**230**	260	91270	**230**	2750	90135

Figure 5.6 shows preloads magnitude after tightening for three samples of M16, with target value of 92591 N, the same target preload used torque tightening method. For the sample 1 preload magnitude after tightening comes out to be 94970 N, with a deviation of nearly +1.3 %. Again plus sign clears that preload magnitude exceeds the target preload, but it is very close to target preload. In the second sample, the preload magnitude is 91270 N, with a deviation of 2.6 %. Finally the third sample reaches a preload magnitude of 90135 N, with a deviation of 3.7 %.

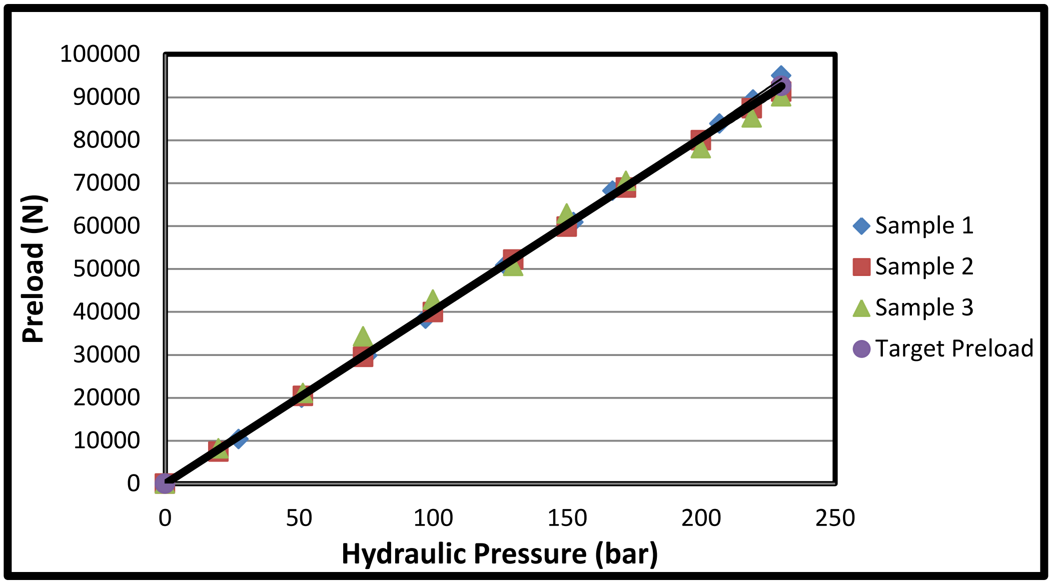


Figure5.6 M16 Hydraulic pressure vs preload curves

For M18 the target preload value is to reach 69% of yield as in the case of torque tightening. Table 5.6 shows the preload magnitudes.

Table 5.6 hydraulic tensioning pressures and preload magnitudes for M18

Sample 1			Sample 2			Sample 3		
Pressure (bar)	$\mu \in$	**Preload Magnitude (N)**	**Pressure (bar)**	$\mu \in$	**Preload Magnitude (N)**	**Pressure (bar)**	$\mu \in$	**Preload Magnitude (N)**
0	0	0	**0**	0	0	**0**	0	0
13	78	3500	**18**	147	6619	**18**	174	7812
25	174	7809	**31**	266	11937	**31**	236	10598
48	357	16021	**52**	451	20262	**52**	438	19684
75	580	26029	**75**	645	28968	**75**	613	27536
99	803	36036	**100**	868	38953	**100**	798	35841
126	1034	46403	**122**	1089	48871	**122**	1053	47269
158	1326	59484	**155**	1351	60629	**155**	1333	59847
180	1519	68168	**180**	1575	70681	**180**	1551	69645
199	1688	75730	**201**	1752	78647	**201**	1766	79287
234	1984	89014	**225**	1970	88408	**225**	1903	87523
242	2081	93367	**242**	2152	96598	**242**	2100	94286

Figure 5.7 shows preloads magnitude after tightening for three samples of M18, the target value of preload is 97222 N. For the sample 1 preload magnitude after tightening comes out to be 93367 N, with a deviation of nearly 3.9 %. For the second sample of M18, the preload magnitude is 96598 N, with a deviation of 0.6 %. The third sample of M18 reaches a preload magnitude of 94286 N, with a deviation of 3.0 %.

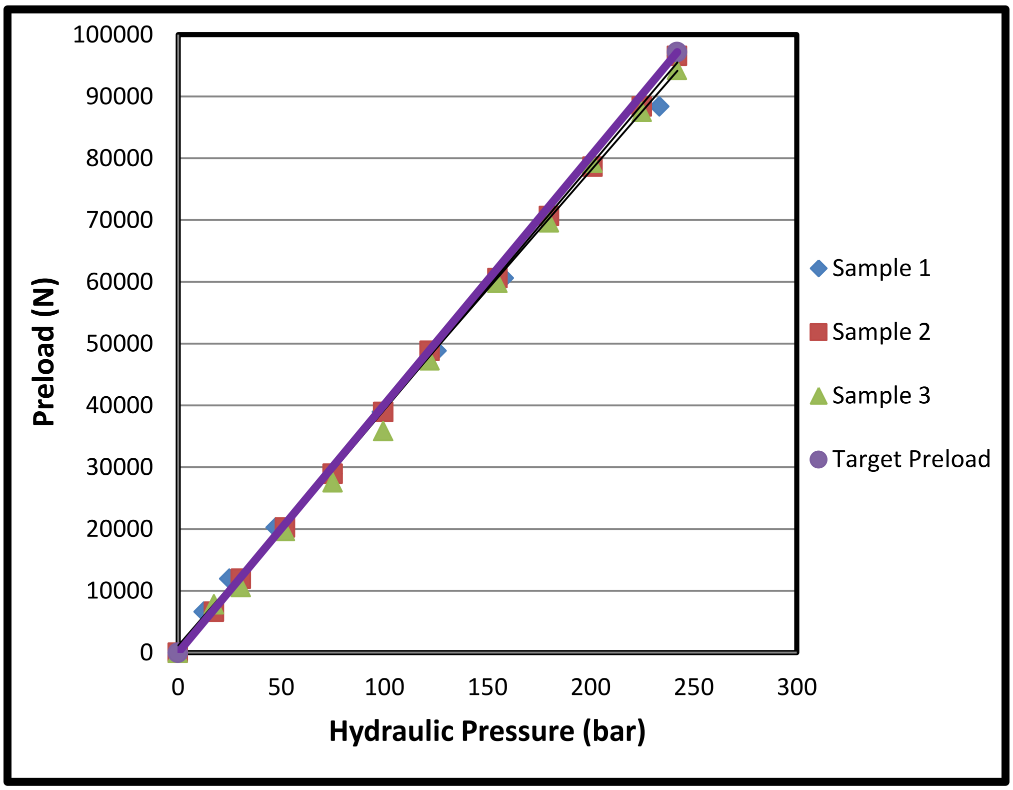


Figure5.7 M18 Hydraulic pressure vs preload curves

Tightening tests with hydraulic tension method shows that the final magnitudes of preload are much closer to target values for 3 different sizes of the bolts compared with results of torque wrench method. The Hydraulic tension method is accurate, because the most important parameter, namely the traction load, is perfectly controlled through the hydraulic pressure in the tension system, so the load does not depend on the various friction coefficients in the assembly.

