Constructive and experimental examination of reliability and robustness in dynamically stressed bolted joints

Master Thesis in Energy Technology

Thermal Engines

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Preface

This master thesis was written with the help of supervisors Karl-Heinrich Grote and Lars Magne Nerheim of the Department of Mechanical and Marine Engineering at Western University of Applied Sciences. Thank you for the valuable input, guidance and general discussions all the way through this thesis.

Secondly, I would like to express my gratitude to my manager, Per Varhaug, for giving me the opportunity to finalize my master degree with the completion of this thesis. I would also like to thank my colleagues at KvK for their assistance during this time.

Finally, yet equally important, I would like to thank my family and friends for their support and motivation through this period of combined work and study.

Abstract

Loosening and failure of bolts and nuts is, and has historically always been, a challenge for industries all over the world. This thesis examines the bolted connections of a local plough manufacturer (hereafter referred to as KvK). KvK has had challenges with loosening and failing bolts for several decades.

The structure of this thesis is to first go through the bolting philosophies of KvK and then to look at bolt literature. These two will then be compared and an evaluation of the different variables in a bolted connection are described and analyzed. Changes in two variables, the bolt preload and the friction diameter, were evaluated theoretically. The change in preload was then simulated and then experimentally tested at the KvK test centre for validation. The results show a significant improvement in resisting loosening with increased preload. A discussion regarding the results, result validity and a conclusion follows. The thesis rounds out with suggestions for further work, references and appendixes.

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Nomenclature

A_s = Stress Area [mm^2]

a = Marginal Slip [mm]

D_a = Chamfer Diameter [mm]

D_{Ki} = Inside Diameter of bearing surface at head or nut [mm]

D_{km} = Effective Friction Diameter [mm]

d = Diameter [mm]

d_2 = Pitch Diameter [mm]

d_a = Inside Diameter of plane bearing head area [mm]

d_h = Hole Diameter [mm]

d_{ha} = Chamfer Diameter of clamped parts [mm]

d_w = Outside Diameter of plane head bearing surface [mm]

E = Elastic modulus [GPa]

F = Preload [kN]

F_m = Assembly Preload [kN]

k = Nut factor

L_c = Clamp Length [mm]

ΔL = Bolt Elongation [mm]

M = Torque [Nm]

M_a = Assembly Torque [Nm]

M_g = Thread Torque [Nm]

M_k = Surface Friction Torque [Nm]

M_L = Loosening Torque [Nm]

P = Pitch

α = Flank Angle [deg]

φ = Helix Angle [deg]

$\rho`$ = Angle of Friction [degree]

μ_g = Coefficient of friction in threads

μ_k = Coefficient of friction under bolt head/nut

1. Introduction

The plough is an essential tool for most farmers. Ploughing mechanically limit disease and ensures a good yield during harvest. Being dragged behind ever more powerful tractors through rough soil, the plough experiences severe loads during its lifetime. The turning motion of a reversible plough and the impacts from stones in the soil lead to a tough life for the bolts and nuts holding the plough together. Historically, ploughs has been plagued with bolt loosening and failure. Workers at a local plough production site (hereafter referred to as KvK) remember market reports regarding bolt failures going back several decades. In figure 1, a plough is shown in use.

Figure 1 – Plough being dragged behind a tractor

Today loosening and failing of bolts has become one of the major negative feedbacks from customers of KvK. The marketing division reports that competitors use “frequent bolt failures” as a sales argument against KvK ploughs. For a time, farmers having to replace bolts was not paid much attention too, as the price of replacing a failed bolt was considered low. The quality expectations of today`s farmers have greatly increased and they no longer tolerate the unexpected downtime that a failed bolt brings. With KvK being a high quality brand, the failure of bolts also leads to a loss of perceived quality of the entire brand. Structured work towards identifying, reducing and eliminating root causes for bolt loosening and failure has therefore started at KvK. For example; during the spring of 2018, the surface paint was identified as one of the main factors in causing loosening and failure of bolts. A new paint is now being tested and internal processes are under evaluation to improve this situation.

