TAMPEREEN TEKNILLINEN YLIOPISTO
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TORQUE MOMENT – TENSION FORCE RELATION FOR ANCHOR BOLTS

Master of Science Thesis

Examiner: Doc. Kristo Mela
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ABSTRACT

VILLE PALVANEN: Torque moment – tension force relation for anchor bolts
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The main purpose of this thesis was to study the relationship between torque and tension for HPM and PPM anchor bolts. This was done by determining the torque coefficients and torque – tension curves. The impact of lubrication and corrosion were taken into account.

There is always some scatter in achieved preload. Scatter in friction, accuracy of tightening method, tolerances and operator skills are some of the factors that affects to the preload accuracy. Because the scatter, the minimum and maximum tightening torques are typically given by the manufacturers.

Torque coefficient describes the torque – tension relation and is determined experimentally. It includes things like the friction and is made to simplify and reduce the number of factors in joint calculations. But like preload, there is always some scatter in torque coefficient even when the bolts are from the same lot.

Tightening method selection should be based on cost optimization and circumstances. Highly accurate method can reduce the number of needed bolts or decrease dimensions but will be more expensive. It should be also decided how accurate method is needed. The costs increase unnecessarily if the high accuracy is chosen when it is not needed.

Test set-up included a bolt, two nuts, two washers, two column shoe base plates and two steel plates. The measurements were made by using hydraulic jack, hydraulic pump, torque wrenches and torque multiplier. Two lubricants were used, one is considered as a low friction lubricant and the other is considered as a basic lubricant. Some of the bolts were left outdoors to be exposed to the corrosion. Improvements to the tests were made gradually when new problems occurred.

The torque coefficient was 0,21 to HPM bolts. The results for PPM bolts are not public. The lubrication decreased, and corrosion increased the coefficient value. Corrosion had a major impact to the torque – tension relation. The bolt size didn't affect the torque coefficient, but it had a significant impact to the torque – tension curve. The curve analysis was easier to make with smaller bolts since the torque multiplier increased the scatter with larger bolts.

TIIVISTELMÄ

VILLE PALVANEN: Kiristysmomentin ja pulttivoiman välinen suhde ankkurointipulteissa
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Avainsanat: kiristysmomentti, pulttivoima, k-kerroin, ankkurointipultti

Työn päätavoitteena oli selvittää kiristysmomentin ja pulttivoiman suhde HPM- ja PPM-ankkurointipulteille. Tätä varten määritettiin k-kertoimet ja momentti – pulttivoimakäyrät. Lisäksi tutkittiin voiteiden ja ruosteen vaikutusta näihin molempiin.

Esijännityksessä ilmenee aina jonkin verran hajontaa. Tähän vaikuttaa mm. kitkakertoimen hajonta, kiristysmenetelmän tarkkuus, toleranssit ja kiristäjän taidot. Hajonnasta johtuen valmistajat usein ilmoittavat kiristysmomentille minimi- ja maksimiarvot.

K-kerroin kuvaa momentin ja pulttivoiman välistä suhdetta, ja se määritetään kokeellisesti. K-kerroin sisältää useita tekijöitä, esimerkiksi kitkan, ja sitä käytetään yksinkertaistamaan laskuja ja vähentämään muuttujien määrää. Kuten esijännityksen tapauksessa, myös k-kertoimessa esiintyy hajontaa jopa samassa pulttierässä.

Kiristysmenetelmän valinta tulisi perustua kustannusten optimointiin ja olosuhteisiin. Erittäin tarkoilla menetelmillä saatetaan säästää pulttien määrissä tai dimensioissa, mutta nämä menetelmät ovat myös kalliita. Jos erityisen tarkalle kiristykselle ei ole perusteltua tarvetta, kulut nousevat tarpeettomasti.

Yksi koejärjestely sisälsi pultin, kaksi mutteria, kaksi aluslevyä, kaksi pilarikengän pohjalevyä ja kaksi teräslevyä. Mittaaminen suoritettiin hyödyntämällä hydraulista tunkkia, hydraulista pumppua, momenttiavaimia ja momenttikerrointa. Kaksi voidetta oli käytössä: toinen oli esimerkki pienen kitkan voiteesta ja toinen oli esimerkki perusvoiteesta. Osa pulteista jätettiin ulos altistumaan korroosiolle. Testijärjestelyä paranneltiin sitä mukaan, kun uusia ongelmia ilmeni.

K-kertoimen arvoksi saatiin HPM pulteille 0,21. PPM pulttien osalta testitulokset päätettiin olla julkaisematta. Voide laski ja ruoste nosti k-kertoimen arvoa. Ruosteella oli erityisesti vahva vaikutus momentin ja pulttivoiman väliseen suhteeseen. Pultin halkaisija ei vaikuttanut oleellisesti k-kertoimeen, mutta momentti – pulttivoimakäyrään sillä oli suuri vaikutus. Käyrien analysointi oli helpompaa pienemmille pulteille, sillä suurempien pulttien kohdalla momenttikerroin lisäsi testipisteiden hajontaa.

PREFACE

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Tampere, 22.8.2018

Ville Palvanen

CONTENTS

LIST OF FIGURES

LIST OF SYMBOLS AND ABBREVIATIONS

AR	As Received
DTI	Direct Tension Indicator
F.E.D.S.	Fastenal Engineering & Design Support
L	Lubricated
R	Rusted
TUT	Tampere University of Technology

A_h	hydraulic jack cylinder effective area
A_s	nominal stress area
C	constant
d	nominal diameter of thread
d_m	mean diameter of thread
d_1	basic minor diameter of external thread
d_2	basic pitch diameter of external thread
d_3	minor diameter of external thread
E_1	modulus of elasticity at room temperature
E_2	modulus of elasticity at elevated temperature
F	bolt load
F_{etl}	bolt load with external tensile force
F_{ecl}	bolt load with external compressive force
ΔF_{ibf}	increase in bolt force
ΔF_{ijf}	increase in joint force
ΔF_{rbf}	reduction in bolt force
ΔF_{rjf}	reduction in joint force
F_p	preload
$F_{p,C}$	design preload
F_{p1}	preload at room temperature
F_{p2}	preload at elevated temperature
$F_{s,Rd}$	design slip resistance of preloaded bolt
F_u	ultimate force
f_{ub}	ultimate tensile strength
F_y	yield force
f_{yb}	yield strength
H	height of the fundamental triangle of the thread
k	torque coefficient
K	stiffness of the bolt
k_i	individual value of k-factor
k_m	mean value of k-factor
k_s	given parameter
Δl	bolt stretch
M_m	mean value of moments
$M_{r,1}$	torque reference value for k-class K1
$M_{r,2}$	torque reference value for k-class K2
n	number of friction surfaces
O_b	reference point for bolt at zero load

O_j	reference point for joint at zero load
p	pitch of the threads
p_h	pressure from the hydraulic pump monitor
R	ratio
r_b	effective thread contact radius
R_{high}	high-end ratio
R_{low}	low-end ratio
r_t	effective thread contact radius
T	torque
T_{at}	mean torque for AT-2000 Vaseline Spray
T_b	bearing surface friction torque
T_{gr}	mean torque for Molykote G-Rapid Plus paste
T_{high}	high-end input torque
T_{in}	input torque
T_{low}	low-end input torque
T_{max}	maximum tightening torque
T_{min}	minimum tightening torque
T_{out}	output torque
T_r	mean torque for rusted bolts
T_t	thread torque
T_u	ultimate torque
T_y	yield torque
V_k	coefficient of variation of the k-factor for the preload
V_{TOT}	total accuracy
$V_{TOT,MP+}$	total positive accuracy for torque multiplier
$V_{TOT,MP-}$	total accuracy
$V_{TOT,WR+}$	total positive accuracy for torque wrench
$V_{TOT,WR-}$	total negative accuracy for torque wrench
α	helix angle, thread angle
β	half of the thread profile angle
γ_{M3}	partial factor for a material property
θ_r	nut rotation
λ	lead angle
μ	slip factor
μ_b	friction coefficient between the turning head/nut and its bearing surface
μ_t	friction coefficient between threads
μ_{tot}	total friction coefficient

1. INTRODUCTION

What happens when we tighten bolted joints? When using a torque wrench, the torque is applied to the joint when we turn the wrench. The nut rotates, and the bolt stretches. This creates tension which is called preload. Preload creates clamping force and compresses the other members together in a joint. This provides stability and compactness to bolted joints.

Achieving the exact preload value in bolt is almost impossible. The reason is that the preload is dependent on hundreds of factors like the bolt condition, tightening method, operator skills and tightening sequence just to name a few. This problem leads to the fact that usually the manufacturer gives two tightening torques, minimum and maximum, within the actual tightening should be targeted. Minimum torque should be high enough so that even in the most unfavorable situation the joint will still work, and maximum torque should be small enough not to lead the joint into failure.

The torque coefficient takes into account such things as the friction and operator factors. The coefficient is determined experimentally and is used to describe the relationship between tightening torque and bolt tension. It would be desirable to have the same torque coefficient among all the bolts in the joint. However, there will be always scatter in torque coefficient, even between the bolts from the same lot. The friction is the main factor which affects the torque coefficient. Therefore, in order to have the smallest scatter in torque coefficient we need to make sure the friction scatter is the smallest possible. This is done by using lubricants or checking that the bolt condition is consistent.

There are many ways to control the preload and choosing the right tightening method is important part of the bolted joint design. Methods are based on the processes that happen during tightening. For example, torque, turn and stretch control are ways to control the preload in joint. By using the different methods and tools, we can achieve different tightening accuracies. Accurate tightening methods might be expensive but with provided accuracy, the bolts can be dimensioned lighter and cheaper. On the other hand, the use of a very accurate method, when it is not needed, increases the costs unnecessarily.

The main purpose of this thesis was to determine torque moment – tension relation to HPM and PPM anchor bolts. This was done experimentally by determining the torque coefficients and torque – tension curves for the bolts. Test specimens were made from HPM 20, HPM 24, HPM 30, HPM 39, PPM 30, PPM 39 and PPM 52 bolts. Some of the bolts were tested as received, some with lubrication and some were exposed to the corrosion.

The test set-up included a bolt, two nuts, two washers, two column shoe plates and two steel plates. The measurement was done by using hydraulic jack, hydraulic pump, torque wrenches and torque multiplier. The tests were made in Lahti between December 2017 and February 2018.

The theoretical section is presented in chapters 2 and 3. In Chapter 2 has been explained the behavior of bolted joints and the factors that have an impact to the achieved preload. In Chapter 3 has been presented the different methods to control the preload. Both chapters also include relevant sections from the Eurocode standards.

The experimental section is presented in chapters 4 and 5. Chapter 4 consists introductions about the test equipment and lubrications. Here also the test procedure, problems and improvements have been discussed. In Chapter 5, the test results have been calculated and analyzed.

The summary section is presented in chapters 6 and 7. In Chapter 6, some guidelines have been given about the factors that should be noted during the bolt design and tightening process. The final conclusions can be found from Chapter 7.

2. BOLTED JOINTS

Bolted joints are widely used because of the possibility to be assembled and unassembled easily. The joint usually includes bolts, nuts and washers. The bolt is clamping or/and pinning together two or more objects. Clamping force is more common and one of the most important forces to consider when designing bolted joints. Clamping force is achieved by preloading the bolt but this is not an easy task as we can see later.

Friction plays major role in preload accuracy. The friction coefficient is used as a parameter to describe the different condition or coatings on a bolt. It can be as high as 90% of the input torque which is used to overcome the friction so even small changes in friction can result a large variation of preloads [1]. Self-loosening, temperature and tightening sequences are among other factors that will affect to the preload accuracy.

2.1 Bolted joint

Let's start with the question of what is the difference between a screw and a bolt? Even though the answer is not unambiguous it can be usually defined with the presence or absence of a nut. The bolt comes with a nut and the screw does not. Bolt is tightened by torqueing the nut and screw is tightened by torqueing the head.

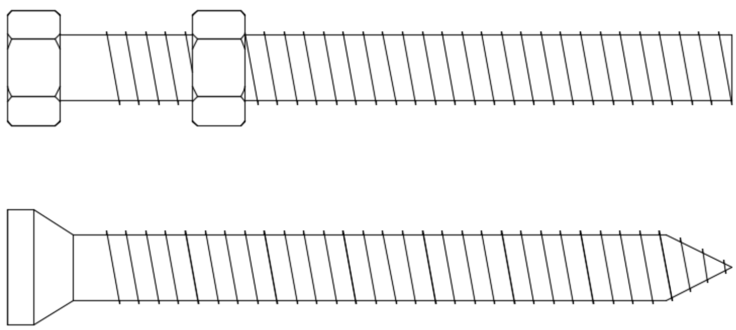

Figure 1. *A bolt (top) and a screw*

According to EN ISO 1891 [2], the concepts of a bolt and a screw are understood differently in North American and European markets as it has been stated in the following citation. “In the North American market, a “bolt” is understood as an externally threaded fastener to mate with a nut and to be nut-side tightened. In Europe, a “bolt” is an externally threaded fastener with a partly threaded shank. In the North American market, a “screw” is intended to be head-side tightened and to be mounted in threaded-through holes or blind holes. In Europe, a “screw” is an externally threaded fastener with a shank threaded up to the head.”

A bolt can attach the joint members together by two methods. It can prevent the joint members from slipping each other as a pin or it can clamp the members together as a spring. Most often the bolt will clamp the members together and this creates tensile stress to the bolt. In calculations, the model of tensile stress in Figure 2, is typically used. The assumption is that tensile stress is zero in both ends and threaded section has a little higher stress than elsewhere. The effective length can be assumed to be the length between the midpoints of a head and a nut or two nuts. The reality is far more complex, but we can use this model and still achieve an adequate accuracy at least when long and thin bolts are concerned. When bolt gets shorter and diameter larger, the assumption about effective length becomes more inaccurate as it can be seen from the Figure 3. [3]

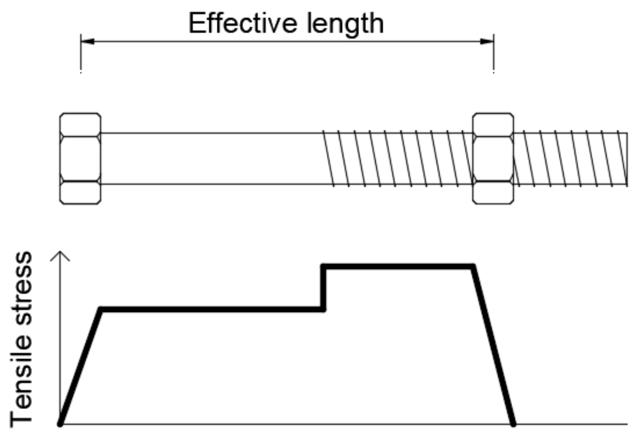

Figure 2. *The simplified model of tensile stress distribution in a bolt. Based on reference [3].*

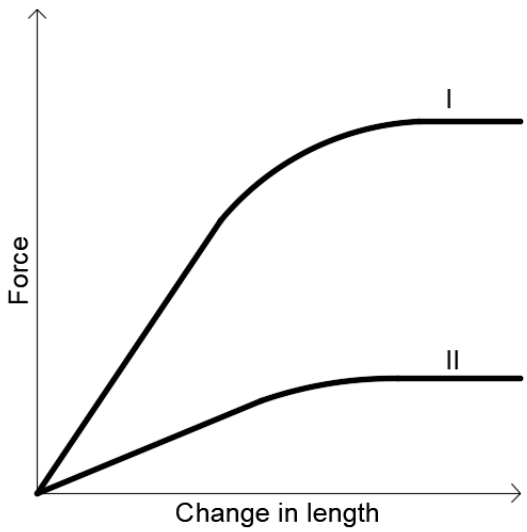

Figure 3. *The difference that the shape of the body makes. There are two curves, (I) is short and stubby bolt and (II) is long and thin bolt. In short and stubby bolt, the change in length is more difficult to estimate. Based on reference [3].*

According to EN ISO 898-1 [4], the nominal stress area for a bolt can be calculated from the equation

$$A_s = \frac{\pi}{4}\left(\frac{d_2 + d_3}{2}\right)^2 \tag{1}$$

where d_2 is the basic pitch diameter of external thread in accordance with EN ISO 724 and d_3 is the minor diameter of external thread. The minor diameter of external thread can be calculated from the equation

$$d_3 = d_1 - \frac{H}{6} \tag{2}$$

where d_1 is the basic minor diameter of external thread in accordance with EN ISO 724 and H is the height of the fundamental triangle of the thread in accordance with EN ISO 68-1. Nominal stress areas have been presented in Table 1.

Table 1. *Nominal stress areas A_s for bolts according to EN ISO 898-1 [4].*

Thread	**Nominal stress area**
M12	84,3
M16	157
M20	245
M22	303
M24	353
M27	459
M30	561
M33	694
M36	817
M39	976

Nominal value of the yield strength can be calculated from the bolt's ultimate tensile strength when using ISO standards. For example, let's take the bolt class 10.9. The first number (10) indicates the ultimate tensile strength which in this case is 1000 MPa. The latter number (9) indicates the parameter which the ultimate tensile stress needs to be multiplied to get the nominal value of yield strength. So, in this case it would be 1000 MPa * 0,9 = 900 MPa. Nominal values of the yield strength and ultimate tensile strength have been collected to Table 2 according to EN 1993-1-8.

Table 2. *Nominal values of the yield strength f_{yb} and the ultimate tensile strength f_{ub} for bolts according to EN 1993-1-8 [5].*

Bolt class	**4.6**	**4.8**	**5.6**	**5.8**	**6.8**	**8.8**	**10.9**
f_{yb} (N/mm^2)	240	320	300	400	480	640	900
f_{ub} (N/mm^2)	400	400	500	500	600	800	1000

From the values in Table 1 and Table 2 can be calculated the force needed to yield a bolt

$$F_y = f_{yb}A_s \tag{3}$$

and the force to reach the ultimate tensile strength

$$F_u = f_{ub}A_s \tag{4}$$

In Figure 4 are elastic curves. On the right, there is an illustration of what happens when we tighten a bolt past the yield. After reaching the yield point, permanent deformations start to occur. Additionally, the yield point will change into new location (from point A to point B) in the curve. Usually bolts can have numerous of yield points before breaking down. [3]

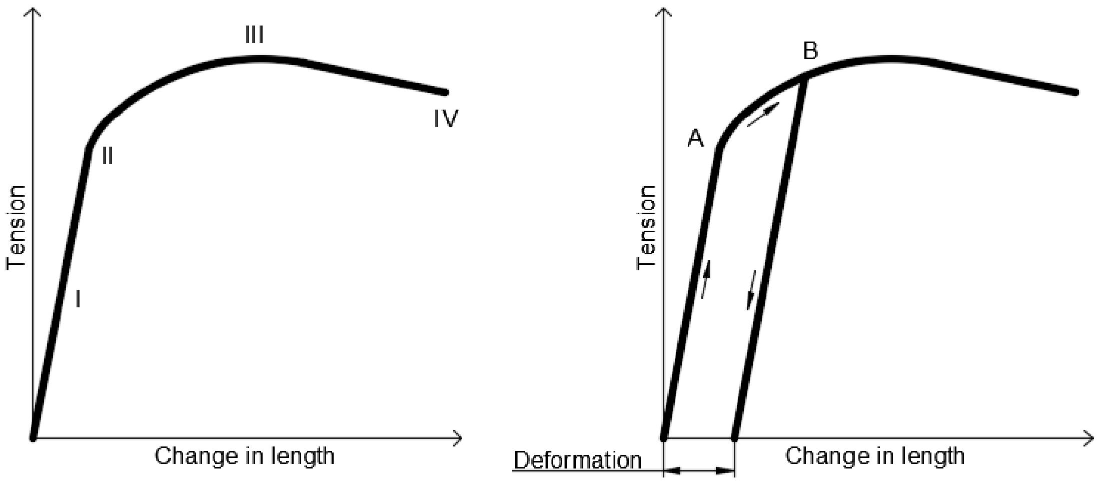

Figure 4. *Elastic curve and a bolt which has been tightened past yield point. Four points of interest: (I) elastic region, (II) yield strength, (III) ultimate strength and (IV) rupture. Based on reference [3].*

Besides the yield strength, different types of strengths should be considered when designing bolted joints. This includes thread stripping strength, shear strength, brittle fracture strength, strength in extreme temperatures, fatigue strength and stress corrosion cracking strength. Other factors that will affect the design of bolted joints are clamping force, materials, dimensions and tolerances for example. Clamping force is probably one of the most important factors as we can see later. [3]

2.2 Preload in bolted joint

As it was mentioned before, typically the bolt is clamping together two or more things and the joint needs a clamping force to work effectively. An inadequate clamp basically means an unreliable joint [3]. The clamping force is created when a nut is tightened, and the bolt starts to stretch. The stretch creates a tension force to the bolt and compression to the joint members. This load in a bolt is called preload. Without stretching, the bolt will not create any clamping force. And to be precise, it is not only the bolt but the bolt-nut-washer-system that creates the clamping force.

Preload can be divided into two phases according to Bickford [3], to the initial and residual preload. Initial preload occurs when the first tightening happens, and residual preload means the remaining tension at the end of the tightening process. Preload provides reliability and compactness to the joint [6]. It improves slip, seismic, fatigue and long-term resistance [5]. Preloaded bolt allows the service loads to transfer directly or using the increased friction in bolted joints [7]. Often the joint reliability is determined by the preload and how it changes over time [7].

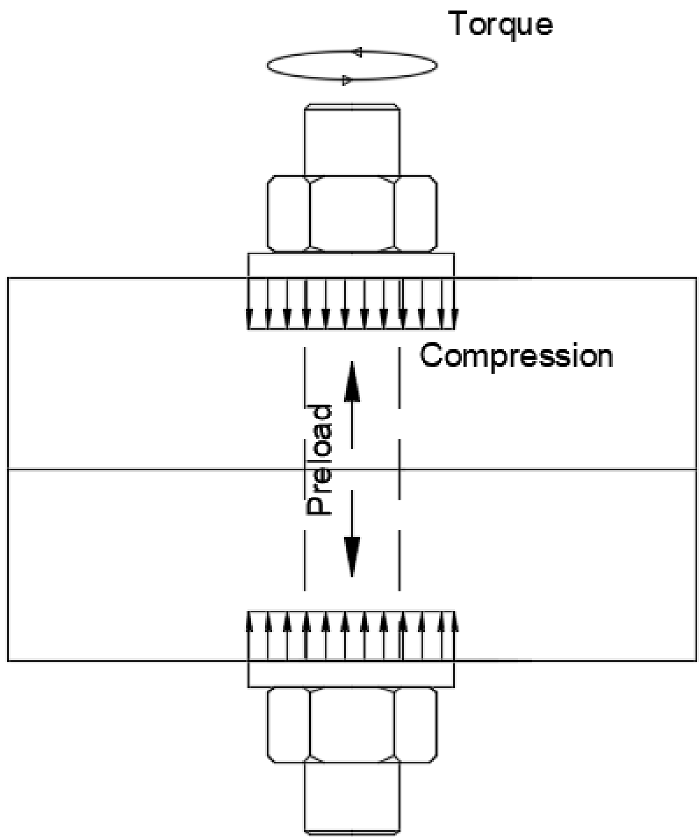

Figure 5. *Tightening creates preload to the bolt and compression force to the joint. Preload is typically equal and opposite to the compressive force in the joint.*

But what is an adequate amount of preload? That is not an easy question. The clamping force should be high enough to minimize self-loosening, load changes or joint slip and leaks. But the clamping force cannot be too high because of excessive stress and damages in the joint members. In addition, incorrect preload can create problems such as static failure, vibration loosening of the nut, fatigue failure and joint separation. It can also add the costs and need of materials since it increases the number of needed fasteners or dimensions. [3]

Besides problems with the incorrect preload, tightening an exact value of preload is found to be impossible since there will be always some uncertainties such as the friction coefficient, tool accuracy and operator skills [3]. This leads a scatter in preload and to the fact that to the desired preload has been often given minimum and maximum values. So, it is more like a range of preloads rather than a single preload value.