5.2.3 Comparison of Preload Magnitudes Results of Two Tightening Methods

Study results of preload magnitudes have been used to create tables 5.7 and 5.8 for conventional torque tightening method and hydraulic tension method. Percentage deviations were calculated based on the difference value of target value and actual measured bolt tension value. A plus sign of deviation means that actual value has above the target value. As seen from both tables, deviations of torque method reach as high as 40%, while majority of results are in between 10-30% with an exception of 2.6%. However, all deviations are below 10% in hydraulic tension method with a maximum of 9.6%. These results support the claim of having a more accurate preload of a bolted connection by the method of hydraulic tension which can help effective use of both bolt

Table5.7 Final preloads and percentage deviation of preload magnitude in conventional torque tightening method

Bolt size	Target preload (N)	Sample 1		Sample 2		Sample 3	
		Final preload (N)	Deviation %	Final preload (N)	Deviation %	Final preload (N)	Deviation %
M14	68810	47906	30.3	52697	23.4	49284	28.3
M16	92591	59301	36.7	96092	+2.6	55621	40.6
M18	97222	84348	13.2	129214	+32.9	106852	+9.9

Table5.8 Final preloads and percentage deviation of preload magnitude in hydraulic tension tightening method

Bolt size	Target preload (N)	Sample 1		Sample 2		Sample 3	
		Final preload (N)	Deviation %	Final preload (N)	Deviation %	Final preload (N)	Deviation %
M14	68810	63525	7.6	62199	9.6	64050	6.9
M16	92591	94970	+1.3	91270	2.6	90135	3.7
M18	97222	93367	3.9	96598	0.6	94286	3

Analyzing final preload results in both tightening methods, it is clear that a hydraulic tension method is more accurate from conventional torque tightening method to reach the preload target level, generally it seems that our trusty torque wrench is not that accurate measuring tool to achieve preload closer to target preload value, this fact is includes the all conventional torque tightening, and the greater the accuracy required to change the tightening method from conventional torque tightening to hydraulic tension method. The reason that torque wrench is so inaccurate is due to friction in the threads between bolt and nut and under the bolt head/ nut and washer.

5.3 Parasite Torsion Stress

In addition to the desired axial tension stress, torque tightening introduces an additional "parasite" torsion stress in the bolt during tightening which can, in some cases, can reach over 30% of the tension stress. This stress component, by its nature, is not present in hydraulic tension method since no torsion stress apears in bolt.

The resulting equivalent total (Von-Mises) stress in the bolt is then calculated by using both axial and torsion stress components. Some of the torsion stress may be released/relaxed by immediately spring back when conventional torque tightening complete. The amount of relaxation depends on the friction under the bolt head or nut with structure.

5.3.1 Parasite Torsion Stress Effect on Conventional Torque Tightening

Figure 5.8 shows three components of stress (axial, torsion and total von-misses) in three samples of M14 bolts tightened by the conventional torque wrench method. For sample 1 the axial stress magnitude is 383 Mpa and torsion stress magnitude is 118 Mpa, while equivalent stress reached 435 Mpa. It is seen that in sample 1, after tightening complete, the torsion stress - which attains 30 percent of the traction stress - leads to an increase in the equivalent stress of approximately 12 %. For sample 2 the axial stress magnitude is 422 Mpa and torsion stress magnitude is 105 Mpa, equivalent stress reached reached 459 Mpa. The increase in the equivalent stress approximately 8 %. For sample 3 the axial stress magnitude is 394 Mpa and torsion stress magnitude is 124 Mpa while equivalent stress after tightening equal 449 Mpa. The increase in the equivalent stress approximately more than 13%

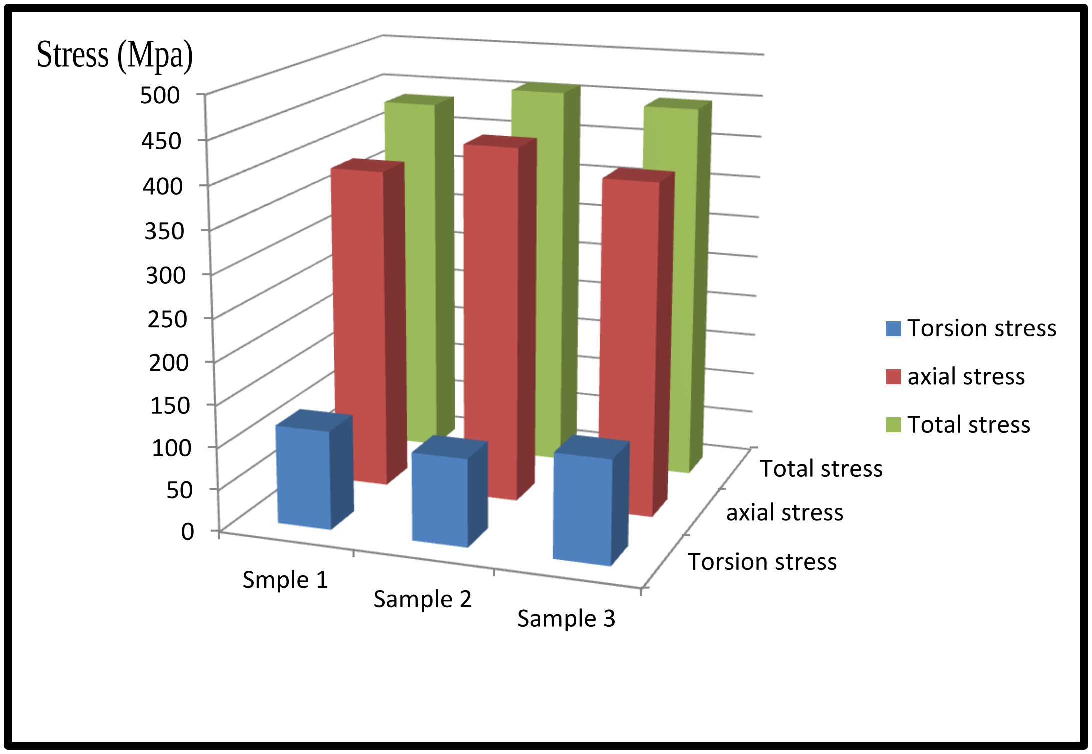


Figure 5.8 Total stresses in M14 after conventional torque wrench tightening

Although parasite torsional stress in bolts arises during torque tightening and subject to varying amounts of relaxation after torque tightening complete, the additional “parasite” torsion stress reaches value need to be added to the designer stress calculation. For M 14, in sample 1 torsional stress represents 30% of axial stress and in sample 2 torsional stresses representing nearly 24% of axial stress, while in sample 3 torsional stresses representing 30% of axial stress in the bolt during tightening by conventional torque tightening.