The minimum clamping force is determined by the accuracy of tools, the designed loads and how critical the joint is [3]. The material strength in connected parts, limits the preload which can be used. The stress can be reduced by reducing the preload or increasing the area of the nut. Usually the joints are overdesigned to make sure the minimum clamping force is achieved [3]. Figure 6 illustrates the benefits of high preload compared to the low preload. The maximum clamping force is typically determined by the yield strength of the bolt. In many applications, it is not desirable that the bolt starts to yield.

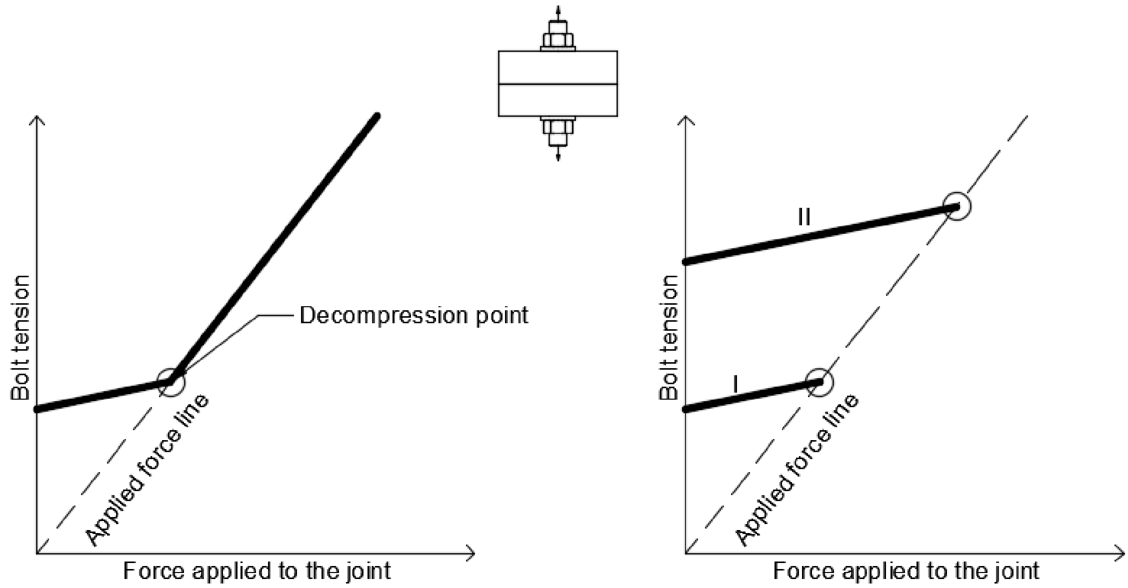

Figure 6. *When the applied tensile force increases it will meet the bolt's preload. This is called decompression point. Higher preload (II) increases the decompression point comparing lower preload (I). Based on reference [8].*

Another important factor is snugging. Snug torque is the torque required to pull the joint members together so that metal to metal contact occurs and snug point is reached. It should be taken care of that the snug torque is high enough to close the gaps. This is usually determined by experimentally. [8]

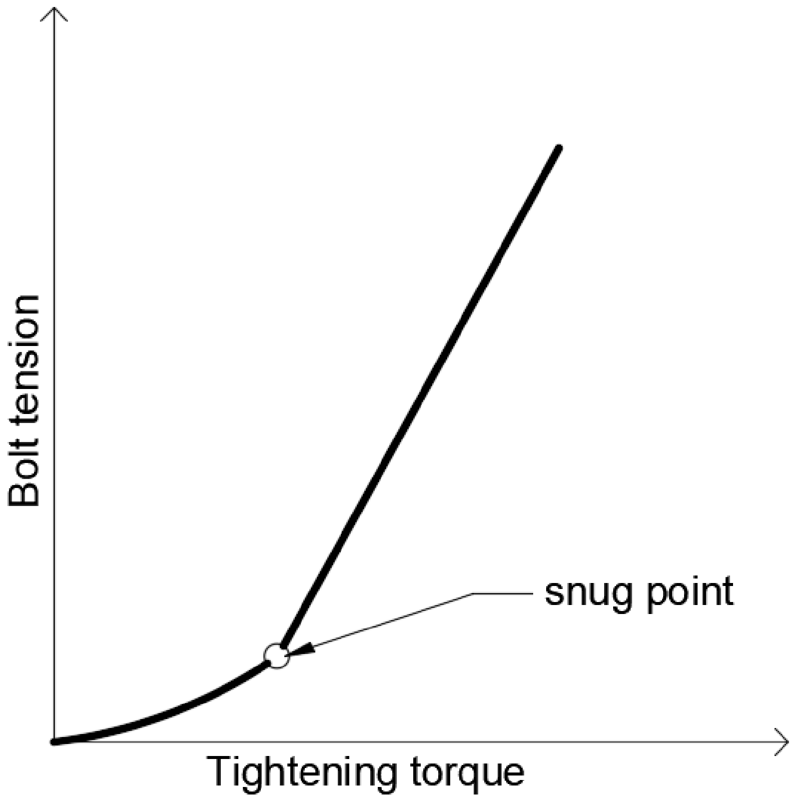

Figure 7. *Snug point in torque-tension relation*

Before reaching the snug point, the nut is loose and the torque – tension relation is not linear. The bolt tension starts to increase more rapidly when the nut contacts to the metal. After the snug point, the torque – tension relation is linear until something in the joint starts to yield. Finding the snug point is not an easy task and it varies a little in each bolt.

2.3 Friction in bolted joint

In physics we can divide friction into two categories, static and dynamic friction. Static friction works when the two objects are at rest in relation to each other. This is the force which resists the relative movement between these two objects. Dynamic friction occurs when the two objects are in relative movement of each other. Usually static friction is higher than dynamic friction. [8]

With threaded fasteners, the friction can be described by using the friction coefficient. It is dimensionless parameter and describes the friction without reference to the shape and dimensions of the bolt [9]. The friction coefficient can be divided into two categories, head and thread friction. Combining these two we can estimate the total friction coefficient.

Larger friction coefficient means that larger torque is needed to overcome it. This leads to the smaller preload with the same tightening torque. The friction coefficient has a large effect on torque – tension relation. Usually around 90% of the input tightening torque is used as overcoming the friction and the rest 10% is used as creating preload for unlubricated bolts. Because most of torque is needed to overcome the friction, even small differences in friction can lead large variation in preload. Friction coefficient is dependent on surface roughness, coating type and thickness, lubrication, material class, sliding speed and repeated use. [1]

Tightening and loosening a threaded fastener multiple times have an impact on the friction coefficient. W. Eccles, I. Sherrington and R.D. Arnell made an article in 2010 [10] where it was found that the friction coefficient will increase during the first four tightening before settling to the value of about twice of that in the first tightening. As a result, the clamp force reduced during the first four tightenings. These tests were made with zinc-plated fasteners. So, tightening multiple times changes the coefficient of friction and therefore the given friction and torque is relevant only during the first tightening.

Friction is important because it holds the nut tightened onto a bolt. On the other hand, when we reduce the friction, needed preload can be achieved with smaller torque. Additionally, the required preload is not achieved if the friction is too high. If it is too low the bolt will be overstressed. For these reasons, it is important to know the friction factors for adequate accuracy. The friction coefficient cannot be either under or overestimated. For static loads, it is better to keep the friction as small as possible to minimize the torsional stress. For dynamic loads the friction will prevent self-loosening. [10]

2.4 Joint diagram and external loads

The basic joint diagram has been presented in Figure 8. Joint diagram is used to illustrate the joint in service. O_b and O_j are the reference points for bolt (O_b) and joint (O_j) at zero load. When the force is applied, the bolt starts to stretch leading to the positive change in length while the joint starts to compress leading to the negative change in length. Usually, the joint is stiffer than the bolt and therefore the slope in bolt's side is less steep. It is common that the slope in the bolt side is around one-third to one-fifth of the slope in joint side. Preload (F_p) is used as reference force. [3]

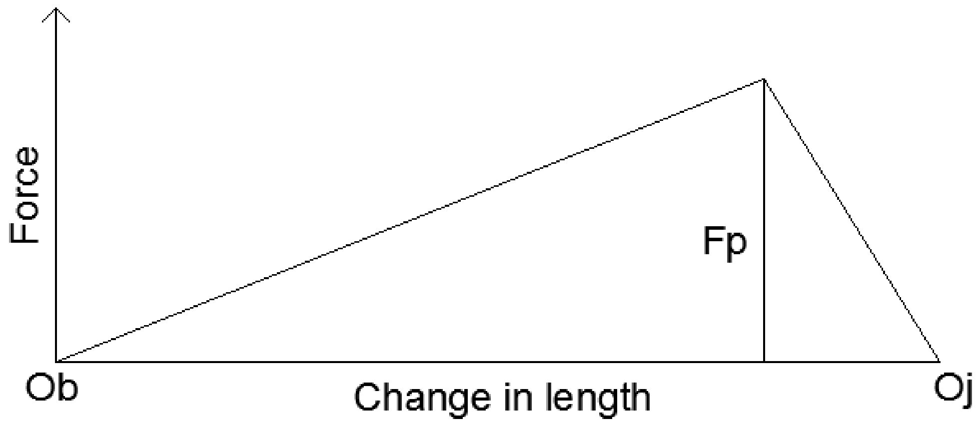

Figure 8. *Basic joint diagram. Based on reference [3].*

When the external force is applied, it will change the joint diagram [3]. In Figure 9, there has been applied external tensile force. As a result, both the tensile force in bolt and change in bolt's length will increase. In addition, the compressive force in the joint and the negative change in length will decrease. When the external tensile load affects the joint area, most of it is taken by the members around the bolt, relaxing the compression in joint members [11].

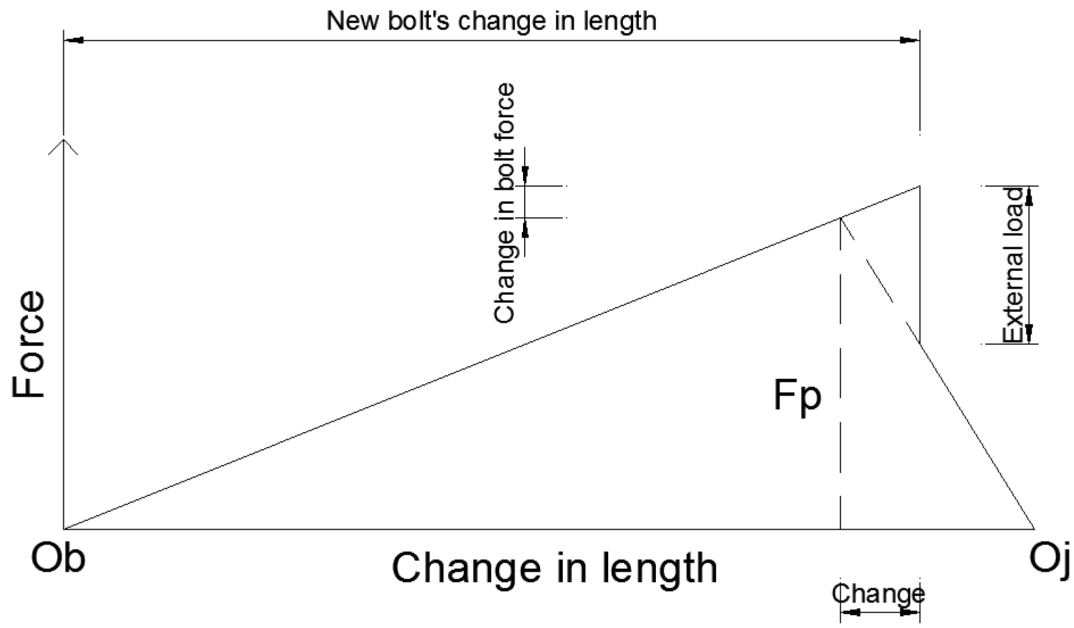

Figure 9. *External tensile force applied to the joint diagram. Based on reference [3].*

External load equals the changes in bolt and joint forces. With external tensile load, it can be calculated as

$$F_{etl} = \Delta F_{ibf} + \Delta F_{rjf} \quad (5)$$

where F_{etl} is the external tensile load, ΔF_{ibf} is the **increase** in bolt force and ΔF_{rjf} is the **reduction** in joint force. With external compressive force, the equation is

$$F_{ecl} = \Delta F_{rbf} + \Delta F_{ijf} \quad (6)$$

where F_{ecl} is the external compressive force, ΔF_{rbf} is the **reduction** in bolt force and ΔF_{ijf} is the **increase** in joint force.

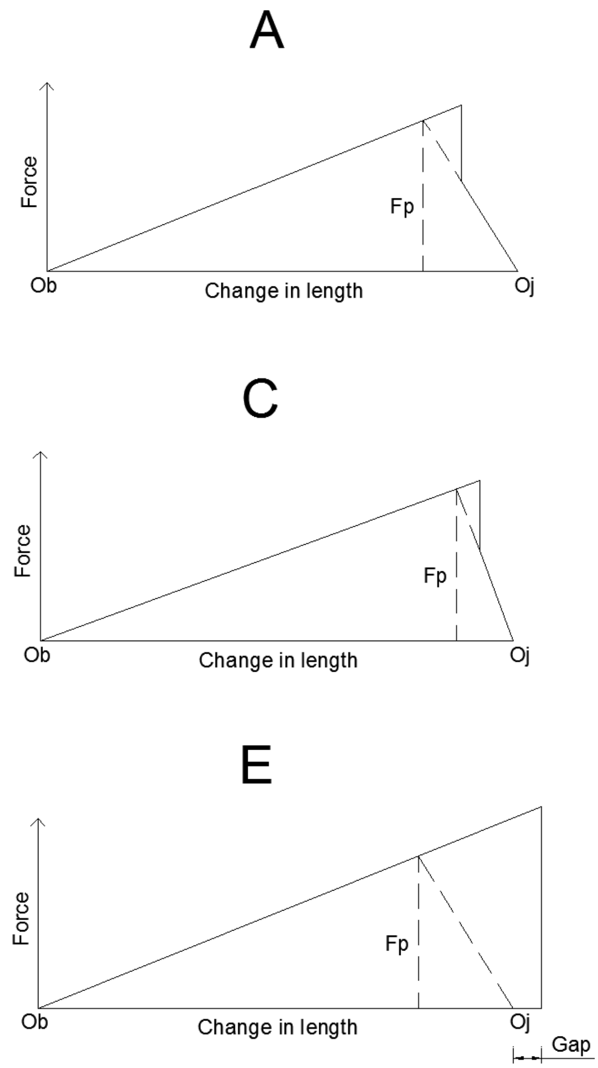

Figure 10. *Different types of joint diagrams. (A) external tensile load, (B) external compressive load, (C) hard joint, (D) soft joint, (E) critical load and (F) yielding point of bolt has been reached. Based on references [3, 8].*

Different types of joint diagrams can be found in Figure 10. Diagrams A and B show the difference between external tensile and compressive forces. While external tensile load increases the bolt length and force, the external compressive force does the other way around. With external compressive force, the bolt's change in length and bolt force reduces and the joint's negative change in length and force will increase. It can also be seen how the original preload value is reduced and the new preload is marked in figure as F_b. So, external loads can impact to the preload level in a bolt.

In diagrams C and D has been presented hard joint and soft joint. In hard joint, the slope with joint force is steeper than in normal joint. In soft joint, the slope with bolt force is steeper than in normal joint. In diagram E there is a critical tensile load that exceeds the compressive force in joint. This causes a gap between the joint members. In diagram F, the bolt force has exceeded the yield point.

2.5 Other factors affecting the amount of preload

Friction and external loads have an impact to the achieved preload. However, there are hundreds of factors playing its role to achieving or maintaining the desired preload in a bolt. These things are such as relaxation, self-loosening, temperature changes, tool accuracy, tightening sequences and bolt material class. The forces in a bolted joint, including clamping force, will change over time and the behavior during assembly process will affect the joint's performance in service. [3]

Fasteners may start loosening if there are dynamic loads and this will reduce the preload in joint [12]. It is caused by slip between the threads or joint members [3]. Self-loosening can happen due to creep, embedment, stress relaxation and vibration loosening [8]. From these creep, embedment and stress relaxation will not completely loosen the bolt because the loosening happens without a nut rotation. With vibration loosening, usually a nut starts to rotate.

Short-term relaxation happens after initial tightening and occurs as an initial loss of tension. Embedment is the most common reason for this and the loss can be 10-20% or sometimes even higher if soft gaskets or conical joints are involved. Some other factors that will affect the short-term relaxation are bolt length, number of joint members and tightening speed. With longer bolts the percentage of relaxation is smaller than with smaller bolts. Additionally, increasing the joint members may increase relaxation. Bolts need some time to settle and if tightening happens too fast the relaxation increases. The amount of relaxation is usually determined by experimentally. In order to get back to the original preload, the joint needs to be retighten after initial losses. [3]

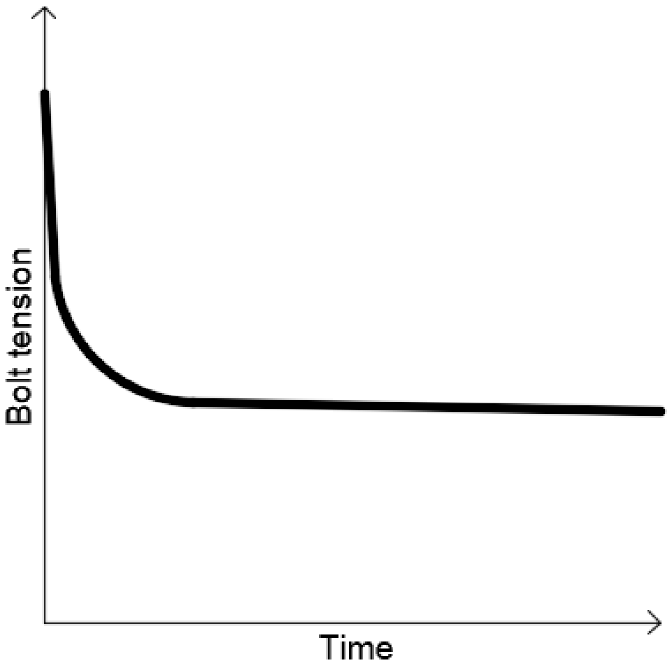

Figure 11. *Bolt tension over time. Most of the relaxation happens during the first seconds or minutes but usually it will never stop completely. Based on reference [3].*

When there are multiple bolts in a joint, the tightening sequence has an impact to preloads unless the bolts have been tightened simultaneously. Let's take an example, where we have a joint with multiple and identical bolts. This is based on Bickford's book [3] where the equivalent example was introduced. At first, we tighten the first bolt to the desired preload. When we tighten the second bolt the first one will relax and loose some of the initial preload. And again, when we tighten the third bolt the first two bolts will relax a bit more. As a result, we have now three bolts with three different residual preloads, even though the initial preload applied to each bolt was the same.

Temperature changes will also affect the current preload in bolt [3]. When the bolt temperature rises, the modulus of elasticity decreases. The preload in a bolt can be assumed to decrease in a same ratio and is estimated as:

$$F_{P2} = F_{P1}\frac{E_2}{E_1} \tag{7}$$

where F_{P2} is the preload at elevated temperature, F_{P1} is the preload at room temperature, E_2 is the modulus of elasticity at elevated temperature and E_1 is the modulus of elasticity at room temperature.

Usually there is an equal and opposite relationship between the tension and clamping force. But sometimes this is not the case as can be seen in the Figure 12. There are two examples when the tension in a joint is actually less than tension in a bolt. In case A, there is a tight fit and it will take some force to push the bolt through the hole. This will lead to the difference in a tension between the joint and the bolt. In case B, there is tightening from below the joint. It is not common situation but may occur sometimes. In this case the weight of a lower half will create tension to the bolt even when there is zero clamping force. [3]

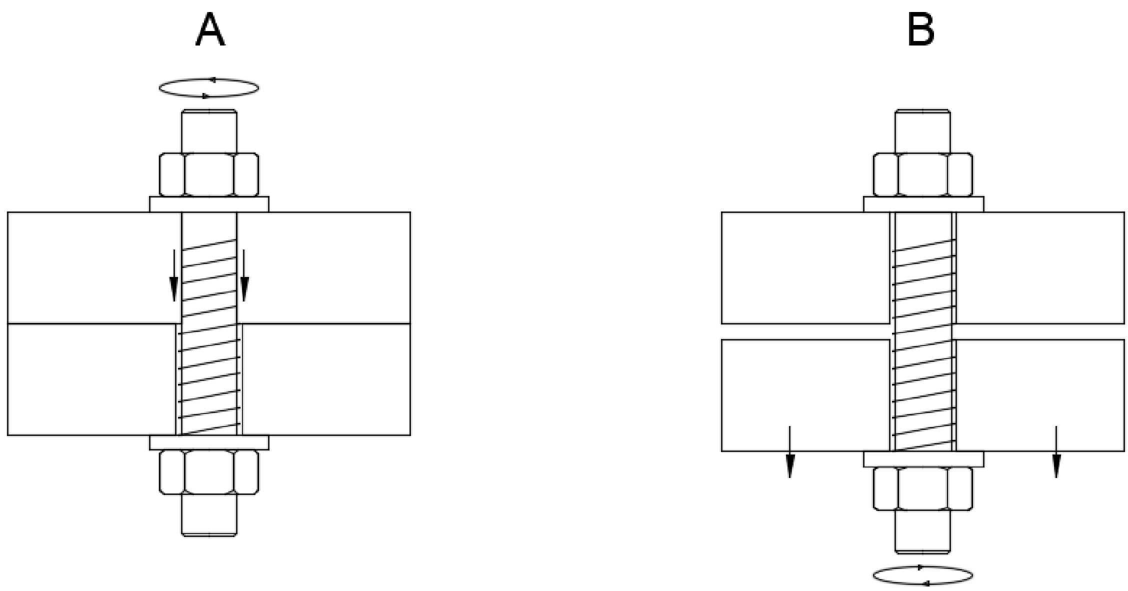

Figure 12. *Bolt joint with a tight fit (A) and tightening from below (B). In these cases, the tension in a joint is less than tension in a bolt. Based on reference [3].*

Bolt’s dimensions and material class will affect the preload design. With stronger bolts, stronger preload can be used. Furthermore accuracy, tolerances and quality of tools and parts will be factors when achieving desired preload.

2.6 Bolted joint in Eurocode

According to SFS-EN 1993-1-8 [5] bolted joints can be categorized into five categories as listed in below.

Categories of bolted shear connections

- Category A: Bearing type
- Category B: Slip-resistant at serviceability limit state
- Category C: Slip-resistant at ultimate limit state

Categories of bolted tension connections

- Category D: non-preloaded
- Category E: preloaded

According to EN 1993-1-8 [5] only bolt classes 8.8 and 10.9 should be used with preload. The preloading force is

$$F_{p,C} = 0{,}7 f_{ub} A_s \tag{8}$$

where $F_{p,C}$ is the design preload, f_{ub} is the ultimate tensile strength from Table 2 and A_s is the nominal stress area from Table 1.