The equivalent stresses for M16 sample1 tightened by the conventional torque wrench method shown in figure 5.9, the axial stress after tightening complete comes out to be 355 Mpa and torsion stress magnitude of 115 Mpa, while equivalent stress reached 407 Mpa. After tightening complete, the torsion stress - which attains 32 percent of the traction stress - leads to an increase in the equivalent stress of approximately 14 %. For sample 2 the axial stress comes out to be 575 Mpa and torsion stress comes out to be 118 Mpa, torsional stress representing 21% of axial stress, equivalent stress reached a value above the proof stress and equal 611 Mpa. The increase in the equivalent stress approximately 6%. For sample 3 the axial stress magnitude is 333 Mpa and torsion stress magnitude is 102 Mpa, torsional stress representing 30% of axial stress, while equivalent stress after tightening comes out to be 377 Mpa. The increase in the equavalent stress approximatly more than 11%.

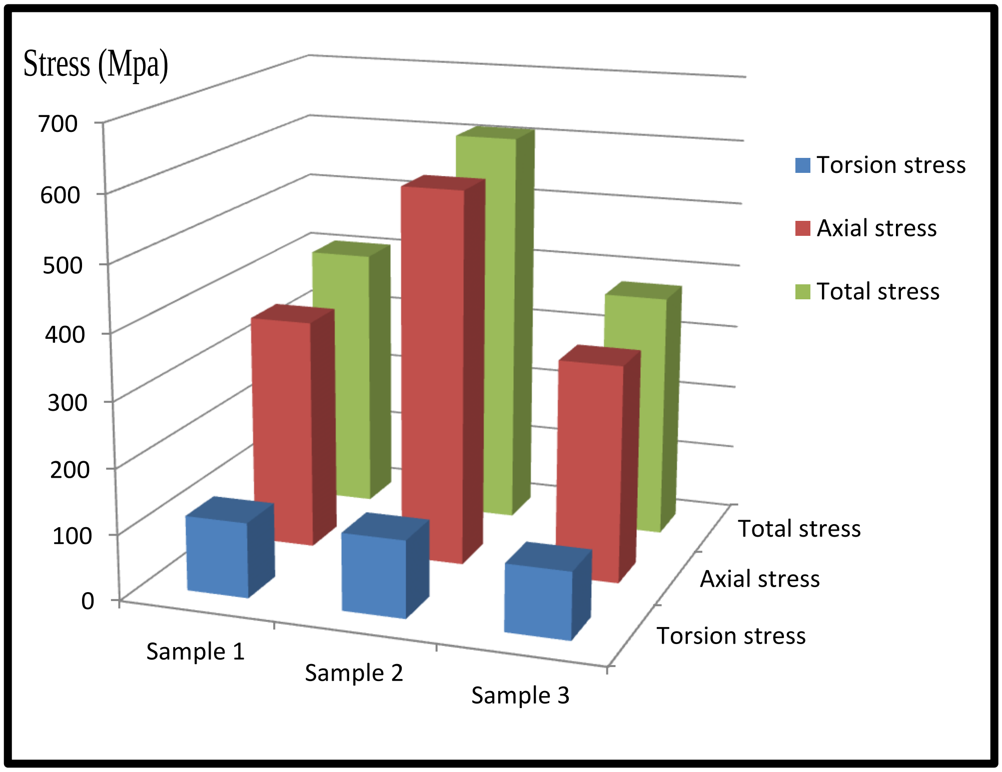


Figure 5.9 Total stresses in M16 after conventional torque wrench tightening

Generally the behavior of bolts for torsion stress effects under conventional torque tightening are same, so approximately the same increase percentage of equivalent stress appears during conventional torque tightening. Figure 5.10 shows three components of bolt stress (axial, torsion and total von-mises) in three samples of M18 bolts tightened by the conventional torque wrench method. For sample 1 the axial strees comes out to be 396 Mpa and torsion stress comes out to be 106 Mpa, while equivalent strees 436 Mpa. It is seen that in sample 1, after tightening complete, the torsion stress - which attains 26 percent of the traction stress - leads to an increase in the equivalent stress of approximately than 10 %.

For sample 2 the axial stress magnitude is 605 Mpa and torsion stress magnitude is 115 Mpa, equivalent stress reached comes out to be 636 Mpa.The increase in the stress approximately 5 %. Addition torsion stress adding parasite stress to equivalent stress and lead to be closer to the yield point, if we continue tightening to make preload closer to target value, in this case, the yield point of the material will probably be exceeded.

For sample 3 the axial stress equal 500 Mpa and torsion stress comes out to be 94 Mpa while total stress after tightening comes out to be 526 Mpa. The increase in the equivalent stress approximately more than 5 %.

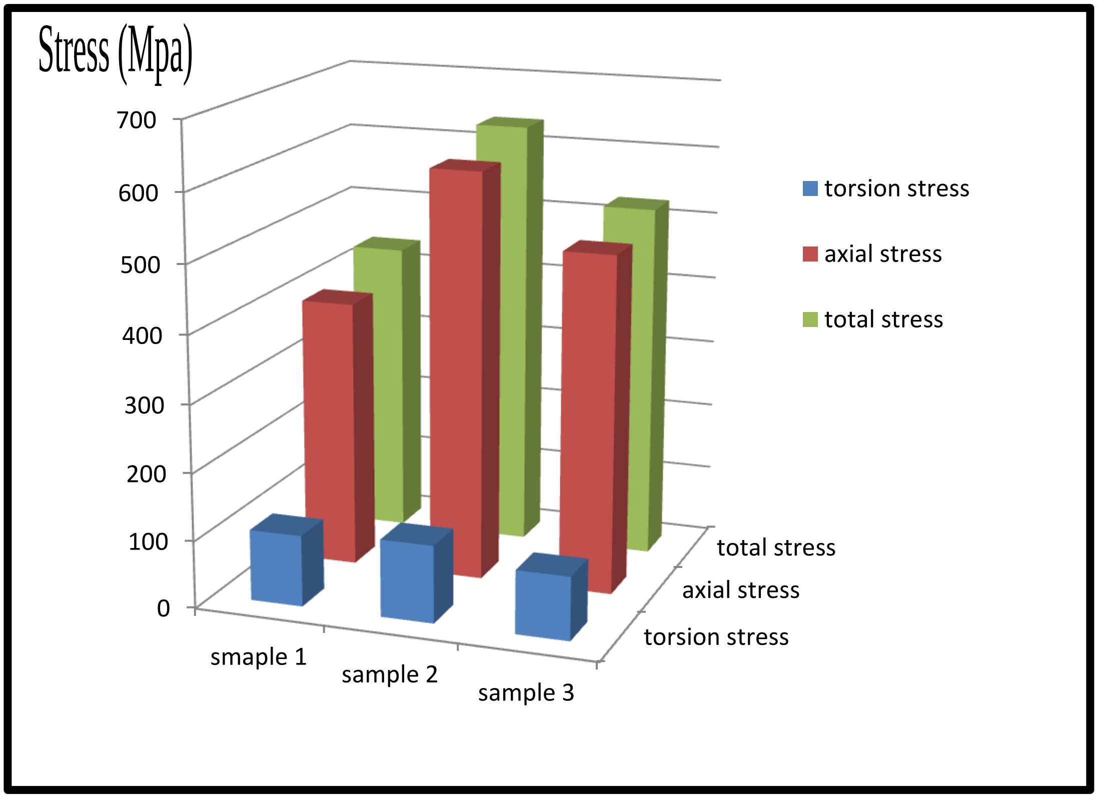


Figure 5.10 Total stresses in M18 after conventional torque wrench tightening

For M18, in sample 1 torsional stress represents 26% of axial stresses and in sample 2 torsional stresses representing 19% of axial stress, while in sample 3 torsional stresses representing 18% of axial stress in the bolt during tightening by conventional torque tightening.

Due to both large inaccuracy and also the parasite effect of the torque tightening method, equivalent stress may go near to yield stress and sometimes over the yield stress of the bolt. Furthermore, the residual torsion stress may increase the risk of spontaneous or gradual loosening at a later stage after tightening is complete.

5.3.2 Parasite Torsion Stress Effect on Hydraulic Tension Tightening

During tightening by hydraulic tension method, within the bolt body only axial stress appears as a result and no torsion stress occurs, total stress results for M14, M16 and M18 shown in figures 5.11,5.12 and 5.13 respectively. For all bolts that tightened by hydraulic tension method the equivalent stresses are equal the axial stresses.

The principle of hydraulic tension tightening, is the preload produced and applied without any torsion, acting on the axial direction of the bolt or screw. Both, better or more accurate tightening of the bolt and having no torsion stress on the bolt body, are the major advantages of the hydraulic tension method over torque wrench method.

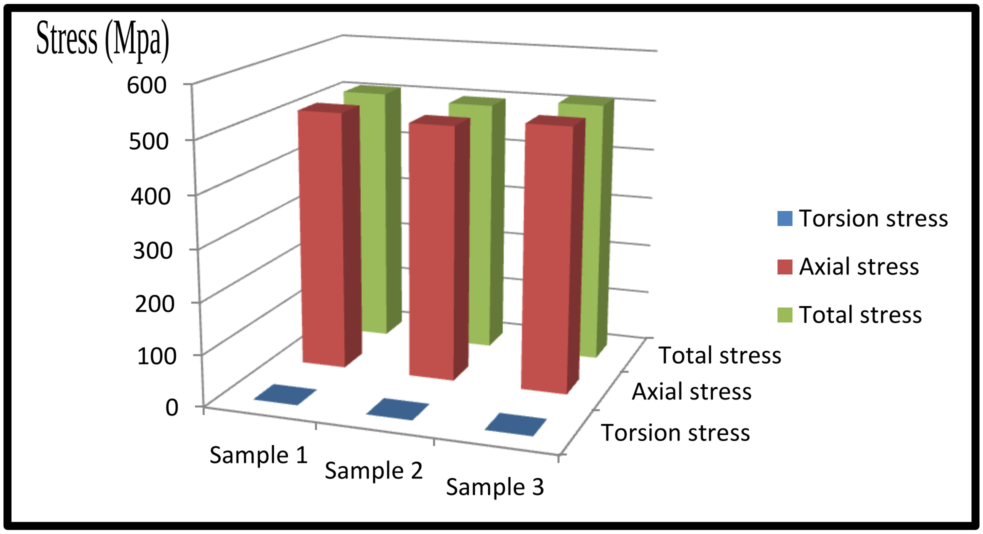


Figure5.11 Total stresses in M14 after hydraulic tension tightening

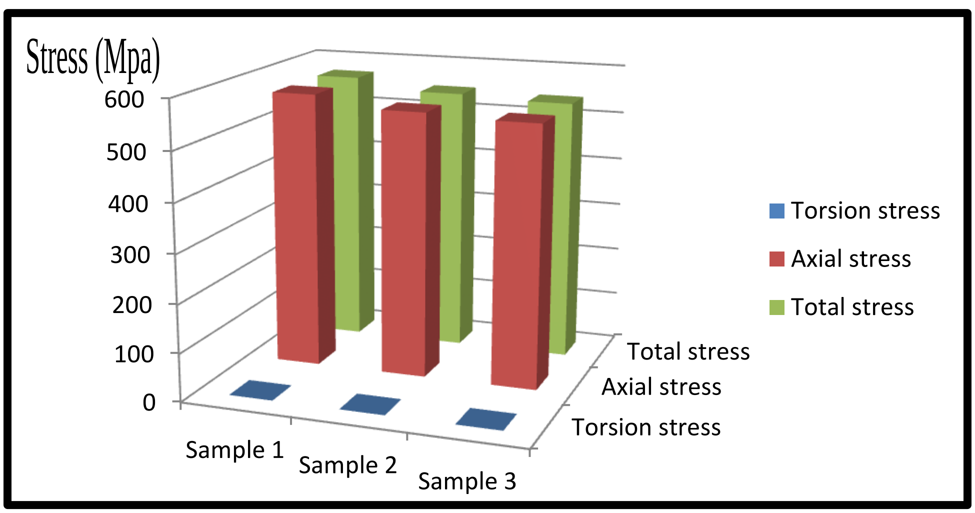


Figure5.12 Total stresses in M16 after hydraulic tension tightening

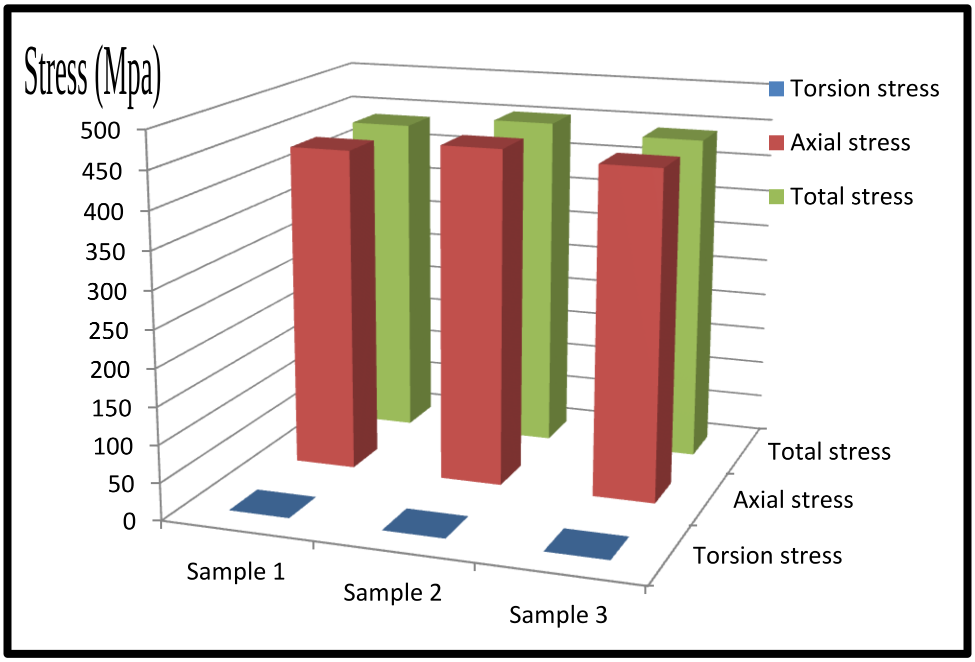


Figure5.13 Total stresses in M18 after hydraulic tension tightening

5.4 Static Embedment Relaxation

The process "embedment relaxation" appears in threaded joints which lead to reduction in preload magnitude after tightening completed. The embedment relaxation is phenomena between contact surfaces, each surface is pressed together, and the high spots are crushed and deformed to form a surface capable of supporting the load. Under optimum joint conditions expect preload static relaxation between 1 and 11 percent. All tightened bolts in this study final preload magnitude are decreased for both tightening methods, for example samples 3 in toque tightening methods for M 14 after tighten completed reached 49284 N, after 1 hour preload relaxed and reached 47066N. Sample 2 in hydraulic tensioning method after tightening completed comes out to be 62199 N, after 1 hour preload relaxed and reached 59404N.

5.5 Dynamic Test Results and Discussions