Table 3. *$F_{p,C}$ value using Equation 8.*

Bolt class	Bolt diameter									
	12	**16**	**20**	**22**	**24**	**27**	**30**	**33**	**36**	**39**
8.9	38	70	110	136	158	206	251	311	366	437
10.9	59	110	172	212	247	321	393	486	572	683

According to EN 1993-1-8 [5] the design slip resistance of preloaded bolt is

$$F_{s,Rd} = \frac{k_s n \mu}{\gamma_{M3}} F_{p,C} \tag{9}$$

where k_s is given parameter in Table 4, n is the number of friction surfaces and μ is the slip factor given in Table 5.

Table 4. *k_s parameters [5]*

Description	**k_s**
Bolt in normal holes.	1
Bolts in either overzised holes or short slotted holes with the axis of the slot perpendicular to the direction of load transfer.	0,85
Bolts in long slotted holes with the axis of the slot perpendicular to the direction of load transfer.	0,7
Bolts in short slotted holes with the axis of the slot parallel to the direction of load transfer.	0,76
Bolts in long slotted holes with the axis of the slot parallel to the direction of load transfer.	0,63

Table 5. *Slip factors [5]*

Class of friction surfaces	μ
A	0,5
B	0,4
C	0,3
D	0,2

According to SFS-EN 1993-1-8 [5], if a slip-resistant connection is subjected to an applied tensile force in addition to the shear force tending to produce slip, the design slip resistance per bolt should be taken as follows for category B connection:

$$F_{s,Rd,ser} = \frac{k_s n \mu (F_{p,C} - 0{,}8F_{t,Ed,ser})}{\gamma_{M3,ser}} \tag{10}$$

and for category C connection:

$$F_{s,Rd} = \frac{k_s n \mu (F_{p,C} - 0{,}8F_{t,Ed,})}{\gamma_{M3}} \tag{11}$$

Some additional excerpts from the Eurocode that are relevant to the bolted joints and preload.

- According to SFS-EN 1993-1-8 [5]: “The bolt which has tightened to the smallest required preload and then loosened shouldn’t be used anymore after this.”
- According to SFS-EN 14399-7 [13]: “Preloaded bolts are very vulnerable for manufacturing changes and that’s why it is important that one manufacturer is responsible for the whole connection.”
- According to SFS-EN 1090-2 [14]: “For non-preloaded bolt joints, the snug tight is enough and it can be accomplished by a regular spanner without an extension arm.”

3. TIGHTENING METHODS

Understanding the circumstances is essential when choosing the right tightening method. The more accurate preload leads to lighter and cheaper solutions because there are less uncertainties. However, this also leads to more expensive tightening methods. Choosing the right method is often based on cost optimizing. The other main thing to be consider is the space where tightening is done. In tight spot, the lack of space will be a significant factor when choosing the right method.

The tightening process includes the following factors [3]. The **torque** is applied to the nut, the nut **turns**, the bolt **stretches** and the **preload** is created. Therefore, the preload can be controlled by based on different factors.

Tightening methods that are based on the tightening factors:

- Torque Control
- Turn Control
- Combined Control

Tightening methods that are based on the bolt factors:

- Stretch Control
- Direct Preload Control
- Ultrasonic Control

The easiest and cheapest ways to control the preload are the tightening methods that are more based on tightening factors. The most commonly used tightening method is torque control. With these methods, however, also the least accurate outcome is achieved. But in most cases the accuracy is still adequate. The tightening methods that are more based on bolt factors directly, are more accurate but will be also more expensive. In Table 6 has been presented control accuracies. This means the accuracy of which the control method produces the bolt tension [3].

Table 6. *The accuracies of different tightening methods according to Bickford [3].*

Tightening accuracy	
Method	**Accuracy**
Torque control with hand wrench	±30
Torque wrench plus multiplier	-70 to +150
Turn-of-nut	±15
Strain gaged load washers	±15
Strain gaged bolts	±1
Air powered impact wrench	-100 to +150
Hydraulic tensioners with vernier gage read-out	±20
Bolt stretch (micrometer)	±3 to 15
Ultrasonic control	±1 to 10

3.1 Torque Control

As Bickford [3] states, usually the relationship between torque and preload is linear between the snug and yield point. This can be presented with a constant:

$$torque = preload * constant \tag{12}$$

$$T = F_p * C \tag{13}$$

Hao Gong, Jinhua Liu and Xiaoyu Ding [15] have presented some analytical expressions for the constant. The original sources for the equations below are from Motosh and Nassar SA and Yang X.

$$T = F_p \left(\mu_b r_b + r_t \frac{\tan\alpha + \mu_t \sec\beta}{1 - \mu_t \sec\beta \tan\alpha} \right) \tag{14}$$

$$T = F_p \left(\mu_b r_b + \frac{p}{2\pi} + \frac{\mu_t r_t}{\cos\beta} \right) \tag{15}$$

$$T = F_p \left(\mu_b r_b + r_t \frac{\tan\alpha + \mu_t \cos\alpha \sqrt{1 + \tan^2\alpha + \tan^2\beta}}{1 - \mu_t \sin\alpha \sqrt{1 + \tan^2\alpha + \tan^2\beta}} \right) \tag{16}$$

where T is the input torque, F_p is the preload, μ_b is the friction coefficient between the turning head/nut and its bearing surface, μ_t is the friction coefficient between threads, α is the helix angle, β is the half of the thread profile angle, r_b is the effective thread contact radius, r_t is the effective thread contact radius and p is the pitch of the threads.

These equations seem to be quite complex. However, there is one more simple way to express the constant:

$$T = F_p d k \tag{17}$$

where T is the input torque, F_p is the achieved preload, d is the nominal diameter of thread and k is a non-dimensional torque coefficient. Sometimes this expression can be found too inaccurate, especially if the critical joints are concerned because the scatter in torque coefficient is too great [7]. Even if the perfect torque has been achieved, there can be 20-30% variation in preload [3].

It should be noted that the torque coefficient is not the same thing as the friction coefficient. The torque coefficient is determined experimentally, and it can differ in laboratory and construction site. Torque coefficient sums up things like friction, operator skills and tool accuracy [3]. For example, higher friction means higher torque coefficient. But the exact torque coefficient is impossible to determine and there will be always some scatter. This leads to the scatter in preload as well. Torque coefficient can be only used in calculations with the same friction conditions and geometric proportion [9]. Also, the names of nut factor or k-factor are widely used.

For example, in RakMK B7 from year 1996 (has been repealed since), the torque coefficient of 0,18 was used under label of partly oily coating [16]. Bickford has also presented some torque coefficients but only as a reference. Some of these are presented in Table 7.

Table 7. *Torque coefficients for some of the bolt conditions [3].*

Torque coefficients			
Bolt Condition	**Min**	**Mean**	**Max**
Non-plated/As received	0,158	0,2	0,267
Cadmium-plated	0,106	0,2	0,328
Zinc-plated	0,075	0,295	0,53
Black oxide	0,109	0,179	0,279
Machine oil	0,10	0,21	0,225
Moly paste or grease	0,10	0,13	0,18

Torque coefficients can vary a bit in different sources. For example F.E.D.S. (Fastenal Engineering & Design Support) have determined torque coefficients for non-plated as 0,20-0,30, for cadmium-plated as 0,11-0,15 and for zinc-plated as 0,17-0,22 in their laboratory [17].

As it was mentioned before, the torque coefficient is typically determined experimentally. However, J.P. Shigley and L.D. Mitchell [18] have presented an equation for the torque coefficient. The equation is

$$k = \left(\frac{d_m}{2d}\right)\left(\frac{\tan\lambda + \mu_t \sec\alpha}{1 - \mu_t \tan\lambda \sec\alpha}\right) + 0{,}625\mu_b \tag{18}$$

where d_m is the mean thread diameter, d is the nominal bolt diameter, λ is the lead angle, μ_t is the friction coefficient in threads, α is the thread angle and μ_b is the friction coefficient under head or nut. Both friction coefficients are typically about 0,15 and therefore the torque coefficient would be around 0,2 [18].

If the torque – tension relation scatters too much the accuracy of the preload decreases. Usually minimum and maximum torque values are determined to a bolt. These are dependent on desired minimum and maximum preload values and the scatter in the torque coefficient. The minimum and maximum torque should be designed so that still with the highest torque coefficient the minimum preload can be achieved and on the other hand with the lowest torque coefficient the maximum preload won't be exceeded. The principles are shown in Figure 13.

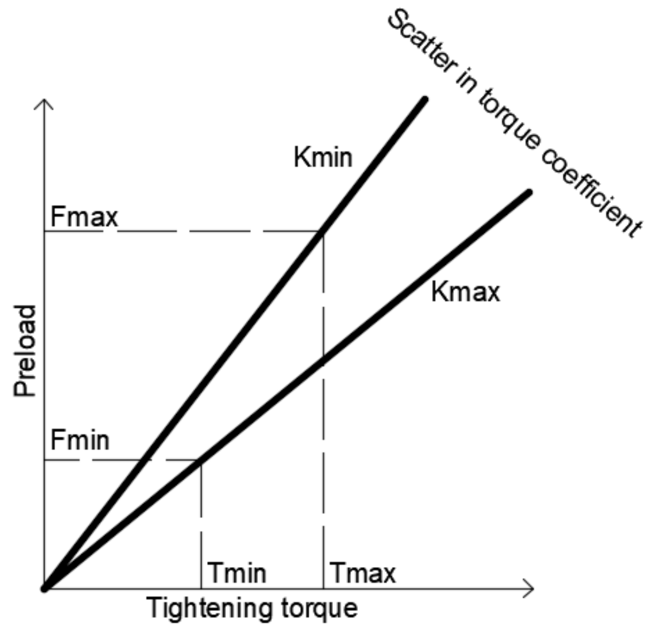

Figure 13. *An illustration of how to define the minimum and maximum tightening torque. Based on reference [19].*

Untightening the nut immediately after the tightening process, needs usually around 20% less torque than during the tightening [8]. After this, the needed untighten torque increases and after about a half day, it can be 10% higher than the tightening torque. Reasons for this are embedment and changes in friction. Even with its uncertainties, the torque control is the most commonly used method [15] and can provide adequate accuracy in most cases

[3]. Its popularity can be explained by low costs and simple tools. In addition, many times the torque control is the only possible choice.

3.2 Turn Control

Turn control can be also called angle control or turn-of-the-nut control. According to Bickford [3], there is a relation between the bolt stretch and the pitch of the threads:

$$\Delta l = p\frac{\theta_r}{360} \tag{19}$$

where Δl is bolt stretch, p is the pitch of the threads and θ_R is nut rotation. This could be lead to the preload by equation:

$$F_p = K\Delta l = Kp\frac{\theta_r}{360} \tag{20}$$

where K is the stiffness of the bolt. However, this equation does not work. For example, if there is a loose nut it will not create any preload when the turn has been made. After a snug point, the turn will create preload in linear measure. But when relative displacement between the nut and bolt is still one pitch in one turn, not all of it is bolt elongation. It is difficult to define what the proportion of bolt elongation is, and it will be also dependent on the joint type. [3]

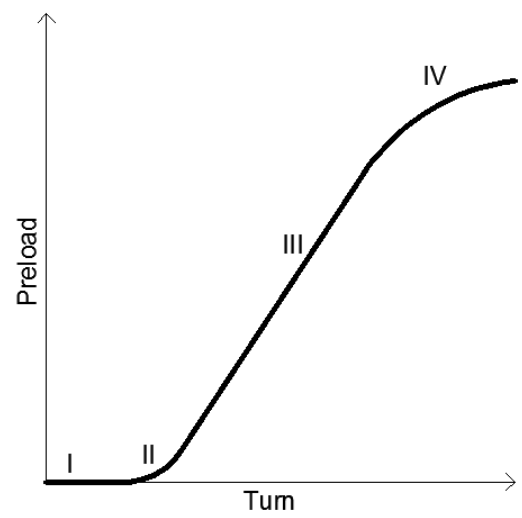

Figure 14. *Common preload - turn relation and the four phases of tightening process. Based on reference [3].*

In Figure 14 has been presented the most common behavior of turn – preload relation. (I) Before the snug point, turning nut doesn't create any preload. (II) Snugging pulls joint parts together and preload starts to increase. (III) All the parts in joint are working elastically and the increase of preload is linear. (IV) Something in joint starts to yield and the increase of preload starts to slow down.

The figure is a good tool but has some limitations, like defining snug point and turn. As it was mentioned in Subchapter 2.2, the snug point is difficult to determine exactly, and it will always scatter a little because of tolerances and differences in friction. Other problem is that we could ignore the friction only if it is possible to measure the relative turn between nut and bolt. Usually this is only possible in laboratory and in construction site the reference point is often a floor or other structure. That is why we need to take the friction into account and it will have an effect to the linear part. As a result, we have many possibilities for the curve and there will always be some scatter between the bolts. The accuracy of turn control decreases. [3]

3.3 Combining Torque and Turn Control

Combined control is based on the fact that the torque is better with initial variables like snugging, but the turn is better way to control the preload after snug point. Therefore, when using torque and turn together to control preload it is possible to use the benefits from both. As a result, torque control is used until we can make sure that joint is snugged and for the final control, we can use turn to create the desired preload to the bolt. [3]

When measuring both torque and turn simultaneously, there is more information available. This is the other reason why combined control is more accurate than just using either torque or turn alone. Measuring both can help to detect problems in the joint. For example, if the threads are galled it takes more torque to make a certain turn than usual, and from this the operator or machine can deduce that there is a problem in the joint. But it cannot detect all the problems and sometimes two problems at the same time could overturn each other like a soft bolt and a high-friction. [3]

Additionally, torque – time control can be used and it is usually cheaper than the torque – turn control. Instead of turn, it will use time as a measure to torque for desired level. Torque – time control will also detect if there are problems in a joint, but it won't give much more accurate results than torque control alone, when there are no problems. [3]

3.4 Stretch Control

Stretch control eliminates many of the problems we faced with torque or turn control such as the friction coefficient and relative turn. According to Bickford [3], the relationship can be described as:

$$\Delta l = F_p \frac{1}{K} \tag{21}$$

where Δl is the change in length of the bolt, F_p is the preload and K is the stiffness of the bolt.

The accuracy of preload is the same as the accuracy of stretch measure. One benefit with stretch control is the possibility to monitor the development of the preload. This could be done even after the tightening process by comparing the initial length to the current length. But there are still some problems that will affect the accuracy of stretch control like grip, dimensional tolerances of the bolt, change in temperature, variations in elastic modulus, plastic deformation (if the tightening is past yield) and bending. [3]

Most often, the stretch control will provide much more preload accuracy than torque and turn control and we do not need to measure the torque or turn at all. However, it will be more expensive and that is why it should be carefully considered if it is needed or not. In addition, the stretch control will not detect the problems in a joint like combined control could. [3]

3.5 Direct Preload Control

In most cases, a direct preload control is not possible. There are some techniques, however, which come close to that. Some of these examples are in Figure 15. DTI (Direct Tension Indicator) washer has bumps which starts to yield plastically when tightening happens and this reduces the gap between the bolt and the washer. Then the gap is measured and the preload is calculated with some accuracy. On the other hand, DTI bolt has a flange head. When tightening the wavy head starts to flatten. The amount of tension needed to flatten the head is known. Unlike with DTI washer, the deformation is elastic and therefore if the nut starts to loosen after tightening, the head starts to gain its wavy shape again. This can be noticed and the bolt is retightened. [3]

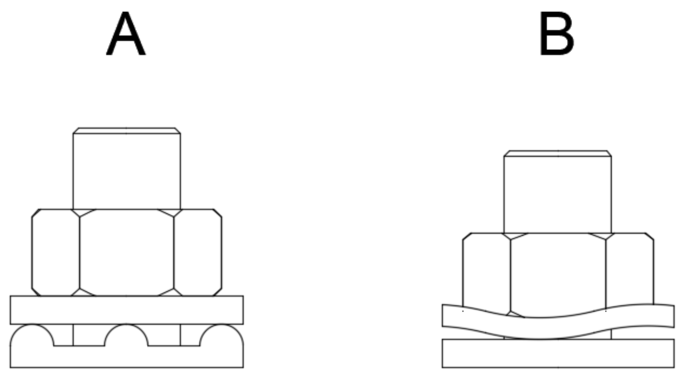

Figure 15. *Direct tension indicator washer (A) and bolt (B). DTI washer has bumps on its surface and DTI bolt has a flange head. Both bumps and a flange head will indicate the amount of preload in the joint when tightening. [3]*

3.6 Ultrasonic Control

Typically, an ultrasonic control device sends a brief burst of ultrasound to a bolt. The burst travels through the bolt and bounce back from the far end. This is called transit time. The device will keep on track the change in transit time during the tightening. There are basically two reasons why the transit time increases. This happens when bolt stretches (the distance increases) or the average velocity of sound decreases in material (the average stress level has increased). Both have a linear relationship with preload. [3]

At first, the bolt length is measured at zero stress and this is considered as initial length. When bolt is loaded, the length is measured again and the difference between the initial length and current length is used to estimate the current preload. Sometimes this can be done during tightening process to give a real-time estimation of preload. The device needs calibration data like temperature and bolt geometry to give an accurate estimation. This method is usually used in critical joints with a few bolts, calibrating other tightening methods and in laboratory environments. Bolts with only small amount of stretch are not ideal with this method (short bolts). [3]

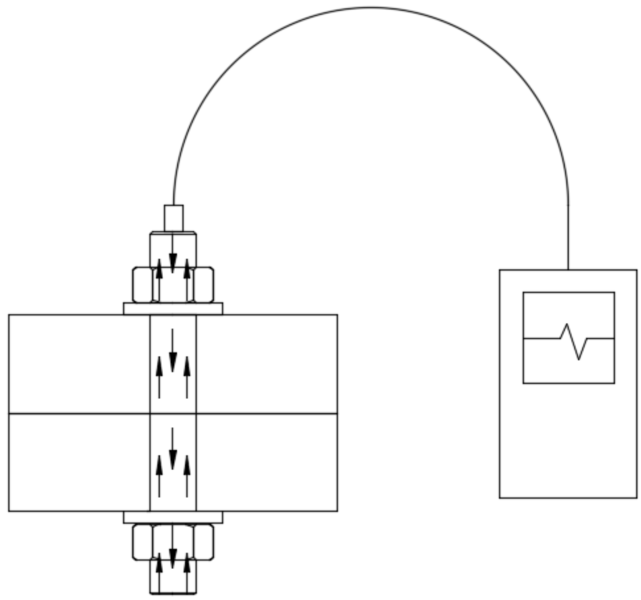

Figure 16. *An illustration of how the ultrasonic device works.*

3.7 Tightening methods in EN 1090-2

According to SFS-EN 1090-2 [14], the tightening should start from the most rigid part of the joint towards the least rigid part. Torque wrenches used in torque method should have an accuracy of ± 4%. If the torque wrench is used only in the first step of the combined method, an accuracy of ± 10% is enough. Checking the torque wrench is needed after any incident like significant impact, fall or overloading for example. Other tightening methods should be calibrated according to the equipment manufacturer.

Table 8. *K-classes for tightening methods according to SFS-EN 1090-2 [14].*

Tightening method	**k-classes**
Torque method	K2
Combined method	K2 or K1
HRC tightening method	K0 with HRD nut only or K2
Direct tension indicator (DTI) method	K2, K1 or K0

Torque reference values are according to SFS-EN 1090-2 [14] for k-class K2 as

$$M_{r,2} = k_m d F_{p,c} \tag{22}$$

and for k-class K1 as

$$M_{r,1} = k_m d F_{p,c} \tag{23}$$

Determined according to Annex H in SFS-EN 1090-2 [14]

$$M_{r,test} = M_m \tag{24}$$

where k_m is the parameter from Table 9, d is the bolt diameter, $F_{p,c}$ is the preloading force using the Equation 8 and M_m is determined according to the procedure relevant to the tightening method to be used. In Table 9 k_i is the individual value of k-factor, k_m is the mean value of k-factor and V_k is the coefficient of variation of the k-factor for the preload.

Table 9. *k-factors [20]*

k-class	**k-factor**	
K0	-	
K1	$0{,}10 \leq k_i \leq 0{,}16$	
K2	$0{,}10 \leq k_m \leq 0{,}23$	$V_k \leq 0{,}06$

When using the torque method, tightening should be continuous and smooth. Tightening should have at least two steps. During the first step, the wrench should be set to a torque value of about $0{,}75 M_{r,2}$ or $0{,}75 M_{r,test}$. All the bolts in a joint should be tightened to the first step before starting the second step. During the second step, the wrench should be set to a torque value of $1{,}10 M_{r,2}$ or $1{,}10 M_{r,test}$. [14]

When using the combined method, the first step is the same as with the torque method. The wrench should be set to a torque value of about $0{,}75 M_{r,2}$, $0{,}75 M_{r,1}$ or $0{,}75 M_{r,test}$ and all the bolts in a joint should be tightened to the first step before starting the second step. The second step is done based on turn and the Table 10. [14]

Table 10. *Combined method, additional rotation during the second step of tightening according to SFS-EN 1090-2 [14]. Total nominal thickness "t" includes all packs and washers. The "d" is bolt diameter.*

Total nominal thickness of parts to be connected	**Further rotation**	
	Degrees	**Part turn**
t < 2d	60	1/6
2d ≤ t < 6d	90	1/4
6d ≤ t ≤ 10d	120	1/3

When using the HRC method, bolts should be tightened with a specific shear wrench with two co-axial sockets which react by torque one against the other. The equipment does not need calibration and the preload requirement is controlled by the HRC bolt itself. Tightening usually contains two steps and both of them use the shear wrench. The first step has been achieved, when the shear wrench outer socket stops turning. Again, all the bolts in a joint should be tightened to the first step before starting the second step. The second tightening step has been achieved, when the spline end of the bolt shears off at the breakneck. [14]

When using the direct tension indicator method, the first step of tightening has been achieved when the snug-tight has been reached. The beginning of the initial deformation in the DTI protrusions indicates when the snug-tight has been reached. All the bolts in a joint should be tightened to the first step before starting the second step. The second step is done, when the minimum number of feeler gauge refusals have been achieved. More information about the second step can be found from SFS-EN 1090-2 and Annex J or from SFS-EN 14399-9. This method applies to compressible washers which indicates at least the required minimum preload has been achieved. It does not apply to indicators that rely on torsion or direct measurement of bolt preload by use of hydraulic instruments. [14]

3.8 Tightening tools

The simplest way to tighten the joint is to use **a torque wrench**. It is mainly used with torque control and with monitor it can be used in combined method. It is quite inaccurate if the operator is lacking the skill or motivation. Therefore, many of the wrenches in the market are designed with gauges, clicking sounds or some sort of signal to reduce the operator factor in tightening process. Other problems are the friction, geometry and relaxation. [3] According to SFS-EN ISO 6789-1 [21] the tolerances for a torque wrench are 4-6% depending the wrench type and tool calibration should be done in every 5000 operations or every year depending which comes first.

Figure 17. Torque wrench with digital monitor on the left and slugging wrenches on the right. Torque wrench is from ACDelco [22] and slugging wrenches from Powermaster [23].

Torque multipliers will multiply the torque produced by a torque wrench and usually the ratios are 4:1, 10:1 or up to 100:1. Because of the friction in the gear trains, multiplier decreases the accuracy of a manual torque wrench. [3]

Hydraulic wrenches are good when the tightening happens in small spaces. It also provides very high torques [3]. Usually hydraulic wrenches are powered by electricity but sometimes the compressed air is used. It can provide very good repeatability in results and with digital gauges good accuracy [24]. Hydraulic wrench has an accuracy of around 3% and repeatability of 1% [25].

Figure 18. Hydraulic wrench from Cranergy [26].