When the nut on a bolt is tightened, an initial tensile preload is placed on the bolt that must be taken into designer account in determining its safe working strength or external load-carrying capacity. The tension force on the bolt is equal to the force on the joint, which is equal to the preload, this equilibrium will change with the application of external load, When an external load is applied to the structure, the load push the structure parts to separate in the joint, part of this load will cause the further extension of the bolt (increase in bolt preload). Part of the load will result in an increase of the joint thickness, reducing of the compressive load on the joint; since the bolt and joint have different stiffness the external force magnitude will not be same for bolt and joint.

Cyclic loading applied to six samples of M14 bolts used in dynamic load tests, three tightened by conventional torque wrench and other three bolts tightened by hydraulic tension method, the external cyclic load magnitude was 67100N, with frequency 60 Hz, and number of total cycles were 300,000 cycles applied for each bolt. As shown in figure 5.14.

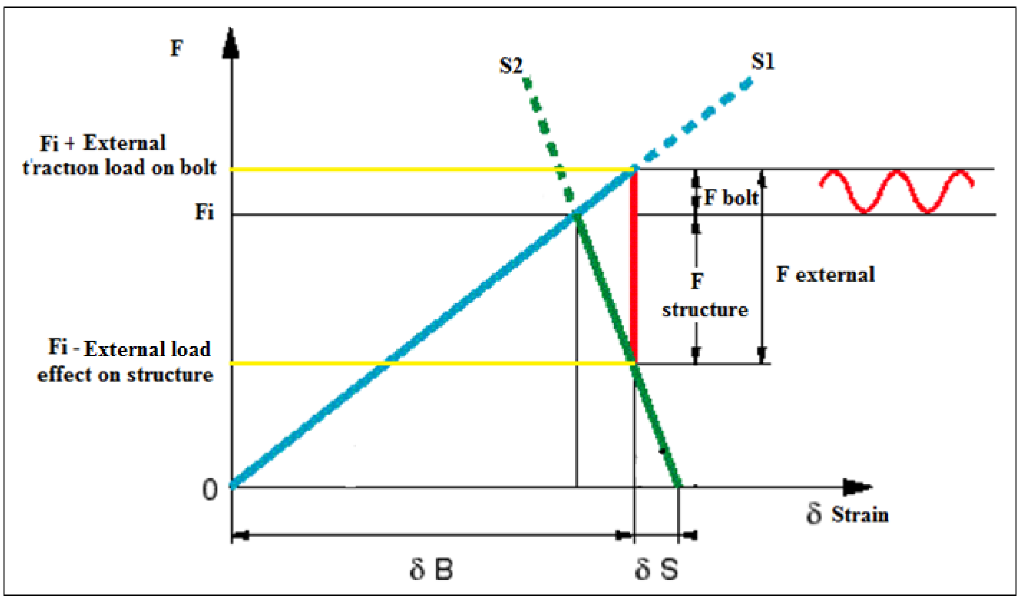


Figure5.14 Load diagram in the event of a cyclic external load effect on bolt

5.5.1 Preload Reduction Tightened by Conventional Torque Tightening Method

Figure 5.15 shows preloads magnitude reduction after applying constant cyclic fatigue loading magnitude (67100 N) for 300,000 cyclic on structure, this fatigue load applied by fatigue testing machine of ZWICK-Vibrophores, the target value of preload after conventional torque tightening is 58500N (corresponding to 85% of the proof load), sample 1 after tightening comes out to be 41042 N the preload magnitude decreases as the number of cycle increases until reached 24819 N. The preload dropped about 16223 N, so preload loss percentage is 39.5%. Sample 2 after tightening comes out to be 44704 N the preload magnitude decreases as the number of cycle increases until reaching 24124 N. The preload dropped about 20580 N, so preload loss percentage is 46%. While sample 3 after tightening comes out to be 27353 N the preload magnitude decreases as the number of cycle increases until reached 19937 N. The preload dropped about 7416 N, so preload loss percentage is 27.1%.

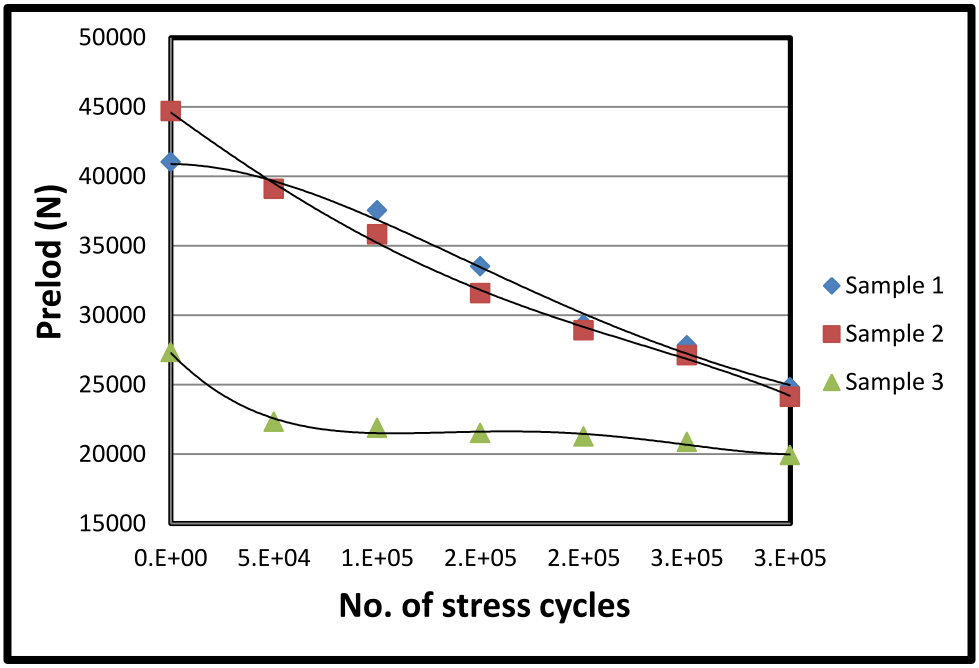


Figure5.15 Preload reduction vs load cycle for M14 bolts tightened by conventional torque wrench method

From the results of cyclic load as shown in figure 5.15, initially there is a rapid drop off preload which has occurred as a result of embedding in the first 50,000 cycles in all samples, this reduction is clear in sample 3, while in sample 1 and 2 preloads decrease regularly after 100,000 cycles.