Impact wrenches can provide relatively high output torques within a light and inexpensive tool. It is, however, quite noisy and inaccurate [3]. Impact wrench use little hammers to create repeated blows and generating a torque and is usually powered by electricity or air [8]. Joint stiffness affects the torque. Impact wrench creates series of pulses of torque rather than one continuous torque [27].

Nut runners are more accurate and quieter than the impact wrenches [3]. It is usually powered by compressed air. Joint stiffness should not affect the torque [8].

Figure 19. Impact wrench on the left and nut runner on the right. Impact wrench is from Ryobi [28] and nut runner is from Atlas Copco [29].

Micrometers are used to measure the change in bolt length during the stretch control. If there is a possibility to reach both ends of the bolt, C-micrometer can be used. Inaccuracies are caused by irregularities in the end of a bolt and operator skills. Some of the problems are reduced when using the depth micrometer. The depth micrometer measures the distance between the end of the stud and the rod in the center. Using micrometers can be clumsy and slow. [3]

Figure 20. C- micrometer on the left and depth micrometer on the right. C-micrometer is from Starrett [30] and depth micrometer from Mitutoyo [31].

Bolt tensioners control the tension in a bolt itself instead of torque like most of the tightening tools (including the ones which use other parameters like turn). This method gives a lot of advantages since it gives the possibility to control the tension itself but has also its own problems like elastic recovery and thread embedment. Bolt tensioner won’t recognize the difference between a bolt tension and clamping force. A name of hydraulic jack is used to describe the tensioner later in this thesis. [3]

Bolt heater raises the temperature in a bolt faster than in the surroundings, creating different changes in length [3]. Heater is a very cheap method but is slow process and requires relatively amount of skills to do it right. Like bolt tensioner, the heater won't detect the difference between a bolt tension and clamping force [3]. This is not a common method and is used usually with very large bolts [8].

Figure 21. *Bolt tensioner on the left and bolt heater on the right. Tensioner is from Durapac [32] and heater is from Sinus-Jevi [33].*

The basics of **ultrasonic measurement devices** are discussed more in Subchapter 3.6. Device measures the transit time of ultrasonic signal (or echo) from the end of a bolt and use it to estimate the change in length. Bolt's geometry and temperature needs to be inputted to the device. [3]

Figure 22. *Ultrasonic bolt tension monitor from Dakota Ultrasonics [34].*

4. TEST ARRANGEMENTS

In this thesis, the major goal was to study the relationship between torque and tension for HPM and PPM anchor bolts. Test specimens were made from HPM 20, HPM 24, HPM 30, HPM 39, PPM 30, PPM 39 and PPM 52 bolts. Therefore, HPM 16, PPM 36, PPM 42 and PPM 60 were left out. Originally, the idea was to test HPM 16 bolts as well, but because of some misunderstandings these never arrived. 12 test specimens were made for each bolt size. The exception was HPM 30 which consisted 15 test specimens. The tests were held at Peikko's factory in Lahti during December 2017 - February 2018.

Test set-up included anchor bolts, column shoe base plates, steel plates, torque wrenches, torque multiplier and measurement devices. Torque wrenches and multiplier was used to increase the bolt tension. For each step, torque, tension and angle readings were marked down. Some of the bolts were lubricated or left outside to expose the weather in order to change the bolt condition. During the tests, some minor problems occurred and the set-up was improved several times.

4.1 Test set-up

Test set-up is presented in Figure 23. The individual set-ups for each bolt can be seen in Appendix A. Bolt length is designed to fit between column shoe base plates, steel plates and hydraulic jack. In this research, HPM thread length was designed according to the Peikko's technical manual but the test specimens had threads starting from both ends, unlike the commercial versions. Besides the bolt length and HPM threads, there was no difference between the test specimens and the commercial bolts.

Washers are designed according to SFS-EN 10052-2 and nuts according to SFS-EN ISO 898-2 and SFS-EN ISO 4032. Column shoe selection is based on which bolt type and size is used. HPKM column shoes are used with HPM bolts and PEC column shoes are used with PPM bolts. During the tests, there was no need for the whole column shoe so only the base plate was used.

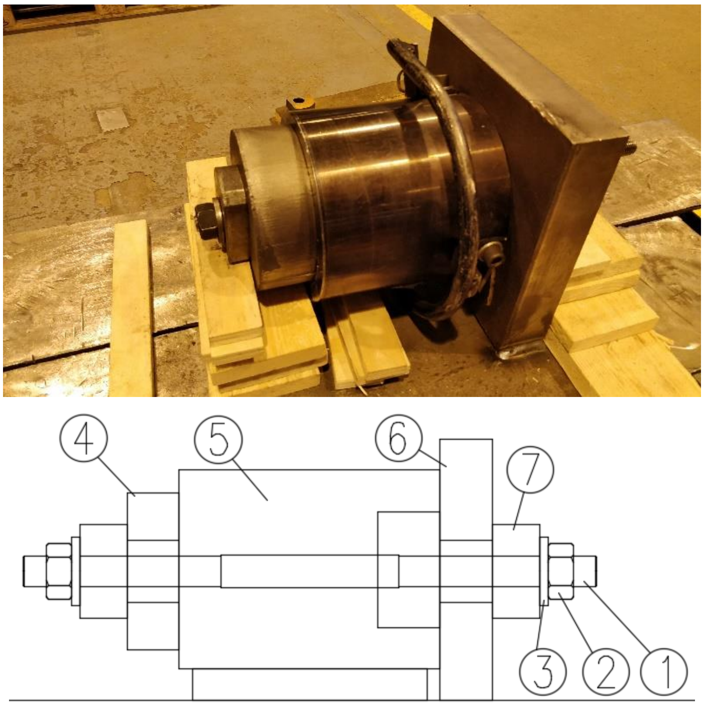

Figure 23. *Test set-up. Bolt (1), nut (2), washer (3), round steel plate (4), hydraulic jack (5), circular steel plate (6) and column shoe plate (7). Outside of the picture is a hydraulic pump which was used to control the hydraulic jack.*

There are two nuts, two washers and two column shoe base plates in each set-up. Nuts and washers are used only once. Column shoe base plates were used multiple times, but it always matches with the bolt type and size. Round steel plate has dimensions 150x50 mm^3 and circular steel plate has dimensions 250x250x50 mm^3. Both are made from material class S355 steel. Steel plates must be used because otherwise the smallest column shoe plates would not fit with the hydraulic jack. The circular steel plate is welded into two supportive I-profile beams. This provides support for the set-up. Sockets and adapters were used between the nut and torque wrench or multiplier. Hydraulic jack and pump are used as measurement devices.

In Figure 23, the set-up is placed on horizontally. However, it was found later that this may cause a minor error to the results. For this reason, majority of the tests were made in vertical direction, but the set-up still included all the parts presented in Figure 23. More on this in Subchapter 4.6.

4.1.1 Anchor Bolts

HPM Rebar Anchor Bolts and **PPM High-Strength Anchor Bolts** are manufactured by Peikko. Both are used in residential and office buildings, warehouses, halls, bridges, dams and power plants, for example. HPM and PPM bolts come with nuts and washers. Nuts are designed according to SFS-EN ISO 898-2 and SFS-EN ISO 4032 while washers are designed according to SFS-EN 10052-2. [35, 36]

HPM is a regular type of anchor bolt while PPM is designed for higher loads and more demanding connections. HPM is made from B500B ribbed reinforcement steel bars and PPM is made from B500B bars and threaded bars with property class 8.8. HPM bolts are found in sizes M16, M20, M24, M30 and M39 and PPM are found in sizes M30, M36, M39, M45, M52 and M60. Bolts can be designed for axial forces, bending moments, shear forces and fire exposure. [35, 36] The examples of test specimens used in the tests can be found from the Figure 24.

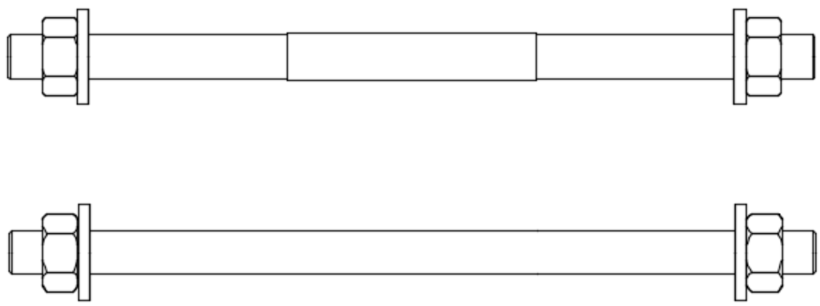

Figure 24. *Test specimens that were made from HPM (top) and PPM bolts. HPM bolts have threads in both ends and PPM bolts have threads through the whole length. Two nuts and washers were manufactured for each bolt.*

There are two types of HPM and PPM bolts. Type L is shorter of the two and has a headed stud which is used to transfer the loads. These are typically used in shallow structures because of their relatively short anchorage length. Type P is long and straight anchor bolt. It transfers the loads by using the bond between the bolt and concrete. Furthermore, the loads are transferred from concrete to the main reinforcement. Type P bolts are usually used in structures with sufficient depth or in shallow structures if the bolts have been bended. The difference can be seen from the Figure 25 and Figure 26. [35, 36]

Figure 25. *HPM L and HPM P Anchor Bolts [35].*

Figure 26. *PPM L and PPM P High-Strength Anchor Bolts [36].*

HPM and PPM anchor bolts are cast into concrete and attached to the base plate of the structure by nuts and washers. This transfers the loads from the structure to the base concrete. Minimum and maximum tightening torques are given in Table 11 for HPM bolts and in Table 12 for PPM bolts. Using a 1,5 kg sledgehammer and by 10-15 impacts with a slogging ring wrench (DIN 7444) or open-ended slogging wrench (DIN 133) usually provides proper torque. The needed space for tightening the nuts needs to be considered in case of nearby structures. The tightening is based on torque control. [35, 36]

Table 11. *Minimum and maximum recommended tightening torque values according to HPM Technical Manual [35].*

Anchor Bolt	T_{min} [Nm]	T_{max} [Nm]
HPM 16	120	170
HPM 20	150	330
HPM 24	200	570
HPM 30	250	1150
HPM 39	350	2640

Table 12. *Minimum and maximum recommended tightening torque values according to PPM Technical Manual [36].*

Anchor Bolt	T_{min} [Nm]	T_{max} [Nm]
PPM 30	250	700
PPM 36	300	1200
PPM 39	350	1400
PPM 45	400	2000
PPM 52	450	3300
PPM 60	500	3800

4.1.2 Column shoes

HPKM column shoes are used with HPM anchor bolts and **PEC column shoes** are used with PPM anchor bolts. Both are manufactured by Peikko. Column shoes and anchor bolts attach the precast concrete column and foundation together. Column shoes are casted into the column and anchor bolts into the foundation. In addition, these can be used to attach two precast concrete columns together. Usually four or more column shoes are needed to achieve moment resisting connection. The same stiffness can be designed as with the continuously reinforced cast-in-situ connections. [37, 38]

Figure 27. *Only the column shoe base plates were used during the tests.*

During the tests, only the column shoe base plates were used and not the other parts of column shoe. An example of base plate can be seen in Figure 27. Plate thicknesses are presented in Table 13. The base plate is made from S355 steel. [37, 38]

Table 13. *Thickness of HPKM and PEC column shoe base plates [37, 38].*

Column Shoe	Thickness [mm]
HPKM 16	15
HPKM 20	20
HPKM 24	30
HPKM 30	45
HPKM 39	50

Column Shoe	Thickness [mm]
PEC 30	45
PEC 36	50
PEC 39	60
PEC 45	60
PEC 52	70

4.1.3 Torque wrenches and multiplier

Norbar SL1 is a manual torque wrench and has a range of 8-54 Nm. It is accurate to ±4%. [39]

Norbar SL2 is a manual torque wrench and has a range of 30-150 Nm. It is accurate to ±4%. [40]

Britool HVT7200 is a manual torque wrench and has a range of 200-810 Nm. It is accurate to ±4% and complies with ISO 6789. [41]

CWalter Mx80 is a torque multiplier. Torque transmission is ±5% and it has maximum output torque of 8000 Nm. The ratio is approximately 1:49. [42]

Figure 28. *Torque wrenches and multiplier used in tests. From the top to bottom: Norbar SL1, Norbar SL2, Britool HVT7200 and CWalter Mx80.*

4.1.4 Hydraulic jack and pump

Tentec WTP3 is an electric driven high pressure hydraulic pump unit that can generate pressures of up to 1600 bar. Pump can be remote controlled with a controller hand grip. Operating environment temperature is between 0°C and 40°C. [43]

The hydraulic jack was manufactured by **Tentec.** In the end, it was unclear what model this specific tensioner was, but it has an effective area of 12048,01 mm^2 and that was all we needed to know. The hydraulic jack was used to measure the bolt load.

Figure 29. *Tentec WTP hydraulic pump and tensioner.*

4.1.5 Lubricants

AT-2000 Spray Vaseline is suitable for general lubrication of the bearings and are used for cables, joints, hinges, rolling and sliding bearings, chains and gears for example. Operating temperature is between -30°C and 120°C. [44]

Molykote G-Rapid Plus is solid lubricant paste with low friction coefficient. It is used for threaded spindles, tooth gearing, moving screws and fitting bolts for example [45]. Operating temperature is between -35°C and 450°C. The friction coefficient for thread is 0,10 and for head it is 0,06 [45]. Torque coefficient should be somewhere around 0,15 [46].

Figure 30. *Molykote G-Rapid Plus and AT-2000 Spray Vaseline.*

4.2 Test procedure

At first, the test set-up was assembled around the hydraulic jack. Column shoe base plate was selected according to the tested bolt type and size. Markings were made to the socket and the end nut to measure the angle and check whether the end nut had been rotated or not. Here, the end nut means “the other nut” which is not tightened. Then the hydraulic jack was activated and the current pressure in the system can be read from the monitor.

Figure 31. *Current pressure in the system can be read from the hydraulic pump's hand grip monitor.*

After the set-up had been assembled, the tightening process could be started. Tightening starts with Norbal SL2 torque wrench between torques 50-150 Nm and after that with Britool HVT7200 between torques 200-800 Nm. Because HPM 20 started to yield before reaching 800 Nm, test points were measured in every 50 Nm torque and after the yield point 20-50 Nm steps were used. For all the other bolts, the test points were measured in every 100 Nm.

Table 14. *Range of used torque wrenches.*

Torque wrench	Min	Max
Norbar SL1	8	54
Norbar SL2	30	150
Britool HVT7200	200	810

Besides HPM 20, other test specimens needed the torque multiplier to reach the yield point. It does not work without any torque wrenches so Norbar SL1 and Norbar SL2 were used with the multiplier. Here the measurement was made by using 5-10 Nm input torque steps which is equivalent to around 200-500 Nm output torque steps.

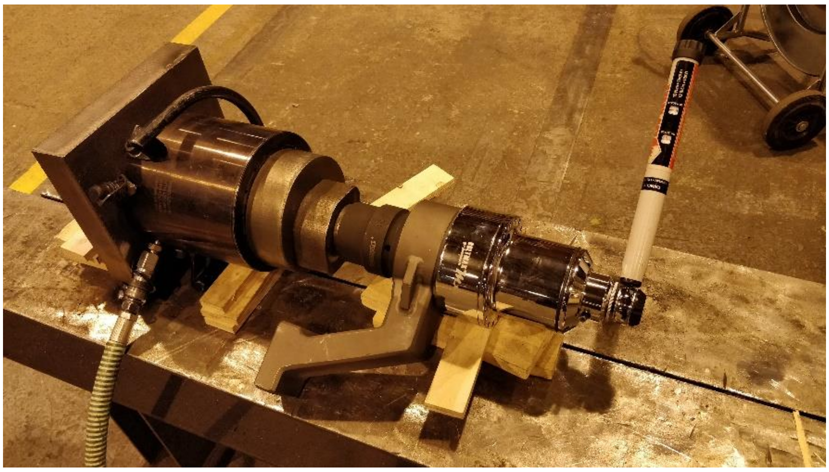

Figure 32. CWalter Mx80 torque multiplier and SL1 torque wrench.

This test method was chosen based on circumstances and equipment available at the time. The method allows to measure torque, bolt tension and nut rotation simultaneously. Tests were held at Peikko's facilities in Lahti between December 2017 and February 2018. During the first tests 11.-15.12.2017 the test set-up was mainly calibrated and improved to achieve reliable data later. Other test periods were held 3.1.-12.1.2018, 22.1.-26.1.2018 and 19.2.-23.2.2018. It took 23 days in total to complete the tests.

4.3 Changing the bolt condition

In order to test the torque – tension relation with different torque coefficients, the bolt condition needed some variations. The variation was caused by lubrication and corrosion. For this reason, part of the test specimens were left outside for exposing the snow and humidity. This is meant to simulate the situation where bolts are left in construction site for some time and affected by corrosion.

Figure 33. Bolts exposing to the snow and humidity. These bolts have the nuts and washers still attached to them during the corrosion process.

Originally, the corrosion test group was left outside to expose the weather. Hopes were that no further aid is needed to achieve enough rust. However, when testing HPM 20 and HPM 30 bolts, it was noted that not enough rust was achieved in this given time period. It was then decided that considering the time limit this master's thesis was done, there was just not enough time for bolts to collect deep rust. For this reason, salted water was sprayed over the bolts that were not tested yet. This was done 26.1.2018.

Figure 34. *Deep rust achieved to the bolts after giving the salt bath.*

Besides the salt bath, there was another factor that differentiates how the bolt was exposed to the corrosion. Some bolts had nuts attached to them during the process while others had the nuts removed. Usually when bolts have been stored outdoors, most likely the nuts and washers are still attached to them. Example of this is in Figure 33. On the other hand, threads would be more exposed to the humidity if the nuts were removed. For these reasons, both methods were tested.

Table 15. *Time and method used for rusting bolts.*

Bolt	Time period	Weeks	Method	Sal-ted
HPM 20	14.12.-3.1.	3	Nuts on	No
HPM 30	14.12.-25.1.	6	Nuts on	No
HPM 39	3.1.-19.2.	6,5	Without nuts	Yes
PPM 30	4.1.-19.2.	6,5	Nuts on	Yes
PPM 39	8.1.-20.2.	6	Without nuts	Yes
PPM 52	12.1.-19.2.	5,5	Without nuts	Yes

For Table 15 has been gathered all the information about the corrosion process. This includes the time period, nut position and the use of salt. There were three test specimens of each bolt size and these were all identical considering the time and method.

Figure 35. *Lubricated HPM 39 bolt, nut and washer with G-Rapid Plus.*

The corrosion is supposed to increase the torque coefficient. Additionally, the coefficient was decreased by lubrication. Two lubricants were used: AT-2000 Spray Vaseline and Molykote G-Rapid Plus. AT-2000 is considered as a basic lubricant and G-Rapid Plus is an example of low friction lubricant. AT-2000 was an aerosol and G-Rapid Plus was paste. These were added to the one end of the bolt, one nut and one washer in high portions to get the effect of lower friction to be clear.

4.4 Test name

Each test specimen was named individually. This helps to identify a specific test later if needed. An example can be seen in Figure 36. Bolt type tells whether the test specimen represents HPM or PPM anchor bolt. Five different sizes of threads were tested. For HPM, sizes M20, M24, M30 and M39 were tested, while for PPM, sizes M30, M39 and M52 were tested.

Bolt condition has been marked after the bolt type and size. Here the following abbreviations have been used: AR = as received, L = lubricated and R = rusted. The test number is used to specify small changes in test methods or bolt condition. For example, if both G-Rapid Plus and AT-2000 lubricants were used, there would be two different test numbers. On the other hand, if there were problems during the tests, any improvements to the set-up would change the test number. So, in other words this means that if there is a same test number for two test specimens which has the same bolt type and size, these can be considered as identical tests.

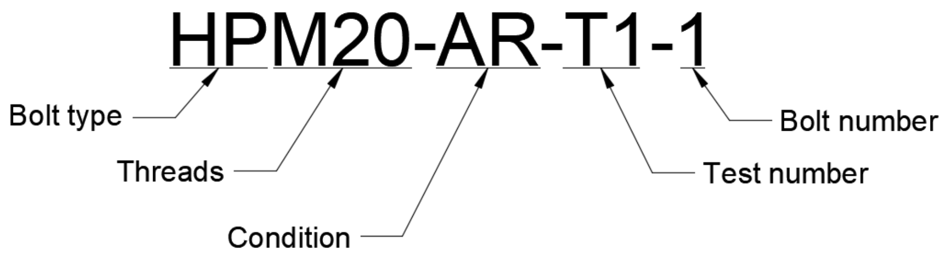

Figure 36. *An example of naming the test.*

Bolt number identifies individual test specimen. This is not dependent on bolt condition or test number. The bolt number always starts from one for each bolt type and size. Therefore, bolt numbers 1-12 are used for each HPM and PPM bolts. The only exception is HPM 30 bolts that have numbers 1-15. Bolt numbers are useful when the test specimens are rechecked later because the whole name would be impractical to mark into the bolt. The bolt number was marked into the end of each test specimen where the tightening was done.

4.5 Angle tests

Besides measuring the torque and tension, the angle was also measured during the tests. Because the original test set-up includes the hydraulic jack, a minor error may occur in the results. The reason is the effective length which is longer than it normally would when the jack is used. That is why the additional control tests were made without the hydraulic jack. The example can be seen in Figure 37. However, when the analysis was completed, it was noticed that the results were not very accurate and the data was basically unusable. The reason was too inaccurate test methods. Also, the control tests had too small sampling to make any reliable conclusions. Therefore, the angle tests had to be left out from the test results.

Figure 37. *Torque – angle control tests were made without the hydraulic jack.*

4.6 Problems and improvements

There was not much of experience how well the test procedure will work. Some minor tests were done successfully earlier but it was not on this scale by using many different bolt sizes and high torques. Therefore, the set-up was improved gradually when the problems occurred.

Originally, the test set-up was placed on horizontal direction as it was presented in Figure 23. However, this led to some problems. When using the torque multiplier, it started to lift upwards the far end of the other beam making the procedure more difficult to handle and probably causing minor inaccuracies to the results. The beam was twisting as it can be seen in Figure 38. This problem would have been solved by welding steel plate to the far end to stop the twisting movement. But it was not the only problem caused by the horizontal test set-up.

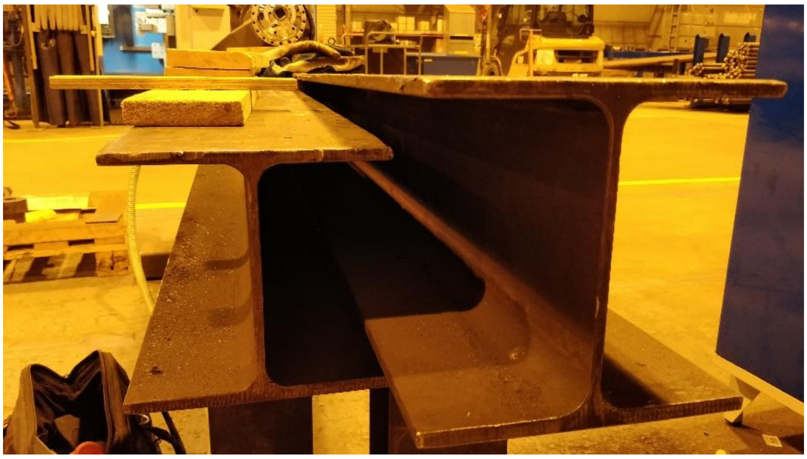

Figure 38. *"The twist" when using torque multiplier in horizontal direction.*

The other concern was whether the equipment placed on horizontal or vertical direction will affect the results or not. There is always the risk that the parts could be leaning on the bolt when the equipment is placed on horizontal direction. This was tested with HPM 20 bolt and, as seen in Figure 39, there is a small difference in the results. It seems that in vertical direction the torque coefficient was slightly smaller than in horizontal direction.

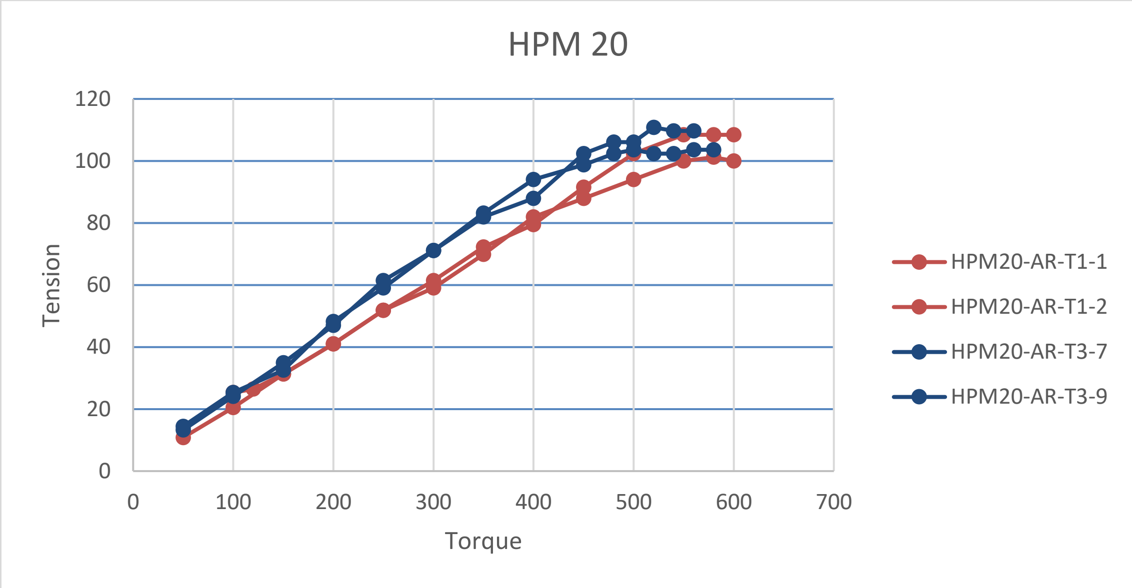

Figure 39. *The difference in torque – tension curve when the test set-up is in horizontal (red) and vertical (blue) positions.*

At this point, it was decided that all the upcoming tests would be done in vertical direction. This solved both of the problems since the twisting did not happen anymore and the uncertainties with the horizontal set-up had been eliminated. The original horizontal and new vertical set-ups can be seen in Figure 40.

Figure 40. *The horizontal and vertical set-ups.*

The new vertical set-up works with HPM 20 bolts since no torque multiplier is needed. However, all the other test specimens need the multiplier and a support for it. The support was done by welding two steel plates into the supportive beams. But it turned out that the support created another problem. It supports the torque multiplier but also carries part of the load. This caused a minor error when the multiplier is used. This error is determined in Subchapter 5.7.2 and it has been taken into account when the total margin for error has been calculated.

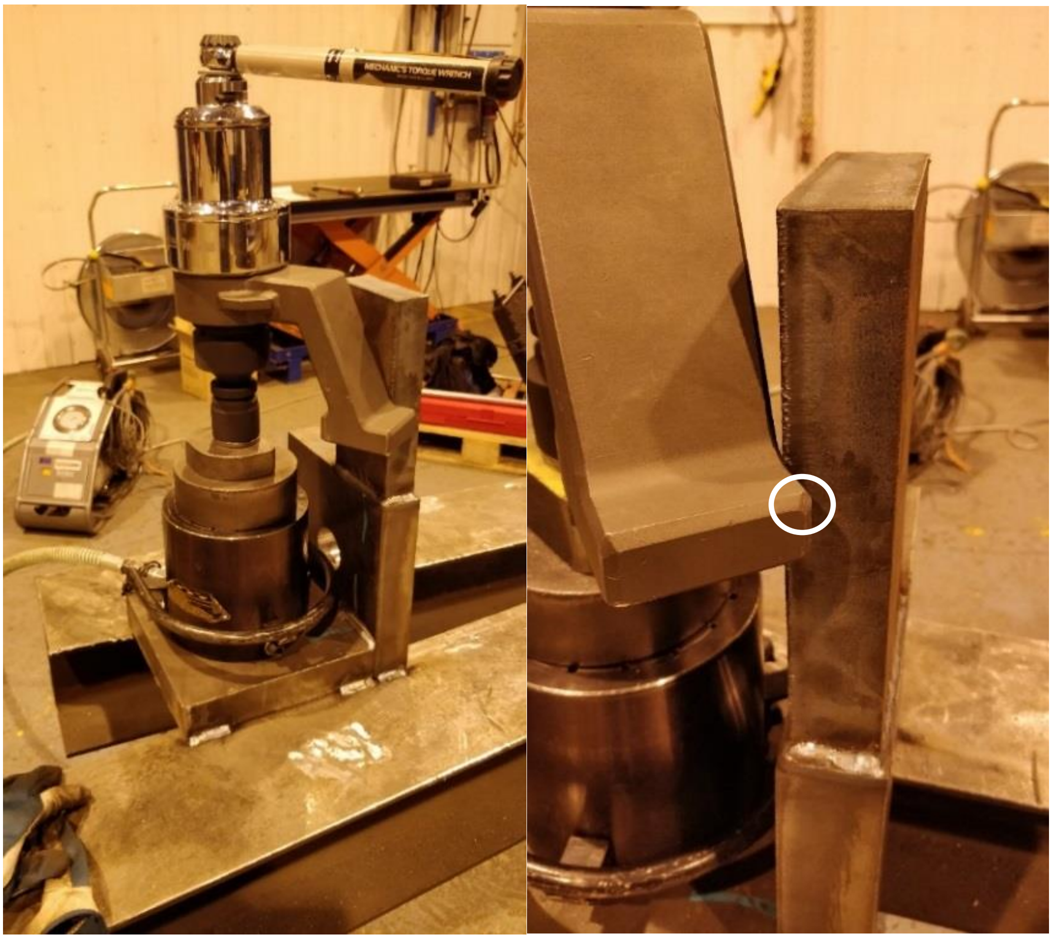

Figure 41. *Support for torque wrench was assembled. This is making a minor error to the results because small portion of the load is carried by the support.*

During some tests, the round steel plate and column shoe base plate started to rotate under the nut. This was caused by the smaller friction in surface of hydraulic jack comparing to washer. With larger bolts, larger washers are used. This increases the friction area and is more likely to make the parts to rotate when the nut is rotated. Therefore, this problem is more relevant when larger bolts are used and that was the case during the tests.

It was a difficult task to stop the rotation. There should be a support against the rotational movement but at the same time, it should not carry any loads and cause errors to the results. Deep throat clamp and wooden pieces were used to stop the rotation. The improved set-up can be seen in Figure 43. It was noted, that this won’t change the bolt load readings, so this is the final set-up and it was used for majority of the tests.

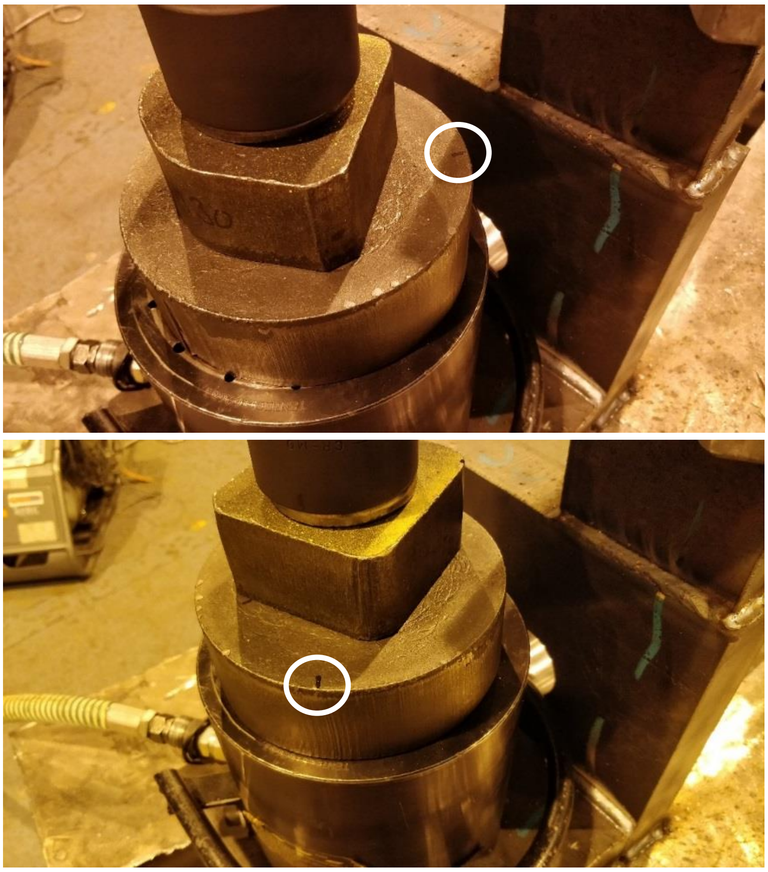

Figure 42. *The rotation in round steel plate and column shoe base plate occurred with larger bolts.*

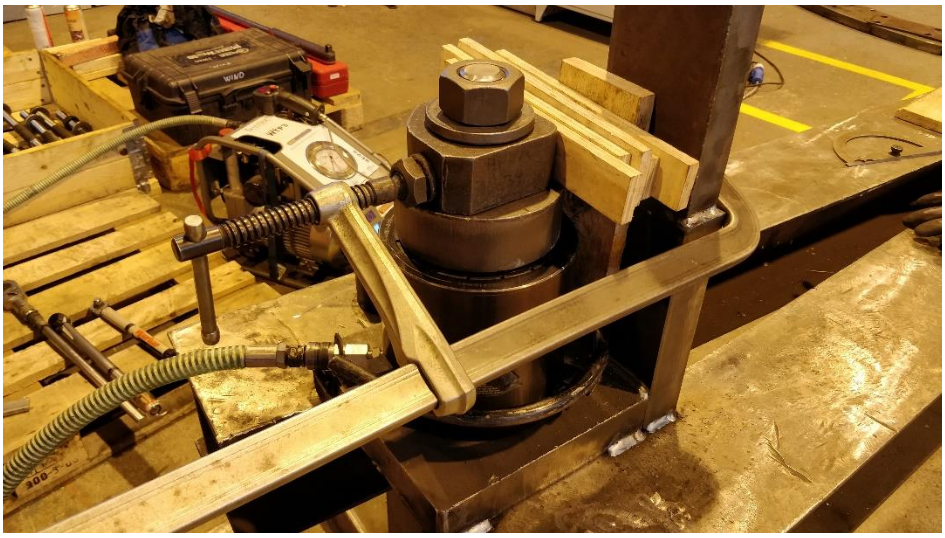

Figure 43. *Preventing the column shoe from rotating solved the problems occurred with rotation.*

Not all the problems were caused by the test set-up. The torque multiplier had its problems too. The ratio that changes the input torque to output torque was not constant or linear as it can be seen from Table 16. It was decided that we can assume the ratio to be linear between the values given by the manufacturer. Since the range of torque wrenches ends at 800 Nm torque and the given multiplier values starts from 2841 Nm, many of the test points were left between these two. Therefore, one additional ratio was experimentally determined. The result was that for 18 Nm input torque the average output torque was 745 Nm and the ratio is 41,1. Then the distance between 745 Nm and 2841 Nm was assumed to be linear. Calculations for multiplier ratios are presented in Subchapter 5.1.

Table 16. *Ratios to CWalter Mx80 torque multiplier were given by the manufacturer.*

CWalter Mx80		
In [Nm]	**Out [Nm]**	**Ratio**
60	2841	47,4
80	3839	48,0
100	5029	50,3
120	5986	49,9
140	6896	49,3
160	7710	48,2

The other problem was caused by the torque multiplier assembly. The assembly took a few minutes, so there was just enough time for short-term relaxation. This causes the bolt tension to drop and a discontinuity point to 800 Nm torque as it can be seen in Figure 44. The drop in bolt tension was calculated in Subchapter 5.7.1 and it was taken into account when the total margin for error was determined.

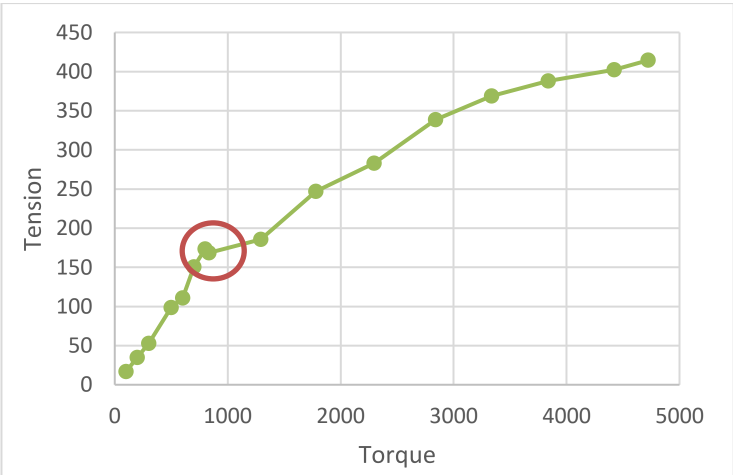

Figure 44. *Short-term relaxation caused a discontinuity point to 800 Nm torque.*

During some tests, the end nut rotated, and this is not supposed to happen. This could be detected by marking the column shoe base plate and nut with a black line. The test specimens where the nut rotation was significant, were left out from the analysis because the caused inaccuracies. Later, double end nut was used, and this stopped the rotation in majority of cases. The nut rotation problem occurred usually with the smaller bolts.

Figure 45. *The nut rotation problem.*

It was noted after a while that the bolt is rotating in each test. This makes the bolt to be tightened from the both ends. When as received and rusted bolts were tested, this should not cause any errors since the torque coefficient is the same in the both ends. But the lubrication was only added to one end and therefore different torque coefficients exists. In theory, this should not cause any errors because the nut is expected to rotate where the friction is the lowest. There might be small inaccuracies with the lubricated bolts if the rotation still occurred. A couple of the bolts also bended, which can make a minor error to the results. This happened because the bolt and washer were not placed centrally and the washer caused the bolt to bend.

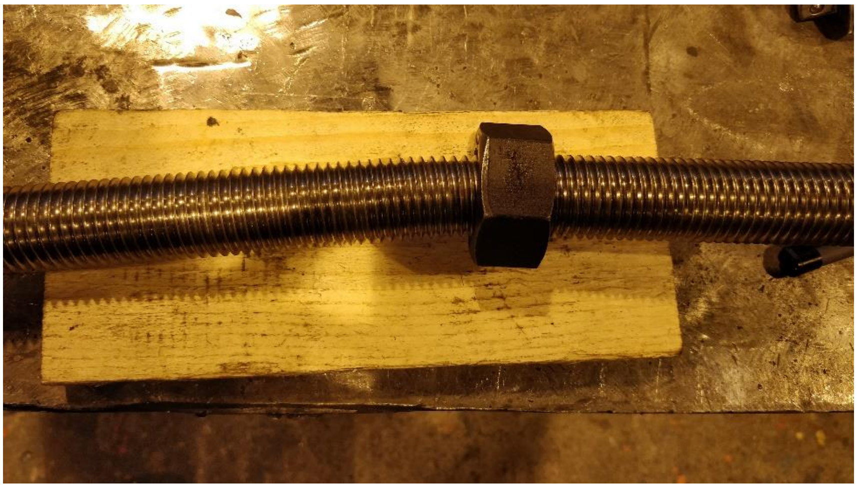

Figure 46. *Bended PPM 30 bolt.*

4.7 Bolt tests in EN 14399-2

According to SFS-EN 14399-2 [47], the following information should be given when tested at bolt assembly manufacturer:

- date of testing
- details of the test equipment
- the single assembly lot number or the extended assembly lot number
- number of bolting assemblies tested
- designation of the bolting assemblies in accordance with EN 14399-3 to EN 14399-10 as relevant
- test clamp length
- speed of tightening
- remarks concerning the execution tests
- tests results according to EN 14399-2
- conclusions

According to SFS-EN 14399-2 [47], additional information should be given when tests are not carried by the bolt assembly manufacturer:

- identification of the laboratory
- identification of the organization ordering the test
- date of reception of the assemblies
- details of the test set-up including rigidity and number of shims
- tightening conditions

According to SFS-EN 14399-2 [47], the following things should be considered during test procedures:

- the test shall be carried out at an ambient temperature range from 10°C to 35°C inclusive
- the tightening shall be carried out by rotation of the nut in a continuous manner and measurements shall be recorded throughout the test
- the speed of rotation of the test shall be from 1 min^{-1} to 10 min^{-1} inclusive
- neither the bolt nor the washer under the nut shall be free to rotate during the test

The tests made for this thesis fills all the requirements except two. The tightening speed was not measured so it is unknown. Additionally, since the tightening was done by hands with the torque wrench and multiplier, the tightening speed was not constant. The other exception was that during our tests the washer and bolt were allowed to rotate.

Table 17. *Parameters to measure for obtaining a certain tightening characteristics according to SFS-EN ISO 16047 [48].*

Tightening characteristics	**Parameters to be measured**				
	Clamp force F	**Tightening torque T**	**Thread torque T_{th}**	**Bearing surface friction torque T_b**	**Rotation angle θ**
Torque coefficient, K	x	x			
Coefficient of total friction, μ_{tot}	x	x			
Coefficient of friction between threads, μ_{th}	x		x		
Coefficient of friction between bearing surfaces, μ_b	x			x	
Yield clamp force, F_y	x				x
Yield tightening torque, T_y	x	x			x
Ultimate clamp force, F_u	x				
Ultimate tightening torque, T_u	x	x			

5. TEST RESULTS

Test points were determined according to the instructions presented in last chapter. But it should be noted that some of the test points were left out from the analysis. Only the test specimens where no major problems occurred were used as a result data to achieve as reliable results as possible.

The main goal was to study the torque coefficient. It was determined for each bolt size and type, for different conditions and for combinations of these. Three conditions were selected: as received, lubricated and rusted. Two different lubricants were used, one is considered as a basic lubricant and other is considered as a low friction lubricant. Two methods for achieving rust were used. One with salt bath and one without.

In addition, two other major subjects were studied based on torque – tension relation. The earlier determined torque coefficients were used and needed torques calculated for reaching the bolt's yield strength and 70% of the tensile strength that is suggested in Eurocode. The other thing was to draw the torque – tension curves and to study these. Then curves were compared to each other and to the yield torques.

Some minor subjects were standard deviation and tensioning the bolt past its capacity. Standard deviation and scatter for torque coefficient were determined to study how the bolt size, type and condition affects the tightening accuracy. In addition, one PPM 30 bolt was tensioned past its capacity by using the hydraulic jack to see how the bolt breaks. Torque – angle relation was also tested but it proved to be too inaccurate and therefore was left out from this chapter.

The margin of error has been studied in the last subchapter. Estimations for short-term relaxation and torque multiplier support caused errors were determined experimentally. The scatter for torque wrench and multiplier were determined from test points and then these were compared to each other to see how much inaccuracies the multiplier produces. At this point, the experimentally determined errors were combined with errors given by equipment manufacturers and the total inaccuracy was estimated.

5.1 Bolt load and torque with multiplier

The bolt load can be calculated by using the hydraulic jack and pump. However, the bolt load couldn't be read directly from the pump monitor, so the equation below was used

$$F = p_h A_h \tag{25}$$

where p_h is the pressure that can be read from the hydraulic pump monitor and A_h is the hydraulic jack cylinder effective area. The load was calculated from each test point and these points were selected evenly with a few exceptions. During the first tests, there were smaller steps between the test points because of calibration. Other exception was when the yielding started. Then occasionally the step distances were reduced to determine the maximum bolt load more accurately.

When the torque multiplier was used, obviously the ratio between input and output torque needs to be known. Like it was mentioned in Subchapter 4.6, the ratio is not constant or linear. In Table 18, the input and output torque are given by the manufacturer. The ratio has been calculated from these given torques.

Table 18. *Given ratios to CWalter Mx80 torque multiplier.*

CWalter Mx80		
In [Nm]	**Out [Nm]**	**Ratio**
60	2841	47,4
80	3839	48,0
100	5029	50,3
120	5986	49,9
140	6896	49,3
160	7710	48,2

This causes two problems. First of all, the torque wrench range ends at 800 Nm and the torque multiplier range starts from 2841 Nm. This leaves many test points between these two. The problem was solved experimentally. At first, the bolt was loaded with torque multiplier at certain level and this was marked as an input torque. Then the torque wrench was used and the torque that matches with the same bolt load was searched. This was marked as an output torque. From the results, mean values were calculated and these are presented in Table 19.

Table 19. *Experimentally determined ratio for 18 Nm input torque.*

Experimental values		
In [Nm]	**Out [Nm]**	**Ratio**
18	745	41,4

The second problem is that the ratio between given torques, for example 70 Nm input torque, is unknown. Here it was assumed that the change of ratio is linear between the known torques. While this might not be exactly true, the error it gives is minimal compared to other uncertainties. The relations between ratios and input torques can be presented as

$$\frac{R_{high} - R_{low}}{T_{high} - T_{low}} = \frac{R - R_{low}}{T - T_{low}} \tag{26}$$

where R_{high} is the high-end ratio, R_{low} is the low-end ratio, T_{high} is the high-end input torque, T_{low} is the low-end input torque, R is the ratio for input torque and T is the input torque. The equation can be restructured

$$(R - R_{low})(T_{high} - T_{low}) = (R_{high} - R_{low})(T - T_{low}) \tag{27}$$

In order to calculate ratio R some changes need to be done

$$R - R_{low} = (R_{high} - R_{low})\frac{T - T_{low}}{T_{high} - T_{low}} \tag{28}$$

Finally, the ratio between given values can be calculated from the equation

$$R = R_{low} + (R_{high} - R_{low})\frac{T - T_{low}}{T_{high} - T_{low}} \tag{29}$$

Therefore, torque when the torque multiplier is used

$$T_{out} = RT_{in} \tag{30}$$

where T_{out} is the output torque from multiplier, R is the ratio that transfers the input torque to output torque (from Table 18, Table 19 or Equation 29) and T_{in} is the input torque applied by torque wrench.

5.2 Torque coefficients

The torque coefficient can be calculated from the equation that describes the torque – tension relation

$$T = Fdk \tag{31}$$

where T is the input torque, F is the bolt load, d is the bolt diameter and k is the torque coefficient. Therefore, the torque coefficient is

$$k = \frac{T}{Fd} \tag{32}$$

In this study, only the test specimens without any major problems considered as a part of the result data. On the other hand, test points that required torque multiplier or where it is on plastic region have been ruled out because of the inaccuracies. Additionally, it was noted that the first test point was more inaccurate than the others. For these reasons the data contains only the test points where is successful set up, torque 800 Nm or less, elastic region and starts from the second test point.

For each test point, the torque coefficient was calculated by using the Equation 32. Based on these calculations the mean and scatter were determined. Here the definition of minimum and maximum coefficient means that there was at least one test point with this value, so these might be quite rare cases.

For Table 20 and have been calculated torque coefficients for each bolt size in as received condition. This data is based on 2-4 bolts and 14-28 test points in each case. Because of the small sampling, there may occur some inaccuracies in the results. The results for PPM bolts have been left out from this public version.