The results have demonstrated that, for the same experimental setup and the same cyclic loading level, preload reduction percentage of the first 50,000 cycle in sample 1 reaches 4.7% this percentage will decrease after 100,000 cycle and reaches approximately 4%, Sample 1 preload reduction was scatter according to cyclic load, after 200,000 cycle the reduction percentage in preload reaches up 12.5%. For sample 2 preload reduction percentages for the first 50,000 cycle in sample 1 reaches 12 % this percentage will decrease after 100,000 cycle and equal approximately 8%. In sample 3 preload reduction percentage of the first 50,000 cycle is 18% this percentage will decrease after 100,000 cycles and equal approximately 2%.

5.5.2 Preload Reduction Tightened by Hydraulic Tensioning Method

Drop in bolt preload was observed during the cyclic load effect for bolts tightened by hydraulic tensioning method. Figure 5.16 shows preloads magnitude reduction after applying constant magnitude 300,000 cyclic loading 67100 N for three samples of M14 tightened by hydraulic tension method. Like conventional torque tightening method the same target value of preload after tighten is 58500N (corresponding to 85% of the proof load) is adopted, sample 1 after tightening comes out to be 49626 N, the preload magnitude decreases as the number of cycle increases until reached 46751 N. the preload dropped about 2875 N, so preload loss percentage is 5.8%. Sample 2 after tightening comes out to be 50374 N the preload magnitude decreases as the number of cycle increases until reached 47381 N, the preload dropped about 2993 N, so preload loss percentage is 5.9%. While sample 3 after tightening comes out to be 50400 N, the preload magnitude decreases as the number of cycle increases until reaching 47880 N. the preload dropped about 2520 N, so preload loss percentage is 5%.

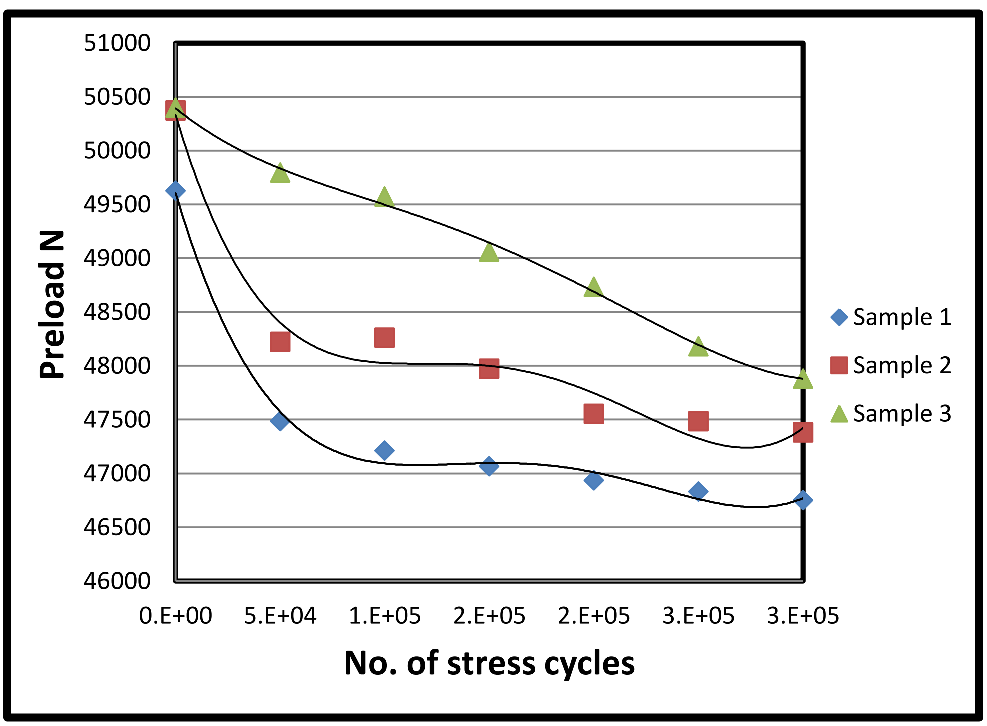


Figure5.16 Preload reduction vs load cycle for M14 bolts tightened by hydraulic tension method

From the results of cyclic load as shown in figure 5.16, also initially there is a rapid drop off preload which has occurred as a result of embedding in the first 50,000 cycles in all samples, this behavior is clear in samples 1 &2, while in sample 3 preload is decrease regularly.

5.5.3 Comparison of Preload Reduction of Two Tightening Methods

Generally bolt tension reduced just after tightening. This drop could be attributed to many different phenomena like embedding and creep, this process called embedment relaxation.

Embedding is the effect of locally yielding the contact surfaces. There are two forms of embedding: static and dynamic embedding. Due to surface imperfections and the surface roughness of contact parts, the actual contact surface is always smaller than the theoretical one. This happens at the interface between the component and the head of the bolt or nut, as well as on the thread surface. This will lead to stress concentrations which will cause a local surface, yielding in contact parts. When the bolted connection is cyclically loaded, this local surface, yielding will increase to reach a constant point and then stopped embedding.

Dynamic embedding occurs when the bolt performs a marginal motion under loading. Due to sliding of the bolt head/ nut on the component, the surface of the component can yield locally. The coating of the component and mechanical strength properties of the bolt head and of the clamped component material are significant parameters. In usual case, dynamic embedding can lead to a pretension loss until fatigue signs effect occurs.

Preload magnitude decreases more during the effect of external cyclic loading in conventional torque tighten method comparing with hydraulic tensioning method; because of the friction effect which is simply appearing in torque tighten. Generally embedment increase by increasing friction between the nut and washers, as well between the nut and bolt threads, because surfaces during rotation of nut to tighten are scratches. By applying cyclic load surface squeeze together. Any nicks or alignment errors are gradually crushed down, bolt threads also embed. While threads are pulled in shear, slightly increasing the thread pitches. At the same time nut threads are compressed and lose a little pitch. Cyclic loading increase embedment, the surfaces presses further together and reduce the bolt's clamping force (preload).