Table 20. *Torque coefficients for tested HPM bolts as received condition.*

Bolt	Test specimen	Min	Mean	Max
HPM 20	2	0,20	0,21	0,23
HPM 24	3	0,21	0,22	0,25
HPM 30	4	0,18	0,20	0,21
HPM 39	2	0,18	0,20	0,22

As the table shows, the bolt diameter doesn't change the torque coefficient very much or there isn't any distinctive pattern at least. Small variation could be explained by small sampling. However, the fact that bolt diameter doesn't affect the torque coefficient, was anticipated. This is because in the equation that describes the relationship between torque and tension, the bolt diameter is presented as a separate variable from the torque coefficient. Therefore, the bolt diameter should not affect to the torque coefficient.

Table 21 consists all the test points that were found reliable and the data is sorted by the bolt condition. Here 3-11 bolts and 21-74 test points have been used. This includes all the bolt sizes. Some of the rusty bolts were given salt bath in order to speed up the corrosion process as it was mentioned in Subchapter 4.3. Therefore, there are two different torque coefficients for rusted HPM bolts. For rusted and salted bolts, only three bolts and one bolt size were used. This may cause minor inaccuracies. All the other results consist five or more bolts, so these can be considered as reliable results.

Table 21. *Torque coefficients for HPM bolts in different conditions.*

Condition	Test specimen	Min	Mean	Max
As received	11	0,18	0,21	0,25
AT-2000	8	0,15	0,18	0,23
G-Rapid Plus	7	0,12	0,16	0,18
Rusted	6	0,18	0,21	0,24
Rusted (salted)	3	0,33	0,40	0,61

As it was expected, the lubrication will decrease the torque coefficient and the corrosion will increase it. AT-2000 is considered as a basic lubricant and G-Rapid Plus as a low friction lubricant. The results seem to support this claim. AT-2000 drops the torque coefficient to 0,18 and G-Rapid Plus drops it to 0,16. At first, the difference to the as received condition doesn't seem to be big but even the 0,02-difference in torque coefficient will have a significant impact to the torque – tension relation as we can see later.

During the first tests to the rusted bolts, there was no big difference in torque coefficient as we can see in Table 21. Simply, the three- or six-week rusting period was not enough to achieve deep rust. The bolts had a rusty surface, but it didn't affect the torque coefficient. At this point, it was decided to give to the rest of the bolts salt bath and wait for a few weeks. This had a significant effect, not only the torque coefficient but also the scatter increased a lot. This makes the behaviour of torque – tension relation more unpredictable and very difficult to handle. However, the salt may also increase the friction a little, so it is unsure if all the added friction comes from the corrosion.

E. Hemmati Vand, R. H. Oskouei, and T. N. Chakherlou tested the torque coefficients for lubricated and non-lubricated bolts in an article made in 2008 [49]. For non-lubricated bolts the result was 0,205 and for greased bolts it was 0,165. Bickford [3] has listed some typical torque coefficients. The coefficient for as received bolt is 0,2 (scatter 0,158-0,267), machine oil is 0,21 (scatter 0,10-0,225) and for moly paste or grease is 0,13 (scatter 0,10-0,18). In Molykote datasheet [46] it was calculated that for G-Rapid Plus the torque coefficient should be somewhere around 0,15. Comparing these examples to the results we got, our torque coefficients seem to be realistic. It should be noted that these torque coefficients are valid only when the elastic part of torque – tension relation are used.

Table 22. The effect of the time to the bolt corrosion.

Bolt	Time	Test specimen	Min	Mean	Max
HPM 20	3 weeks	3	0,18	0,21	0,23
HPM 30	6 weeks	3	0,20	0,22	0,24

There were minor differences how the bolts were exposed to the corrosion. In Table 22 the difference was time. There were 3 bolts and 21 test points used for each test. HPM 20 bolts were left outside for three weeks and HPM 30 bolts for six weeks. Here no salt bath was used, so the difference to the as received bolts is only minimal. The results suggest that three weeks doesn't have an impact to the torque coefficient and six weeks have a slight impact. It would also be logical that over time the torque coefficient will increase. However, since the difference is so small and six weeks is too short time period, these results might be also caused by measurement errors. Therefore, the conclusion that six weeks have a small effect is questionable.

There is no one answer how much is the torque coefficient for rusted bolt since it is dependent on time and other circumstances. The only thing that can be considered as certain is that rust will increase the coefficient value and scatter. Therefore, the estimation on torque – tension relation becomes more inaccurate. This situation should be avoided at least when the time period is several months.

5.3 Calculating the yield and 70% of the tensile strength

The force that is needed to reach the yield strength in a bolt can be calculated from the equation

$$F = f_{yb} A_s \tag{33}$$

where f_{yb} is the yield strength that can be read from Table 23 and A_s is the bolt's nominal stress area that can be read from Table 24.

The force that is needed to reach 70% of the ultimate tensile strength in a bolt, like Eurocode suggests, can be calculated from the equation

$$F = 0{,}7 f_{ub} A_s \tag{34}$$

where f_{ub} is the ultimate tensile strength that can be read from Table 23.

Table 23. *Nominal values of the yield strength f_{yb} and the ultimate tensile strength f_{ub} for HPM bolts.*

Strength	HPM
f_{yb} (N/mm^2)	500
f_{ub} (N/mm^2)	550

Table 24. *Nominal stress areas A_s for HPM bolts used in tests.*

Thread	Nominal stress area
M20	245
M24	352
M30	561
M39	976

On the other hand, the relationship between torque and bolt load can be described with a torque coefficient and the bolt diameter

$$T = Fdk \tag{35}$$

where T is the torque, F is the bolt load, d is the bolt diameter and k is the torque coefficient. From the torque - tension relation the bolt load can be calculated by using the equation

$$F = \frac{T}{dk} \tag{36}$$

The equation above can be combined with the yield strength force and 70% of the tensile strength force. Now we have two equations

$$f_{yb}A_s = \frac{T}{dk} \tag{37}$$

$$0{,}7f_{ub}A_s = \frac{T}{dk} \tag{38}$$

The combination is possible because we are still in the elastic region of the torque – tension relation. The reason is that the torque coefficient doesn’t change. After reaching the plastic region the relationship is not linear anymore and the coefficient is unknown. For example, this principle doesn’t work when the full tensile strength is concerned.

The torque that is needed to reach the yield strength of a bolt can be calculated from the equation

$$T_y = f_{yb} A_s d k \tag{39}$$

The torque that is needed to reach 70% of the ultimate tensile strength of a bolt can be calculated from the equation

$$T = 0{,}7 f_{ub} A_s d k \tag{40}$$

For each bolt type, both torques were calculated. In Table 25 has been presented the torques for reaching the yield strength and in Table 26 has been presented the torques for reaching 70% of the ultimate tensile strength. Additionally, both torques were subdivided into five categories. Torque T_{min} is calculated by using the torque coefficient in the lower scatter of the as-received condition, T is calculated by using the mean coefficient for as received bolts, T_{at} is calculated by using the coefficient for AT-2000 Vaseline, T_{gr} is calculated by using the coefficient for G-Rapid Plus paste and T_r is calculated by using the coefficient for rusted and salted bolts. For lubricated and rusted bolts, the mean values were used. Torque coefficients for HPM bolts were taken from Table 21.

Table 25. *Torque needed to reach the yield strength.*

Yield Strength					
Bolt	**T_{min} [Nm]**	**T [Nm]**	**T_{at} [Nm]**	**T_{gr} [Nm]**	**T_r [Nm]**
HPM 20	437	507	447	387	982
HPM 24	754	873	771	667	1694
HPM 30	1502	1740	1535	1328	3374
HPM 39	3397	3935	3472	3004	7631

Table 26. *Torques needed to reach the Eurocode recommendation of 70% from tensile strength.*

0,7 * Tensile Strength					
Bolt	**T_{min} [Nm]**	**T [Nm]**	**T_{at} [Nm]**	**T_{gr} [Nm]**	**T_r [Nm]**
HPM 20	337	390	344	298	756
HPM 24	581	672	593	513	1304
HPM 30	1157	1340	1182	1023	2598
HPM 39	2616	3030	2674	2313	5876

It can be seen how much the needed torques for rusted bolts differs from the other torques. It should be noted that these are used only as demonstration. Probably other malfunctions may occur before the suggested torque is reached. For example, it is unlikely that for rusted HPM 20 anchor bolt, it is safe to use 950 Nm torque and there are no problems. However, T_{min}, T, T_{at} and T_{gr} can be considered quite reliable.

5.4 Torque – tension curves

Torque – tension curve is an essential part of understanding the relationship between torque and tension. In this subchapter, these curves have been analysed. The torque - tension curves were determined for each test specimens that were considered as reliable results. The same test points were used as earlier when the torque coefficients were determined but with a few exceptions in each case. Here the results for PPM bolts have been left out from this public version.

At first, the torque – tension curves have been compared to the yield torques. These were calculated in last subchapter. In the second part has been analysed what kind of effect the bolt diameter and bolt type have to the curves. Finally, there is comparisons how the lubrication and corrosion change the torque – tension curves.

5.4.1 Yield strength

Torques that are needed to reach the yield strength have been calculated in Table 25. Here only T_{min} and T have been used and the torque – tension curves have been made from as received bolts. Vertical lines are used to indicate the location where the yield torque has been reached. Now also the first test point, points where multiplier is used and the plastic region is included to the data. Plastic region includes a lot of information and is therefore important part of the analysis.

In Figure 47 and Figure 48 are torque – tension curves and corresponding yield torque lines for as received HPM 20 and HPM 24 bolts. From these figures can be seen that the yielding starts somewhere close to the minimum and mean torque. However, because of the scatter, it is not perfectly clear where the yielding starts exactly. These curves indicate that the yield torques calculated earlier seem to be realistic.

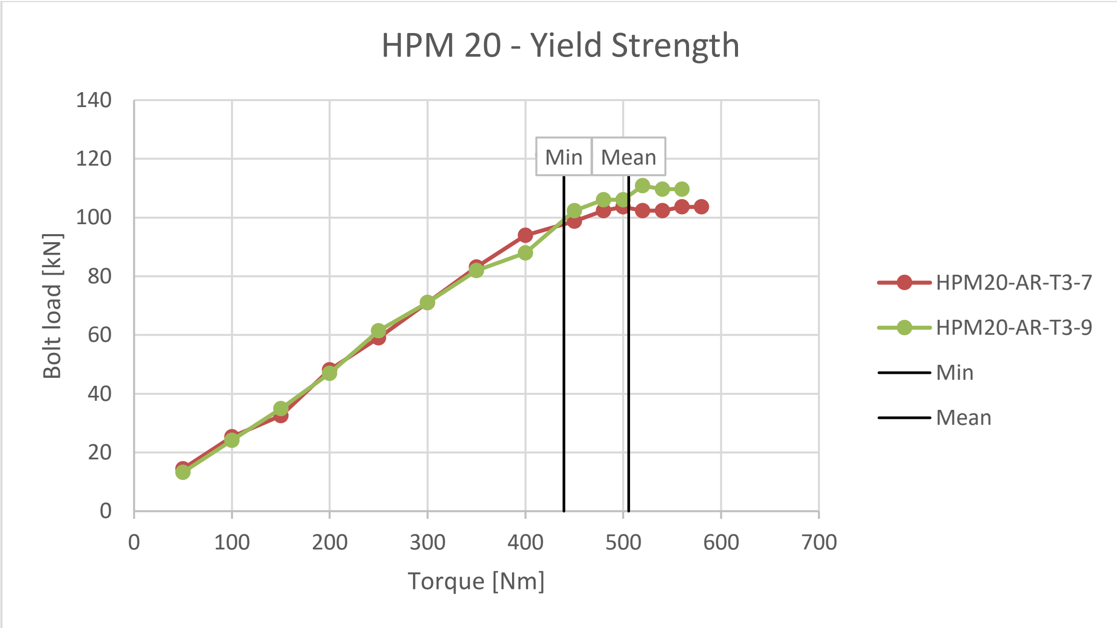

Figure 47. *HPM 20 torque – tension curves compared to the needed torques for reaching the yield strength.*

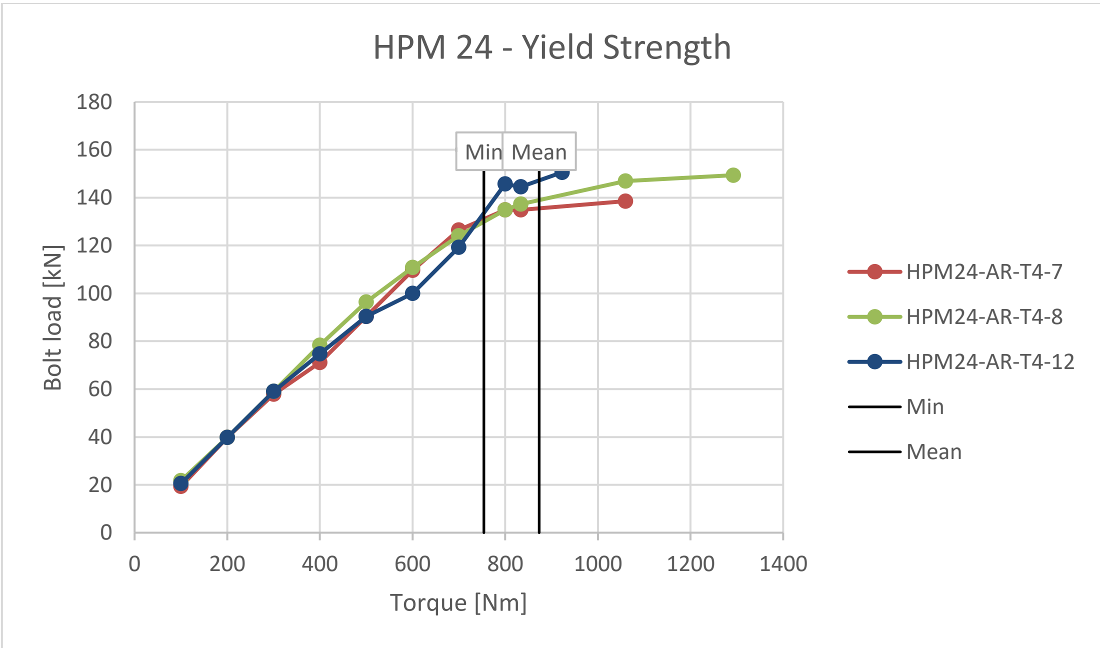

Figure 48. *HPM 24 torque – tension curves compared to the needed torques for reaching the yield strength.*

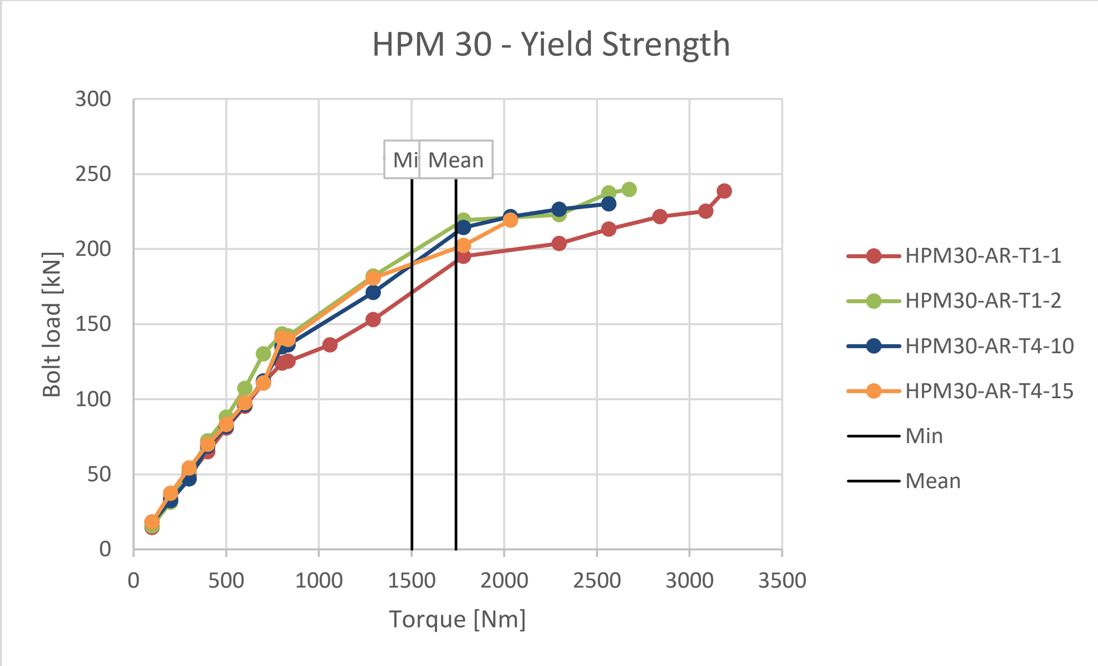

Figure 49. *HPM 30 torque – tension curves compared to the needed torques for reaching the yield strength.*

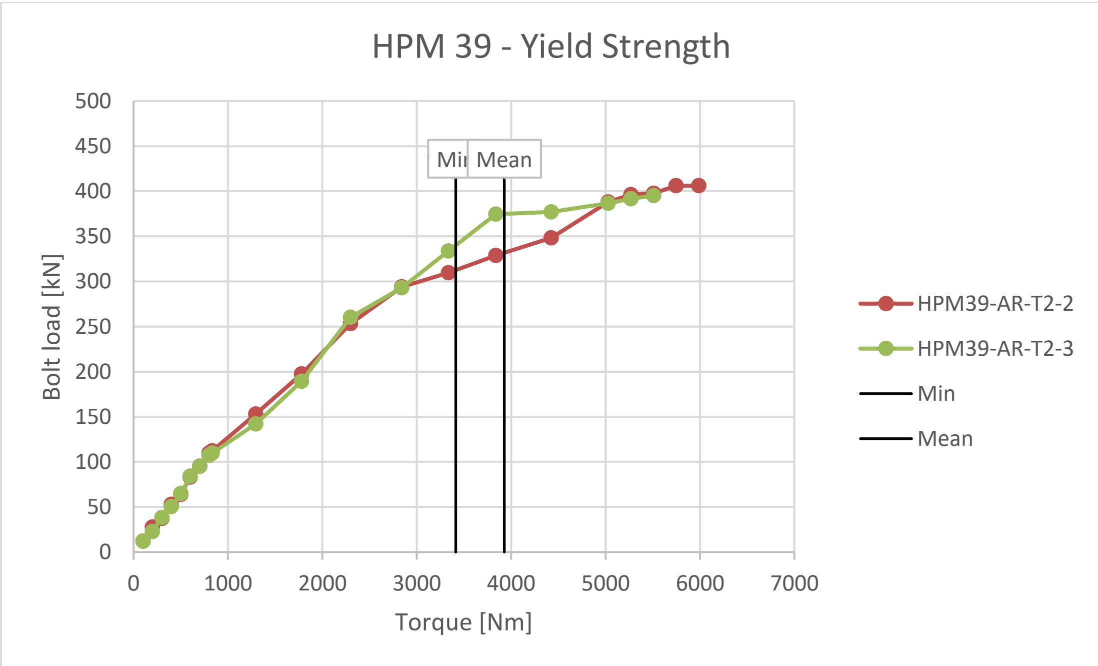

Figure 50. *HPM 39 torque – tension curves compared to the needed torques for reaching the yield strength.*

In Figure 49 and Figure 50 are the torque – tension curves for as received HPM 30 and HPM 39 bolts. It seems that estimating the yield point becomes more difficult when the bolt size increases. The major cause is the torque multiplier. Since the torque wrench range ends at 800 Nm torque, larger bolts need the multiplier to reach the yield point. This increases the scatter between the test points and makes the curve more difficult to estimate. As a result, smaller bolts have more reliable torque – tension curves.

So, what we can learn from these torque – tension curves? First thing is that there is not too much scatter in the curves. Only couple of test points in the HPM 39 curves had significant scatter. This means the test set-up was successful. The second factor we can learn is that determining the yield point gets more difficult when the bolt size increases. When the HPM 20 and HPM 24 curves were studied, our calculations for yield torques seem to be realistic. However, when looking the other curves, it is hard to say. The last thing to be noted is that the torque multiplier makes the curve less accurate and therefore more difficult to analyse.

5.4.2 Bolt size

In this subchapter there is a figure that has been organized by the bolt size. The comparison has been made for as received HPM bolts. In this analysis, only the test points that have 800 Nm torque or less have been used. The reasons for this is that now the figures are easier to compare since the range is the same for all the bolts. Furthermore, the inaccuracies caused by the torque multiplier can be ruled out.

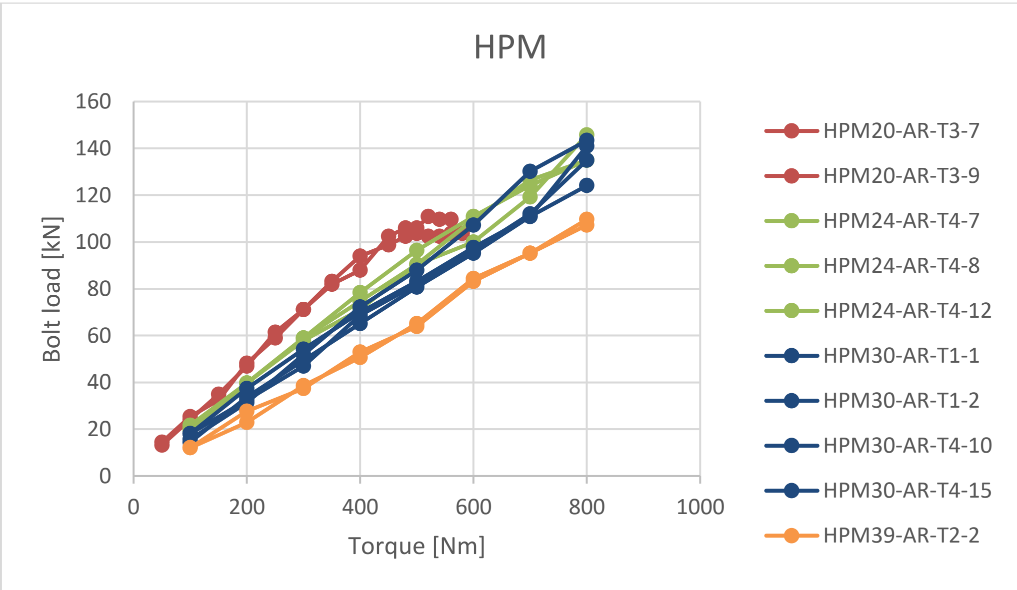

Figure 51. *Torque – tension curves for HPM bolts with torques 800 Nm and less.*

As the Figure 51 shows, the bolt size has a significant impact to the torque – tension relation. Smaller bolts increase the slope of the curve. But there is a simple explanation for this since the equation that describes the torque – tension relation consists bolt diameter as a variable. Therefore, the diameter has a direct impact to the relationship.

5.4.3 Bolt condition

In this subchapter, the impact of lubrication and corrosion to torque – tension curve is studied. This time the data also includes the test points that has torque more than 800 Nm and the points in plastic region. Here the yield torque lines have been used similarly as we did in Subchapter 5.4.1. However, now only the mean torque coefficients have been used and the lines have been drawn also for lubricated and rusted curves. These torque lines have been added to the figures just to demonstrate the differences and make the search for yield point a little easier. Additionally, all the conditions have their own “colour code” to separate the curves easily from each other. Black means as received, orange means rusted, red means rusted and salted, light blue means AT-2000 lubrication and darker blue means G-Rapid Plus lubrication.