Briefly hydraulic tension tightening method reaches target preload by keeping the contact, surfaces, bolt head/ nut and part surfaces as well as thread in good conditions, while conventional torque tightening method scratches the surfaces and threaded because of the friction effect during tightening.

Relatively large reductions in preload values (nearly 25-45%) are recorded in case of torque wrench method for a period of 300.000 cycles of dynamic loading. For the same period of loading cycle, reduction values in preloads are much smaller (5-6%) for the case of hydraulic tension method, tables 5.9 & 5.10 summarizing the dynamic test results of both tightening methods with a clear advantage and reliability of a hydraulic tension method to hold the initial preload.

Table5.9 Cyclic load effect on preload magnitudes tightened by conventional torque wrench method

Sample No.	**Preload after tightening (N)**	**Preload after 300000 cycles (N)**	**Preload loss percentage %**
Sample 1	41042	24819	39.5
Sample 2	44704	24124	46.0
Sample 3	27353	19937	27.1

Table5.10 Cyclic loads effect on preload magnitudes tightened by hydraulic tension method

Sample No.	**Preload after tightening (N)**	**Preload after 300000 cycles (N)**	**Preload loss percentage %**
Sample 1	49626 N	46751 N	5.8%
Sample 2	50374 N	47381 N	5.9%
Sample 3	50400 N	47880 N	5.0%

When the bolts are loose preload, damage to the coating can be spotted on the washers. This is proven with the figure 5.17

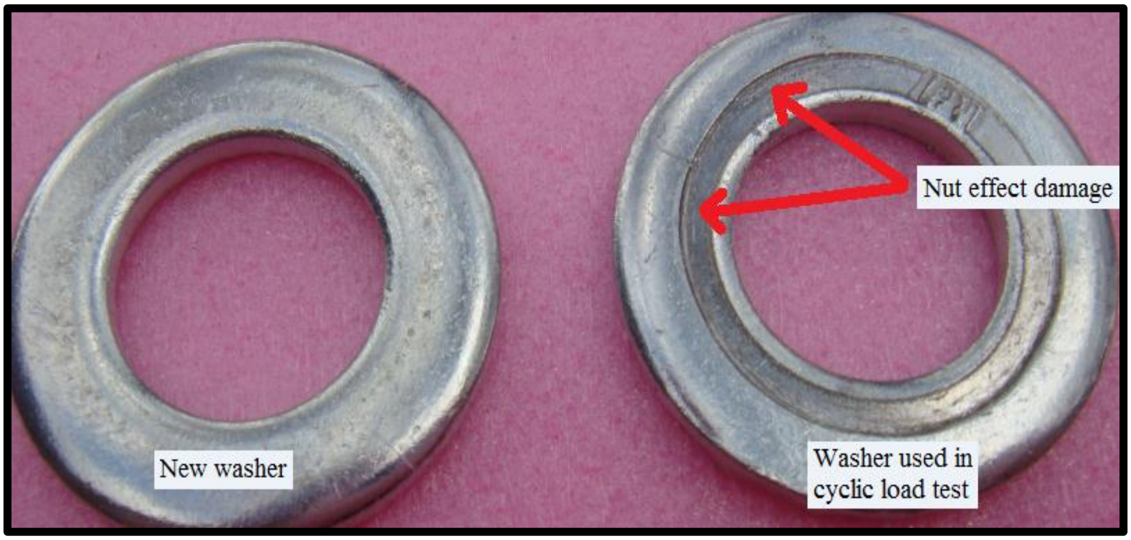


Figure 5.17Conventional torque tightening and cyclic load effect on washer surface

5.5.4 Torsion Stresses Relaxation

During in this study torsional stress is built up in a fastener as it tightened by conventional torque method, while in hydraulic tensioning method torsional stress value are all ways equal approximately zero. Parasite torsional stress in bolts arises during tightening due to friction between the contact surfaces in the nut threads and bolt. So high thread friction increases twisting of the bolts and causes yielding at lower clamp load levels than normal.

Torsional stress subject to varying amount of relaxation after conventional tightening complete, the results in static and dynamic parts shows that torsional stress relax, but still have influence magnitude to equivalent stresses in the bolt. Torsional relaxation depends on many factors; the amount and the rate of torsional relaxation will vary from bolt to bolt also from application to application.

The relation between torsional stress relaxation and external cyclic load for M14 tightened by the conventional torque tightening method shown in figure 5.18. The external cyclic load magnitude of 67100N applied with frequency 60cycle/sec for total number of 300.000 cycles to the structure. During tests, after every 50.000 cycles test is interrupted for a short duration and bolt torsion stress is measured by using FCT type strain gauges present on the bolt body.

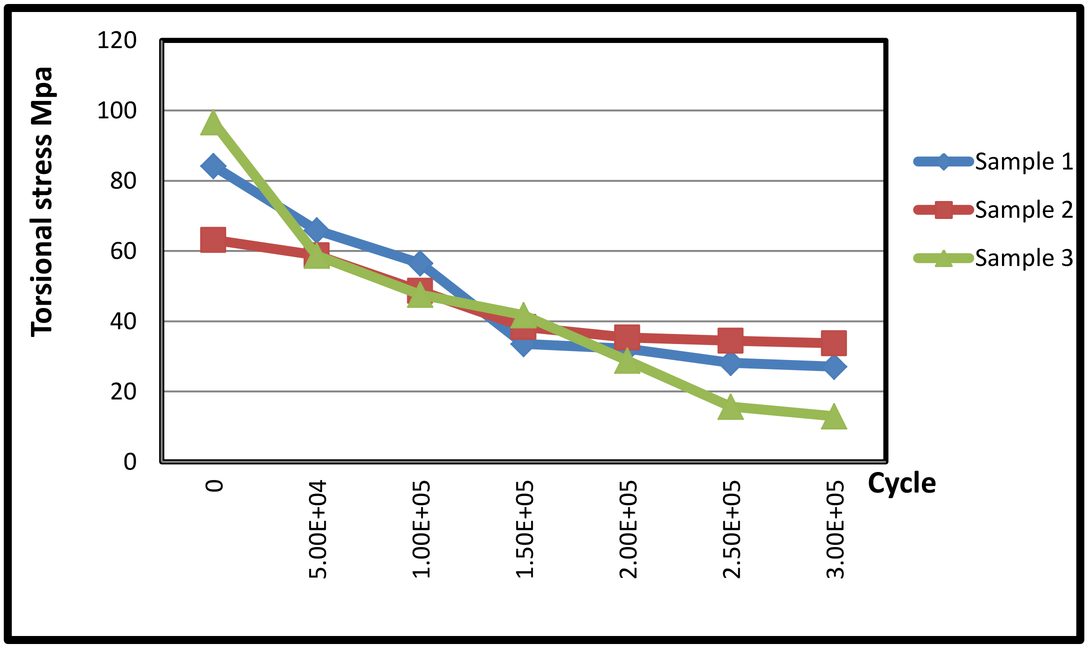


Figure5.18 Torsion stress reduction vs load cycle for M14 bolts tightened by the conventional torque tightening method

From the results of the cyclic loading effect on torsional stresses as shown in figure 5.18, initially there is a rapid drop off torsion stress, which has occurred as a result of embedding in the first 50,000 cycles, this behavior is clear in samples 1 &3, in sample 1 reduction percentage reached 31.5% and in sample 3 reached 45% of the total reduction percentage. In sample 2 torsion stress decreases regularly during 300,000 cycles, in the first 50,000 cycles the reduction percentage reached 13%, as mentioned before, the amount and the rate of torsional relaxation will vary from bolt to bolt.