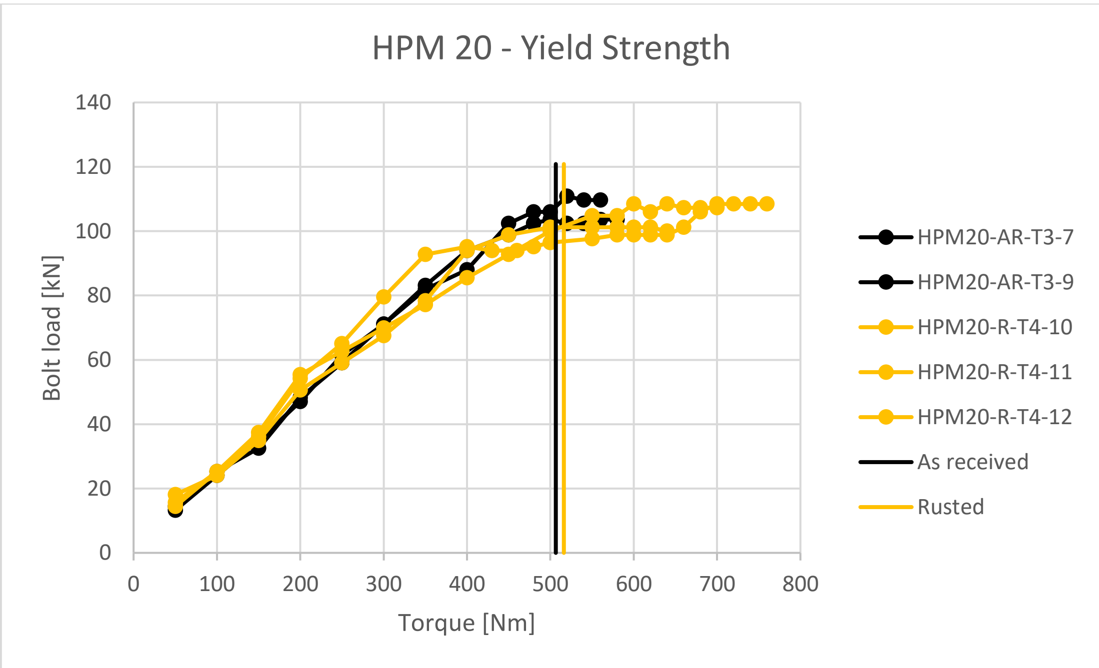

Figure 52. *HPM 20 torque – tension curves and yield torques.*

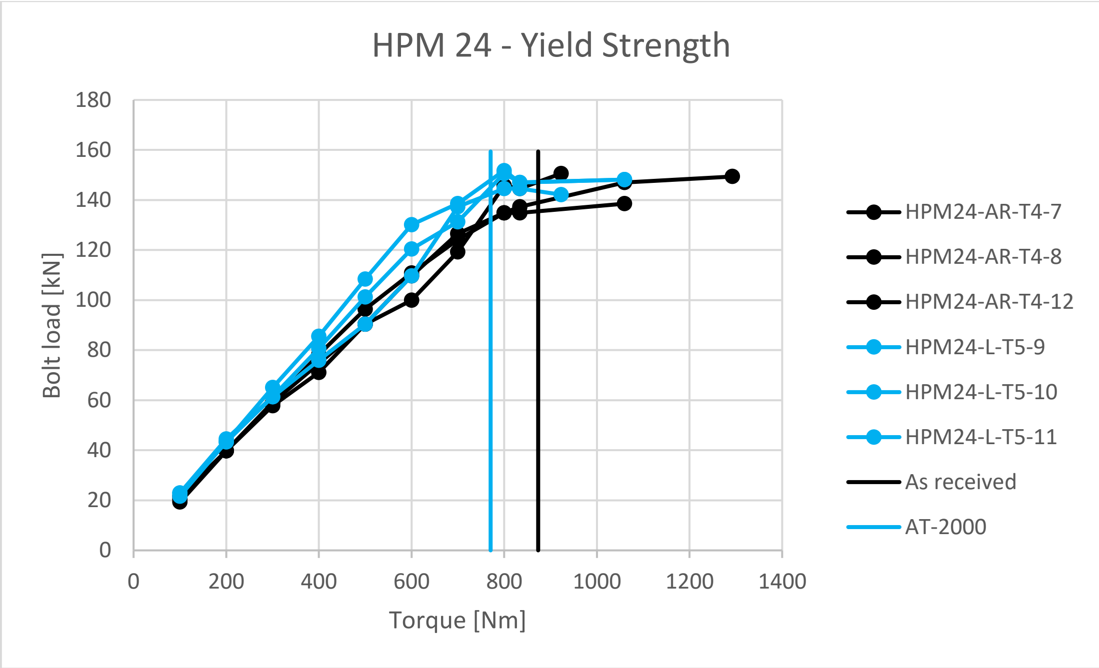

Figure 53. *HPM 24 torque – tension curves and yield torques.*

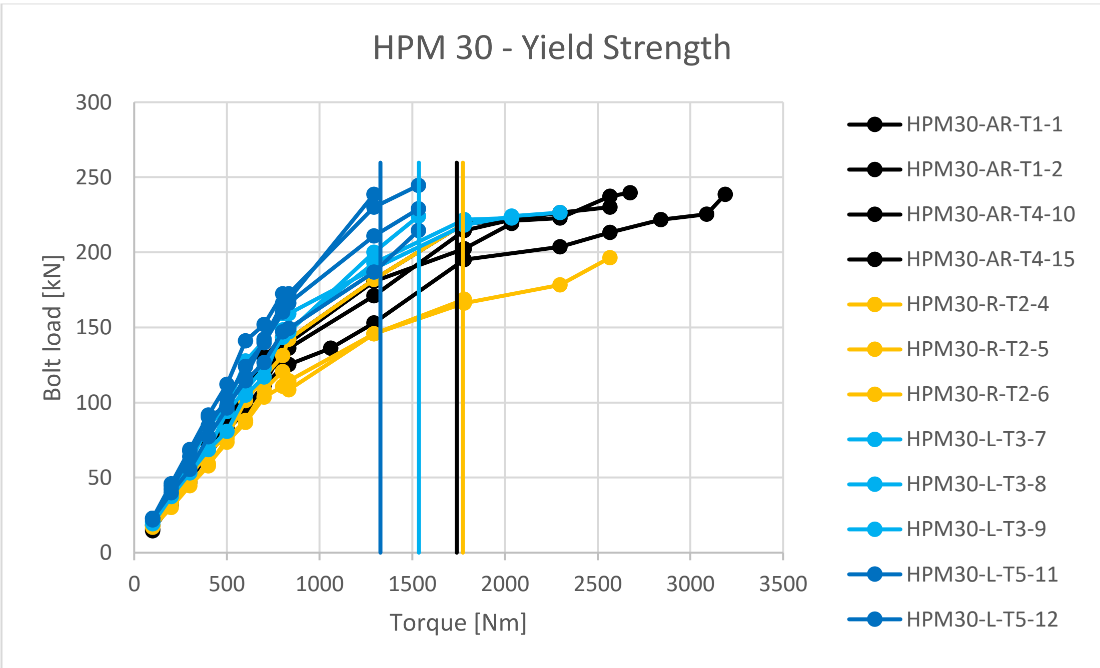

Figure 54. *HPM 30 torque – tension curves and yield torques.*

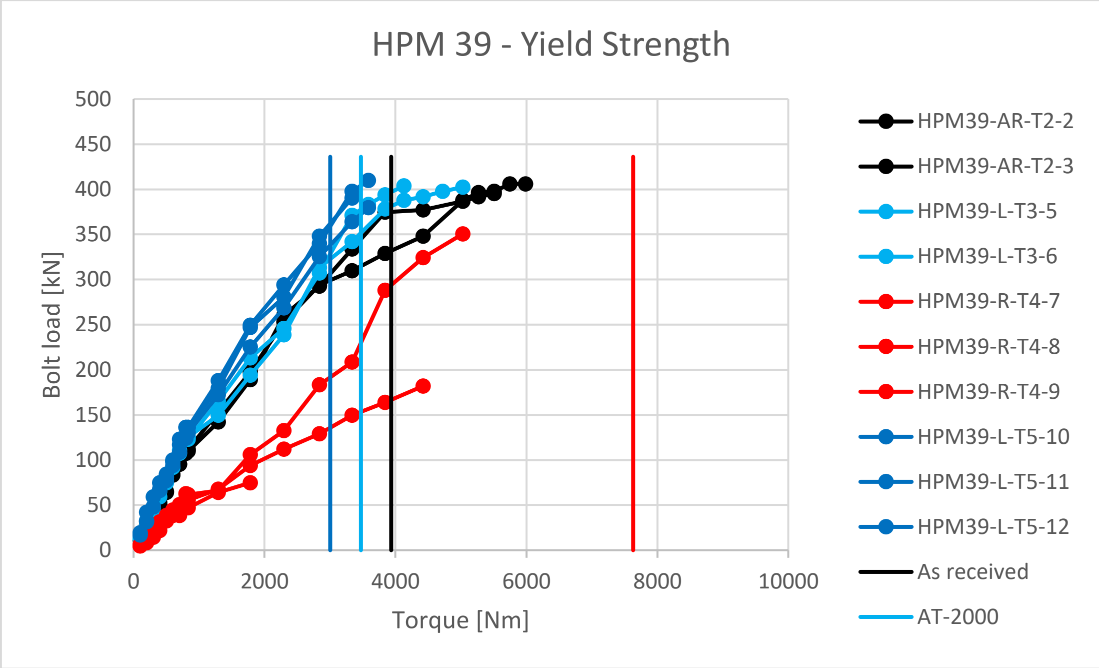

Figure 55. *HPM 39 torque – tension curves and yield torques.*

The lubrication makes the slope steeper and corrosion makes it less steep. As a result, the lubrication shifts the curve to the "left" and corrosion shifts it to the "right". This clearly changes the yield point in a bolt. The yield point is achieved earlier with a lubrication and later when there is rust in a bolt. However, "the peak" in a curve doesn't get any higher or lower because the tensile strength in a bolt is dependent on material class, not the bolt condition. Typically, some natural variations will always occur, but this is caused by the tolerances in bolt's dimensions and material class.

As it could be expected, G-Rapid Plus has a bigger impact to the torque – tension curve than AT-2000 and so does the salt bath over the rusted bolts. Additionally, from the figures it can be seen that especially the corrosion causes a lot of uncertainties. One good example is the rusted HPM 39 bolts since these gathered rust very effectively. The result can be seen from the Figure 55. Here the corrosion had a huge impact, making the bolts basically unusable.

When the proper torque is designed, it should be known what kind of lubrication is used. Additionally, if there is a rusty bolt, it should be known how deep the rust is or the rust should be avoided entirely. The bolt condition plays major role in relationship between torque and tension, especially if the critical joints are concerned.

5.5 Standard deviation and scatter

Standard deviation is used to measure how large is the variation between the test points. It basically tells us how big deviation from the mean value is considered as "normal". Any test points that are outside of the standard deviation, is considered as rare cases. In addition, low standard deviation tells that the points are close to the mean value and high standard deviation tells that the points have more variation and are further away from the mean value. The standard deviation can be calculated from the equation [50]

$$SD = \sqrt{\frac{\sum |x - \mu|^2}{N}} \tag{41}$$

where x is the test point to be calculated, μ is the mean from test data and N is the number of test points. Standard deviation and scatter have been calculated from the torque coefficients. The same test points were used as determining tables in Subchapter 5.2. Here the scatter means how big difference was between the lowest and the highest calculated torque coefficients.

Table 27. *Standard deviation and scatter for tested HPM bolts as received condition.*

Bolt	Test specimens	Standard deviation	Scatter
HPM 20	2	0,0084	0,03
HPM 24	3	0,0125	0,04
HPM 30	4	0,0110	0,03
HPM 39	2	0,0111	0,04

Table 28. *Standard deviation and scatter for tested HPM bolts.*

Condition	Test specimens	Standard deviation	Scatter
As received	11	0,0156	0,07
AT-2000	8	0,0197	0,08
G-Rapid Plus	7	0,0124	0,06
Rusted	6	0,0147	0,06
Rusted (salted)	3	0,0719	0,28

From Table 27 can be seen that there are only minor differences when the bolt size changes. For Table 28 has been calculated standard deviation and scatter for different bolt conditions. It seems that lubrication or slight rust doesn’t have a big impact to the deviation. However, deep rust has a significant effect. This confirms the earlier belief that corrosion will increase the uncertainties in torque – tension relation drastically.

5.6 Breaking a bolt

The limits were tested with one PPM 30 bolt. This was done by increasing the pressure in hydraulic jack. Unfortunately, using the hydraulic jack to create tension to the bolt, made all the relevant torque – tension measurements impossible. But it was possible to see what happens when the bolt is loaded past its capacity. The result can be seen in Figure 56.

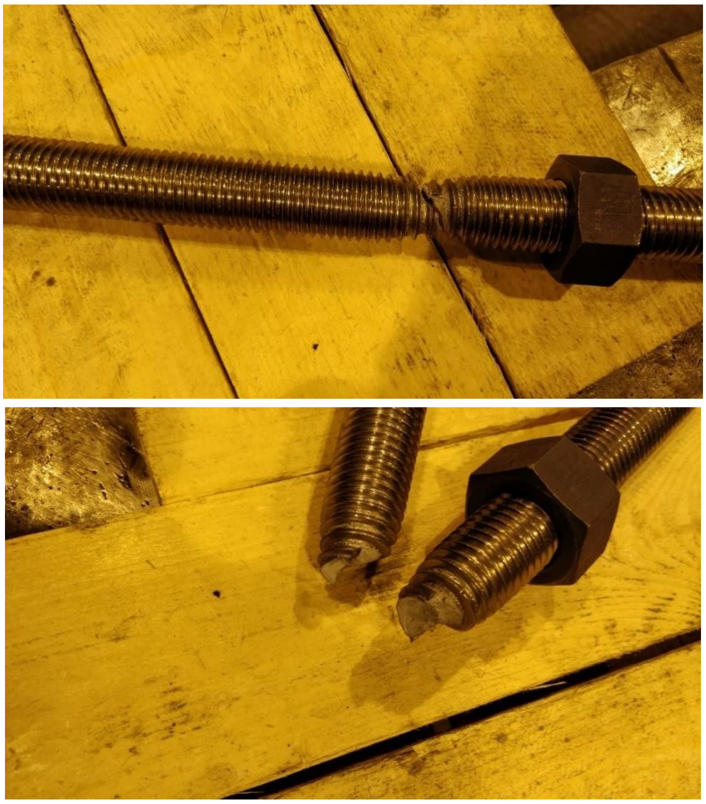

Figure 56. *Broken bolt.*

This is a good example of what happens when the bolt load increases too much. At first, the tension is evenly distributed in a bolt. Then from some weaker point, the cross section starts to decrease causing the tension force to increase. Once the force increases past a certain level the bolt will break from this weak point.

5.7 Margin for error

As mentioned a couple of times before, there are some inaccuracies in the test results. When the torque multiplier was assembled, there was enough time for short-term relaxation to occur. On the other hand, when the multiplier was used, a support was needed. The support carried some of the load and caused an error to the results. For short-term relaxation and multiplier support, the errors were determined experimentally.

The torque multiplier is much more inaccurate than using the torque wrench alone. It was experimentally tested how much is the difference from the range of scatter and standard deviation. Additionally, there were some estimations given in Chapter 3 and Table 6 about the control accuracies. These were compared to the test results and conclusions were made.

Other sources for error are equipment and operator accuracies. Equipment accuracies were given by the manufacturer and operator accuracy was given in Bickford's book [3]. These were combined with the errors that were determined experimentally and the total margin for error was calculated.

5.7.1 Short-term relaxation

Relaxation happens in a bolt all the time but majority of it occurs during the first few minutes. Torque multiplier assembly takes a couple of minutes and therefore short-term relaxation has enough time to affect the results. For this reason, a discontinuity point occurs at the time when the assembly was made. In this case it was the test points with 800 Nm torque.

Table 29. *The bolt load before and after the torque multiplier assembly. The relaxation percentage was calculated from the load decrease.*

Bolt	Before	After	%
HPM24-AR-T4-12	120	117	2,5
HPM30-L-T3-7	132	129	2,3
HPM30-L-T3-8	123	117	4,9
HPM30-L-T3-9	122	118	3,3
HPM30-AR-T4-10	113	110	2,7
HPM30-L-T5-11	138	136	1,4
HPM30-L-T5-12	140	129	7,9
HPM30-L-T5-13	124	120	3,2
HPM30-L-T5-14	143	140	2,1
HPM30-AR-T4-15	116	114	1,7
HPM39-R-T4-7	39	36	7,7
HPM39-R-T4-8	46	45	2,2
HPM39-R-T4-9	51	49	3,9
HPM39-L-T5-10	113	110	2,7
HPM39-L-T5-11	113	111	1,8
HPM39-L-T5-12	108	101	6,5
			3,5

After some tests, it was decided to measure how much relaxation happens during the multiplier assembly. At first, the measure would be taken when the 800 Nm torque was reached. Then the second measure was made after the torque multiplier was assembled.

Using these two values, the relaxation percentage for each test specimen was calculated. However, this consists only the test specimens that were studied after the decision for relaxation measurements was made. The results have been gathered to Table 29 and as for the mean percentage we calculated 3,5. Even though, the greatest relaxation occurs during the multiplier assembly this error estimation can be used with all the test points, before and after the assembly. It should be noted, that the pressure readings in the table are directly from the hydraulic jack monitor. Preload can be calculated from these by using the Equation 25.

5.7.2 Multiplier support

When the torque multiplier is used, a support is needed. However, the support takes part of the load that is meant to be carried by the hydraulic jack. The multiplier carries part of the load through "an arm" which has a contact surface with the support. For this reason, the load given by the hydraulic jack and pump is too small.

Table 30. *Pressure readings before and after the removing of the torque multiplier.*

Bolt code	800 Nm pressure	Before	After	%
HPM24-AR-T4-7	112	122	125	30
HPM24-AR-T4-12	121	125	126	25
HPM30-R-T2-4	109	160	162	4
HPM30-R-T2-5	100	170	174	6
HPM30-R-T2-6	111	163	167	8
HPM30-L-T3-7	132	196	200	6
HPM30-L-T3-8	119	185	190	8
HPM30-L-T3-9	123	186	190	6
HPM30-AR-T4-10	112	191	194	4
HPM30-L-T5-11	138	190	192	4
HPM30-L-T5-12	133	198	201	5
HPM30-L-T5-13	113	178	181	5
HPM30-L-T5-14	143	203	205	3
HPM30-AR-T4-15	117	182	186	6
HPM39-R-T4-7	41	62	68	29
HPM39-R-T4-8	47	151	163	12
HPM39-R-T4-9	52	291	306	6
HPM39-L-T5-10	113	324	331	3
HPM39-L-T5-11	113	340	347	3
HPM39-L-T5-12	103	315	321	3
				9

The error for the support was determined experimentally. After reaching the last test point the readings were marked down. Because the removing of the multiplier changed the digit in the monitor for a little, another reading was marked down without the multiplier. The list of the results can be found in Table 30. Like it was mentioned in the last subchapter, these readings are from the monitor and therefore not the same thing as bolt tension.

The table consists readings before and after the torque multiplier had been removed. Additionally, there is a measurement point at 800 Nm torque. This additional measurement point is needed because the support takes part of the load only when the multiplier is used. Therefore, the pressure that is achieved before the torque multiplier assembly needs to be ignored. This is done by reducing additional measurement point from the "before" and "after" values.

The percentage tells how much the load increased after the torque multiplier was removed. This was done only to the part of the load that has been achieved after 800 Nm torque. It was calculated that the mean difference is 9%. This means that the support carries approximately 9% of the load achieved by the multiplier.

5.7.3 Multiplier inaccuracy

As we know, the torque multiplier gives much more inaccurate results than the torque wrench alone. In Chapter 3 and Table 6 some tightening accuracies were presented. The table suggests for torque wrench ±30% and for torque wrench with multiplier -70% to +150% accuracies. This would mean that the torque wrench is 2,3-5 times more accurate than the torque multiplier.

For Table 31 have been collected scatters and standard deviations from the test points. This was done only to the HPM 39 bolts because these had enough test points in elastic region that were achieved by the torque multiplier. Therefore, it is possible to make the comparison between the torque wrench and multiplier. This was also possible to do for the PPM bolts, but these results are not included to this public version. However, the results from PPM comparisons were very similar.

Table 31. *Scatter and standard deviations for torque wrench and multiplier.*

	Wrench		Multiplier		Scatter relation (%)	Deviation relation (%)
Bolt/Condition	Scatter	Standard deviation	Scatter	Standard deviation		
HPM 39						
As received	0,04	0,0111	0,11	0,0293	261	264
AT-2000	0,03	0,0103	0,09	0,0267	267	260
G-Rapid Plus	0,04	0,0116	0,08	0,0235	175	203
Rusted (Salted)	0,28	0,0719	0,27	0,0840	96	117
					200	211

A percentage difference was calculated for both scatter and standard deviation. This value means how much big are the inaccuracies with torque multiplier comparing to inaccuracies with torque wrench. For example, 50% would mean that the inaccuracies with torque wrench are two times greater than with the multiplier, 100% would mean that the inaccuracies are the same and 200% would mean that the inaccuracies with torque multiplier are two times greater. It was found that the scatter is 2,00 and standard deviation is 2,11 times greater when the torque multiplier is used. Unlike with the accuracy presented in Chapter 3 (-70% to +150%), there was no notable difference between the "negative accuracy" and "positive accuracy". We can assume in our calculations that the use of the torque wrench is 2,2 times more accurate than the use of the torque multiplier and it is the same for both negative and positive accuracies.

5.7.4 Total margin for error

Combining the factors from chapters 4 and 5, the total margin for error can be calculated. Equipment accuracies are given by the manufacturers. For all the torque wrenches accuracy of ±4% is given and for the torque multiplier the accuracy of ±5% is given. Operator accuracy is assumed here to be ±10%, the same was used in Bickford's book [3]. In the last subchapter, the scatter with the torque multiplier was found to be 2,2 times greater than with the torque wrench. This means that controlling the multiplier is more demanding. Therefore, the accuracy for operator with multiplier is assumed to be ±10% * 2,2 = ±22%. Error caused by the multiplier support was found to be 9% and for the relaxation it was 3,5%. These were determined experimentally in subchapters 5.7.1 and 5.7.2. All the estimations are presented in Table 32.

Table 32. *Factors affecting the accuracy of the tests.*

Factor	Accuracy
Torque wrench	±4%
Torque multiplier	±5%
Operator	±10%
Operator with multiplier	±22%
Multiplier support	±9%
Relaxation	-3,5% to 0%

The total margin for error can be calculated from the equation [3]

$$V_{TOT} = \sqrt{V_A^2 + V_B^2 + V_C^2 + \cdots + V_n^2} \tag{42}$$

where V_{TOT} is the total accuracy, V_A is the factor A accuracy, V_B is the factor B accuracy and V_C is the factor C accuracy and so on. Therefore, the total margins for error with the torque wrench are

$$V_{TOT,WR+} = \sqrt{4^2 + 10^2} \approx +11\% \tag{43}$$

$$V_{TOT,WR-} = -\sqrt{4^2 + 10^2 + 3{,}5^2} \approx -11\% \tag{44}$$

And the total margins for error with the torque multiplier are

$$V_{TOT,MP+} = \sqrt{4^2 + 5^2 + 22^2 + 9^2} \approx 25\% \tag{45}$$

$$V_{TOT,MP-} = -\sqrt{4^2 + 5^2 + 22^2 + 9^2 + 3{,}5^2} \approx -25\% \tag{46}$$

The results indicate that the total margin for error is ±11% with the torque wrench and ±25% error with the torque multiplier. It can be seen how the operator accuracy dominates the error estimation. The results also confirm that most of the analysis should be done to the test points where only torque wrench is used to minimize the inaccuracies.