CHAPTER 6

CONCLUSIONS

Although tightening bolts by the conventional torque wrench method is simple, especially for bolts of reasonable dimensions, hydraulic bolt tensioning, with better accuracy and no torsional stress, is now the preferred method of tightening bolts and studs on all critical applications. The quality of a bolted fastener assembly depends on two interdependent parameters, first the design of the assembly, and the second the method used to tighten the bolts.

The study presented here proves that the tightening method of hydraulic tensioner offers the best compromise in both preload accuracy during and after tightening compared with the method of the conventional torque wrench. The static and dynamic experimental results of both tightening methods are compared and the following conclusions are obtained in this study:

1. Deviations of preload after tightening complete from target preload in conventional torque tightening method reach as high as 40%, the average of all tightened bolts in this study, approximately 24%, while all deviations are below 10% in hydraulic tension method with a maximum of 9.6%, and the average of all tightened bolts approximately 4%. Analyzing final preload results in both tightening methods, it is clear that a hydraulic tension method is more accurate from conventional torque tightening method to reach the preload target level, because the most important parameter, namely the traction load, is perfectly controlled through the hydraulic pressure on the tensioner. The load does not depend on the various friction coefficients parameters in the assembly.

2. The hydraulic bolt tensioning method is simple like conventional torque tightening method, but it need more time comparing with the conventional torque tightening method.

3. Parasite torsional stress is built up in a fastener as it tightened by conventional torque tightening method, while in hydraulic tensioning method torsional stress value is all ways equal approximately zero.

4. In the conventional torque tightening method, equivalent total (Von-Mises) stress in the bolt has to be calculated by using both axial and torsion stress components, while in hydraulic tensioning method the equivalent stress is equal to axial stress.

5. Due to both large inaccuracy and also the parasite effect of the torque tightening method, equivalent stress may go near to yield stress and sometimes over the yield stress of the bolt, whereas the tension stress itself remains within admissible limits. The additional "parasite" torsion stress has big influence in bolt failure during tightening.

6. Some of the torsion stress may be released/relaxed by immediately spring back when conventional torque tightening complete. The amount of relaxation depends on the friction under the bolt head or nut with structure, so the relaxiation vary from bolt to bolt also from application to application.

7. All above behaviors of both tightening methods are achieved in M14, M16 and M18, so we can consider above behaviors as general threaded fastener behaviors.

8. Relatively large reductions in preload values (nearly 25-45%) for M14 are recorded in case of torque wrench method for a period of 300.000 cycles of constant dynamic loading, while for the same period of loading cycle, reduction values in preloads are much smaller (5-6%) for the case of hydraulic tension method.

9. The preload magnitude reduction in conventional torque tightening method more than hydraulic tension method under cyclic load effect. The embedment magnitude between contact surfaces which related to friction is the main reason of preload reduction.

10. Most of preload relaxations appear at beginning of effect dynamic load, generally there is a rapid drop of preload magnitude which has occurred as a result of embedding in the first 50,000 cycles in all samples for both tightening methods.

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APPENDIX A

LICOTA
Since:1983

CERTIFICATE OF CALIBRATION

SERIAL NO.&CERTIFICATE NO. 495732

PART NO NTP	SET TORQUE Nm	LOWER Nm	UPPER Nm	ACTUAL READING Nm
☐AQL-N2030 (1/4"DR)	6	5.76	6.24	
☐AQL-N3030 (3/8"DR)	18	17.28	18.72	
(30Nm)	30	28.80	31.20	
AQL-N3110	19	18.24	19.76	
(110Nm)	68	65.28	70.72	
	110	105.60	114.40	
AQL-N4210	40	38.40	41.60	
(210Nm)	120	115.20	124.80	
	210	201.60	218.40	
AQL-N4350	70	67.20	72.80	70.97
(350Nm)	210	201.60	218.40	208.71
	400	386.00	414.00	403.69
AQL-N6500	100	96.00	104.00	
(500Nm)	300	288.00	312.00	
	500	480.00	520.00	
AQL-N6700	140	134.40	145.60	
(700Nm)	420	403.20	436.80	
	700	672.00	728.00	
☐AQL-N6980 (3/4"DR)	140	134.40	145.60	
☐AQL-N8980 (1"DR)	560	537.60	582.40	
(980Nm)	980	940.80	1019.20	

The limits shown and the test equipment used for this calibration comply with the requirements of:

BSEN 26789 & ISO 6789 ASME B107.14M & DIN3122.

The accuracy of the test equipment used is ± 1%

TESTED BY: *MARTY*

INSPECTED BY: *KUAN-CHIN* DATE: *OCT-25-2012*

PLACE OF PURCHASE:

DATE OF PURCHASE:

Appendix B

Manufacture Calibration

Weight - Pressure Table of CFTMAK Hydraulic System.

Büyü Çap: 85 mm

Küçük Çap: 40 mm

Etki Alanı: 44,20 cm2

MANOMETER BASINÇ (BAR)	ETKİ ALANI	ELDE EDİLEN KUVVET (KG)
10	44.2	442
20	44.2	884
30	44.2	1326
40	44.2	1768
50	44.2	2210
60	44.2	2652
70	44.2	3094
80	44.2	3536
90	44.2	3978
100	44.2	4420
110	44.2	4862
120	44.2	5304
130	44.2	5746
140	44.2	6188
150	44.2	6630
160	44.2	7072
170	44.2	7514
180	44.2	7956
190	44.2	8398
200	44.2	8840
210	44.2	9282
220	44.2	9724
230	44.2	10166

BASINÇ (BAR)	ETKİ ALANI	ELDE EDİLEN KUVVET (KG)
240	44.2	10608
250	44.2	11050
260	44.2	11492
270	44.2	11934
280	44.2	12376
290	44.2	12818
300	44.2	13260
310	44.2	13702
320	44.2	14144
330	44.2	14586
340	44.2	15028
350	44.2	15470
360	44.2	15912
370	44.2	16354
380	44.2	16796
390	44.2	17238
400	44.2	17680
410	44.2	18122
420	44.2	18564
430	44.2	19006
440	44.2	19448
470	44.2	20774
480	44.2	21216
490	44.2	21658
500	44.2	22100
510	44.2	22542
520	44.2	22984
530	44.2	23426
540	44.2	23868
550	44.2	24310
560	44.2	24752
570	44.2	25194
580	44.2	25636
590	44.2	26078
600	44.2	26520