In Table 6 some control accuracies were presented. This means the accuracy of which the control method produces the bolt tension. Because of the use of the hydraulic jack and pump, the bolt tension was known for an adequate accuracy at all times. Therefore, the only major source for an error was the tightening process itself. This is why the control accuracy was ignored from the error calculations.

6. INSTRUCTIONS TO THE BOLT DESIGN AND CONSTRUCTION SITE

These instructions are based on our tests and the notes from the Bickford's book [3].

Factors that should be noticed in bolted joint design:

1. The whole bolting assembly should be only from one manufacturer.
2. Acceptable range of preload needs to be selected.
3. Friction needs to be controlled and known to prevent too much scatter in achieved preload.
4. Anticipated range of preload can be determined based on friction and assembly factors.
5. Tightening method and tools should be chosen based on needed accuracy, optimizing costs and past experiences.
6. Preload stability after assembly process needs to be ensured.

Factors that should be noticed when tightening a bolt:

1. Bolt condition needs to be checked and protected from environmental effects like rain to prevent the friction scattering too much friction.
2. Bolt corrosion should be avoided, at least when long time period is concerned. This could be done by oil or tape at the bolt's end.
3. Operator needs to be well trained for bolt tightening and understand the importance especially when critical joints are concerned.
4. Tools should be calibrated according to the instructions.
5. If lubrication is used, the use should be consistent in the whole connection.
6. Uniform torque should be used if possible.
7. As many bolts should be tightened simultaneously as possible. Also, tightening sequence needs to be noticed.
8. Possible changes and errors should be monitored during the tightening process.

Factors that should be noticed when turn measurement is involved:

1. Calibration tests are needed.
2. Turn is measured often as a relative turn between a nut and a bolt. That's why it is important to prevent the bolt from rotating and distorting the results.
3. Tightening in the first phase needs to be adequate in order to be sure that the snugging has been reached.

Factors that should be noticed when stretch measurement is involved:

1. Stiffness and elasticity of the bolt should be monitored.
2. Bolt's end should be checked of possible roughness on the surface that can affect the accuracy of the measurements.
3. Temperature changes should be monitored during the tightening.

Factors that should be noticed when direct preload measurement is involved:

1. Tensioner needs smooth base so that consistent results can be achieved.

Factors that should be noticed when ultrasonic measurement is involved:

1. To get full use of it, this method needs a lot of skills.
2. The device needs to be properly calibrated before use.
3. Bolt roughness may cause inaccuracies to the results.

7. CONCLUSIONS

Bolt is clamping together two or more things and an inadequate clamp basically means an unreliable joint. Therefore, the clamping force can be considered as the most important single factor in bolted joints. This is created by preloading the bolt. Torque is added to the nut, the nut rotates, the bolt stretches and causes preload to the bolt and compression to the joint members. However, achieving an exact value of preload is found to be impossible and the results will always scatter some because of many uncertainties. For this reason, typically the minimum and maximum tightening torques are given as it has been done to HPM and PPM anchor bolts. Too low or high torques may lead the joint into failure.

One of the uncertainties is the friction. Most of the torque is needed to overcome the friction and even a small change in the friction coefficient may cause a big effect on torque – tension relation. Higher friction than anticipated, will cause that the desired preload level is not achieved and lower friction than anticipated, will cause that the preload will be higher than expected. Other uncertainties are external loads, relaxation, temperature changes and tool accuracy. These all can increase the scatter of the achieved preload. During our tests, especially the short-term relaxation and tool accuracy caused some errors.

There are three major questions to be solved when designing preloaded bolted joints. At first, what is the acceptable range of preload so that the bolt joint will work? This is dependent on selected joint type, how critical it is and the purpose of the structure. Secondly, what is the anticipated range of preload? The friction scatter, tightening tools and bolt tolerances among many other inaccuracies should be considered. The third question is how the preload will change over time and the stability in joint is still adequate? There will be changes in preload because of changes in temperature and external loads for example. Relaxation will also decrease the amount of preload, mostly during the first minutes after the tightening process.

There are different options how to control the preload in bolted joints. Torque control, turn control and combined control (torque-turn) are all based on tightening torque even though different variables are measured. Torque control is the most commonly used because the low costs and easy tools. These three methods are not very accurate, even though with the combined control some problems can be detected during tightening. When high accuracy is needed it may be justified to use another preload control. Stretch control, direct preload control and ultrasonic control are based more on bolt's geometry and the preload can be controlled directly by using the bolt factors rather than torque factors. These methods are more accurate but will be also more expensive. Expensive preload control should be used only when it is justified. Otherwise the costs will increase needlessly. We used the torque control in our tests, but additional angle measurements were made.

Torque coefficient is used to describe the torque – tension relation and is determined experimentally. It should be noted that this is not the same as the friction coefficient. The torque coefficient includes the friction but also other factors like operator and tool accuracy. Like preload and friction coefficient, the torque coefficient scatters too. Part of this thesis was to determine the torque coefficient for HPM and PPM anchor bolts. The results didn't offer any big surprises. For HPM bolts the coefficient was calculated to be 0,21. Often for as received bolts the torque coefficient 0,2 is used and therefore our results were anticipated. In addition, the bolt size didn't change the coefficient significantly.

As it was expected the lubrication decreased the torque coefficient value. Molykote G-Rapid Plus was selected as a low friction lubricant and AT-2000 Spray Vaseline was selected as a basic lubricant. Both of these were used in high portions. G-Rapid Plus had a bigger impact and the coefficient was determined to be 0,16. For AT-2000 the coefficient was 0,18. The scatter in lubricated bolts did not seem to be any higher than with the as received bolts.

As a contrast to the as received and lubricated bolts, bolts with rust were chosen to be tested. At first, we had problems with causing deep rust to the bolt. The first tests resulted 0,21 torque coefficients, the same value as with the as received bolts. The major reason was the limited time period. Letting the bolts to collect enough rust outdoors was too slow project considering the given time this thesis was done. The corrosion was speed up with salt bath. After the bath, bolts were left outside again. This had a major impact and now the torque coefficients were around 0,40. However, because we had to speed up the process there is no experience how long the bolts can be left outside to affect the weather before any severe damage is caused. But these results confirm that the corrosion had a big impact to the torque – tension relation and may cause the joint into failure. Corrosion also increased the scatter and standard deviation. The standard deviation was approximately five times larger than with as received bolts. This means that controlling the rusty bolts accurately is much more difficult than controlling as received or lubricated bolts. Therefore, the rusted threads should be cleaned with brush to decrease the friction. Or better yet, the bolt should be protected with oil or tape at the bolt's end, before the corrosion starts to affect at all.

Besides the torque coefficient, torque – tension curves are used to describe the relationship. When calculating the torque coefficients, it was found that the bolt diameter won't change the results. However, when the curves were determined the diameter had a significant impact. The reason can be found from the equation that describes the torque – tension relation ($T=Fdk$). The equation tells that the bolt diameter is separate variable from the torque coefficient. This means that bolt diameter will have an effect to the torque – tension relation but not to the torque coefficient. When the curves were drawn, it was found that the estimation is easier to do with smaller bolts. For larger bolts, the torque multiplier was used and the scatter increased. This made the estimation more difficult.

Some of the subjects were studied only slightly in this thesis and would probably need further research to gather reliable data. For these subjects our results are only as reference because of small sampling. At first, it would be interesting to see how the corrosion time affects the torque coefficient. Because we had to speed up the corrosion process this were never properly tested. It is still unknown how much time it would take to cause some serious damage to the threads. Alternatively, it could be tested how the method and place to store the bolts changes the torque coefficient.

The other subject was the torque – angle or angle – preload relation. These relations were more like an estimation because the use of hydraulic jack and inaccurate angle measurements. The hydraulic jack increased the effective length of the bolt and caused a minor error. A few control tests were made without the hydraulic jack, but this control group had a small sampling. Therefore, no reliable conclusions could be made and this part was left out from the analysis.

All the tests for this thesis were done at Peikko's facilities in Lahti during December 2017 – February 2018. The set-up included anchor bolt, two nuts, two washers, two steel plates, hydraulic jack and pump. Torque multiplier and torque wrenches were used to tighten the bolt. Wooden pieces and deep throat clamp were used to support the set-up. The test specimens were designed to fit the set-up and were made from HPM 20, HPM 24, HPM 30, HPM 39, PPM 30, PPM 39 and PPM 52 bolts.

The set-up was improved gradually when the new problems occurred. Some of the problems were compensated by determining the margin for error. Like often in experimental work, some of our tests were failures and could not be part of the analysis. Only the test data that could be considered reliable was used. However, the tests were successful. There were still enough data to proper analysis and the torque coefficients were very close to the values we could be expecting. For example, G-Rapid Plus were given torque coefficient 0,15 by the manufacturer and we got 0,16, while for as received bolts the coefficient 0,2 is widely used and we got 0,21. For these reasons we can assume that our results are reliable. As a margin for error it was calculated for torque wrench ±11% and for torque multiplier ±26%.

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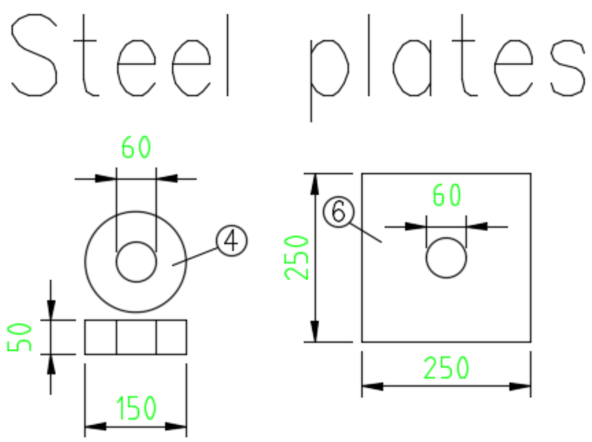
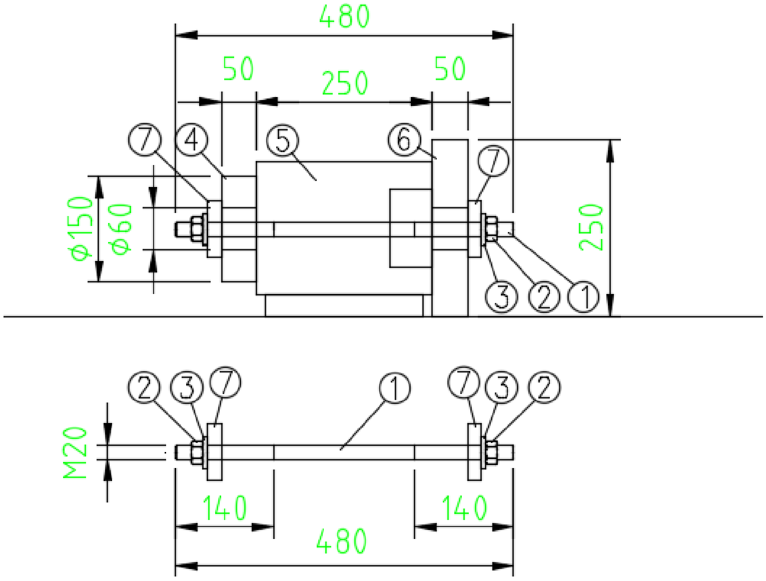
APPENDIX A: DIMENSIONS OF TEST SPECIMENS

The figures have been drawn 16.11.2017. These consists:

1. Anchor bolt
2. Nut
3. Washer
4. Round steel plate
5. Hydraulic jack
6. Circular steel plate
7. Column shoe base plate



HPM 20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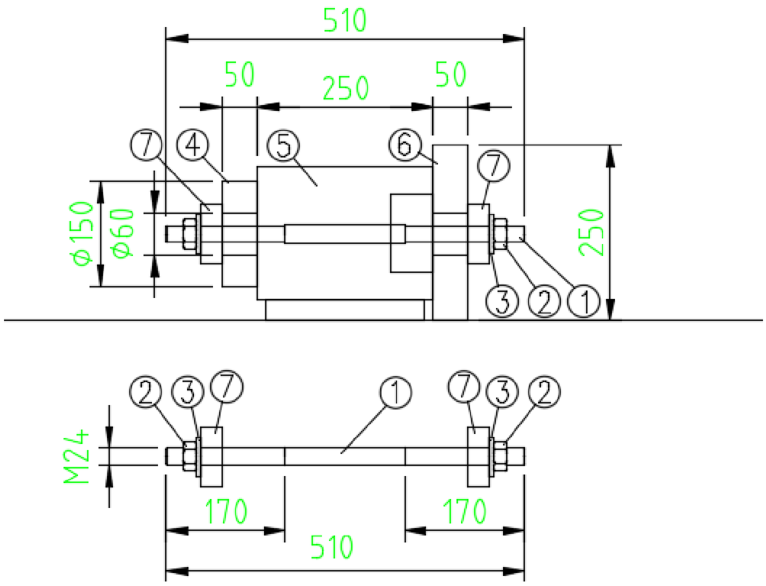
250
50
⌀150
⌀60
250
M20
140
140
480

HPM 24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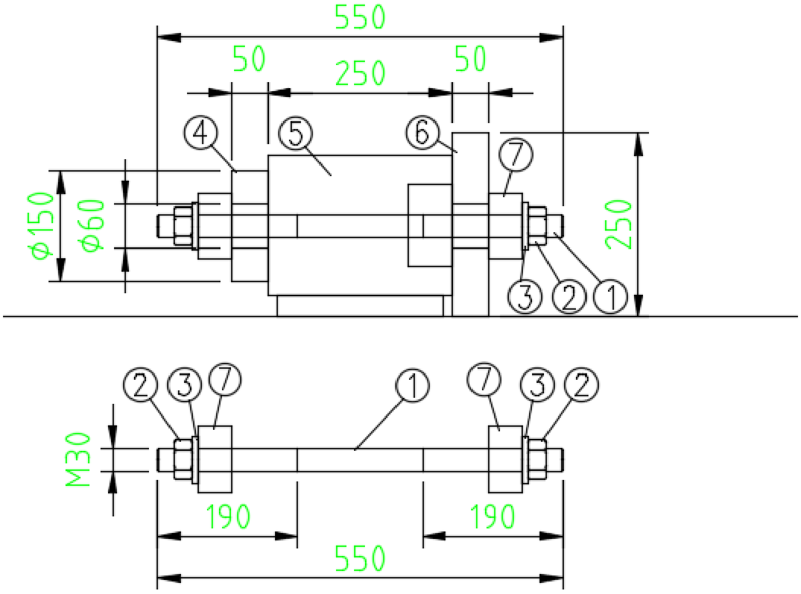
250
50
⌀150
⌀60
250
M24
170
170
510

HPM 30



550
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250
50
4
5
6
7
ϕ150
ϕ60
250
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1
2
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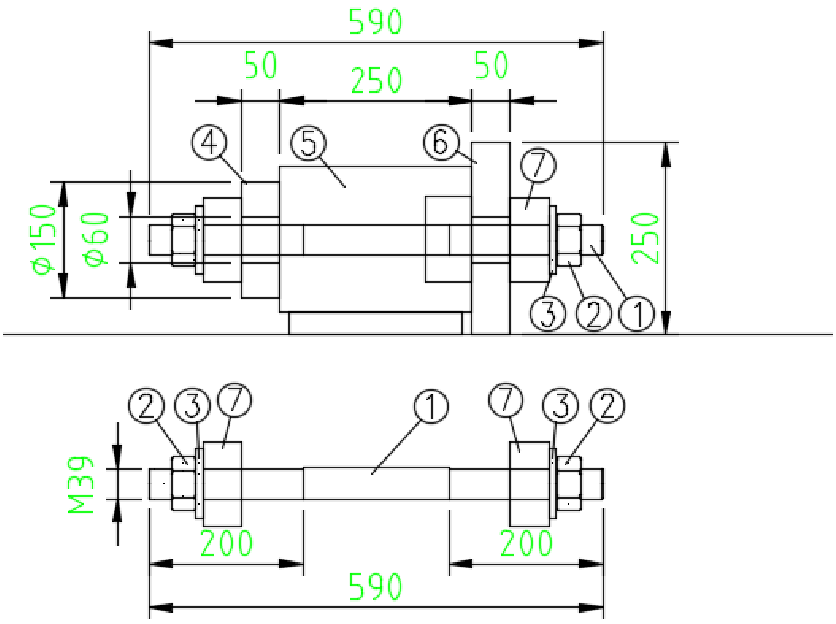
M30
190
190
550

HPM 39



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M39
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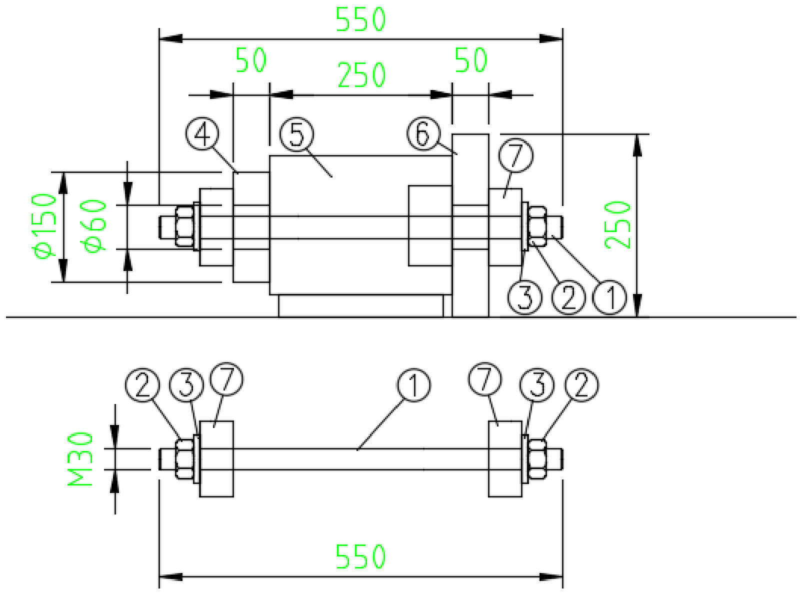
200
590

PPM 30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φ150
φ60
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2
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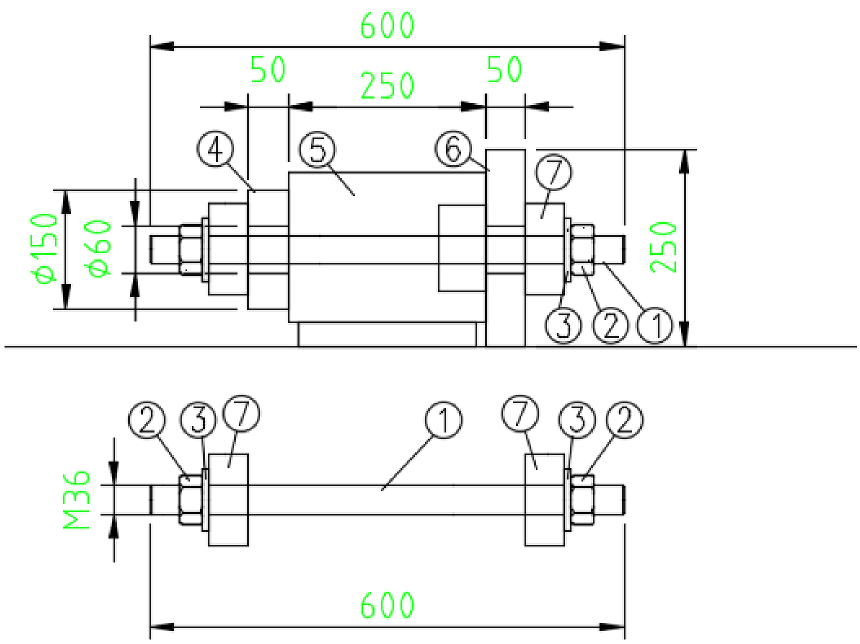
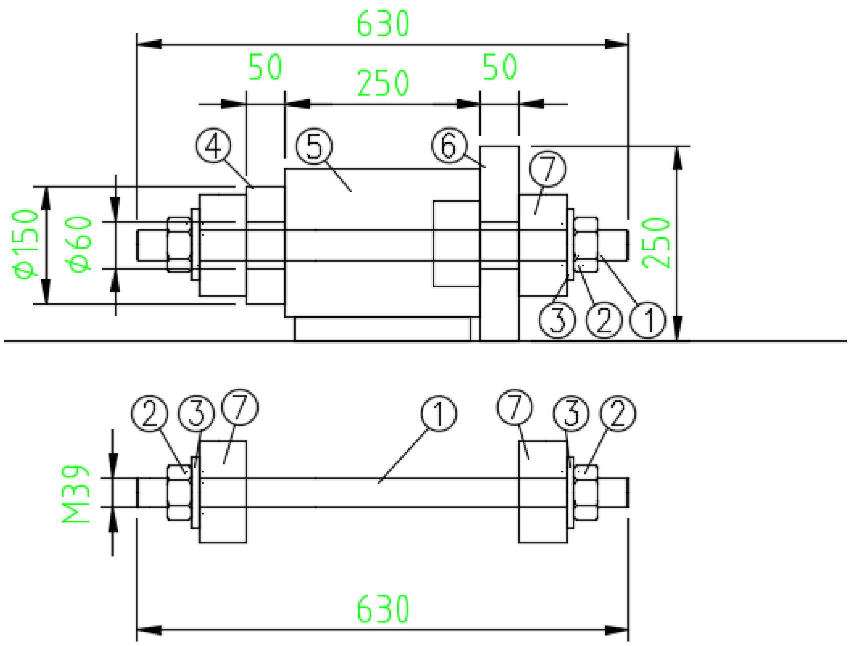
2
M30
550

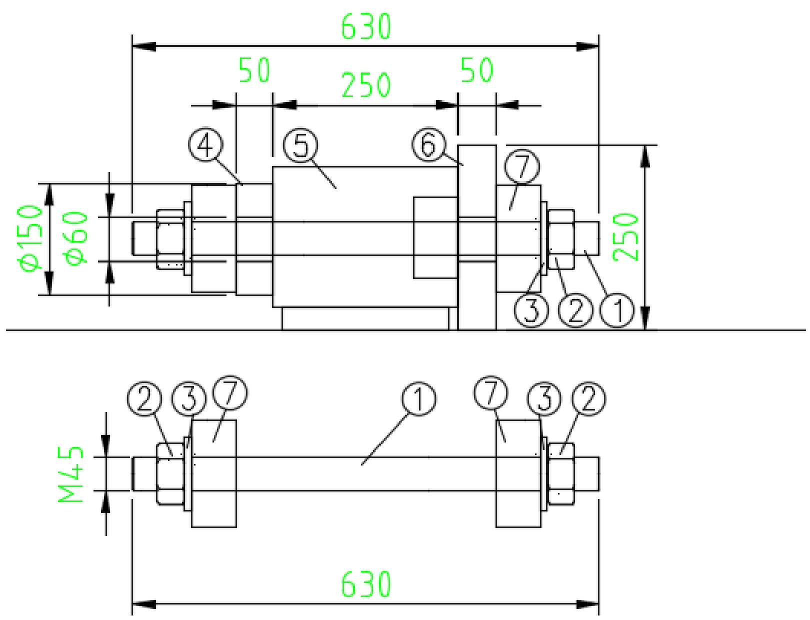
PPM 36


600
50
250
50
4
5
6
7
φ150
φ60
250
3
2
1
2
3
7
1
7
3
2
M36
600

PPM 39


630
50
250
50
4
5
6
7
Ø150
Ø60
250
3
2
1
2
3
7
1
7
3
2
M39
630

PPM 45


630
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50
4
5
6
7
Ø150
Ø60
250
3
2
1
2
3
7
1
7
3
2
M45
630

PPM 52