Challenges with bolted connections under transient loading conditions is not a unique plough problem, bolted joints are getting increased focus worldwide. Vibrationmaster, a company selling test equipment for fasteners, reported in a newsletter that between 2013 and 2017 an 800% increase in scientific articles published on the subject of self-loosening fasteners.
A copy from their newsletter is attached in Appendix A.

This thesis aims to build on the work being done at KvK by providing a theoretical evaluation of the factors affecting a bolted joint. The bolting theory and practices of KvK will then be measured against the literature with the objective to propose improvement of the current practices. One proposed change is then put to the test at the KvK Test Centre. After the test, a discussion of the results and conclusion will follow, as well as suggestions for further work.

Due to the limited timeframe of this thesis, the scope of the investigation has to be restricted. During chapter 2.2, variables will be chosen for testing, and hence variables that are not selected will neither be tested nor further discussed. In additions to the variables not being tested, factors such as the quality of bolts delivered from the supplier, special solutions (locking mechanisms etc.) and design changes on the ploughs themselves will not be examined.

2. Theoretical background

During this chapter, bolts and nuts at KvK and the bolting philosophy used at KvK will be described. This will be followed up with general bolting theory, which will lead to an evaluation of which variables this thesis will propose changes to. In chapter 3, the methodology of these changes will then be described.

2.1 Bolt & Nuts at KvK

KvK uses metric bolts, which range in sizes between M6 to M30. By far the most used size is the M20 bolt, followed by the M16 & M24. The M20 bolt is also the bolt most associated with loosening and failure. The smaller bolts are not associated with much failure and the larger bolts are rarely used.

For bolt sizes M16-M24, KvK bolts are based on standard metric coarse threaded bolts, according to ISO 4016-2011, but with some exceptions. The wrench size of the M16 and M20 bolts have been altered to 27mm (standard is 24mm and 30mm respectively). The change was also done to the M16 and M20 nut. This was introduced in the 1960s for marketing reasons, as farmers would need fewer tools. The plough comes with only two wrenches, and the idea was that the farmer should be able to do everything with only these two wrenches.

There are also some changes to the length of the threaded part on some bolts, where the threaded region has been shortened. This is not the case for bolts tested in this thesis and therefore will not be discussed further.

Figure 2 below shows the M16, M20 and M24 with their respective nuts. For the M16 and M24 there is a plain nut and a metallic lock nut. The M20 has the same, but also has a flanged nut. This was introduced as the reduced size of the M20 nut was creating a too high surface pressure on certain areas. This point will be further discussed in chapter 2.3.

Figure 2– M16, M20 and M24 bolts and nuts at KvK

This thesis will focus on the M20, class 10.9 bolt. This is due to it being the mostly used bolt at KvK. The class 10.9 is a heavy duty class and indicates that the bolt has a maximal tensile strength of 1000 MPa and a nominal yield strength of 900 MPa (Fonas 1972).

KvK mainly uses high-strength classes for its bolts. Of the other high-strength classes, the class 8.8 is used, but the class 12.9 is less used due to experiences with hydrogen embrittlement.

With regards to material, the bolts are based on the ISO 898-1 standard. KvK is currently transitioning into using a Geomet coating on all of their bolts and nuts. This has been done to provide a good rust protection, a consistent coefficient of friction and to comply with EU regulations ("Chrome-six" coatings no longer allowed).

The bolts have a Geomet 500 coating and the nuts have a Geomet 321 base coating and then a PLUS® M friction coating. Datasheets for Geomet coatings can be found in Appendix B. Both of these coatings give a friction coefficient of 0.15 ± 0.03.

In Appendix C, KvK internal torque tables are shown in kilopond meter and Newton meter. The table differentiates between "black" bolts, slightly oiled *("svarte skruer")* and bolts with "passivated Zink" *("El. forsinkede skruer").* Today, passivated Zink is being replaced with Geomet coating, with the same torques being used. There are no longer black bolts of sizes M16-M24 being used at KvK today.

At KvK, the maximal torque value is often seen as the target, this means that for the M20 10.9 bolts, the aim is 599 Nm rather than 572 Nm, which is the average. This has emerged as a "best practice", which may indicate that KvK historically may have struggled more with creating enough clamping force, rather than overtightening the bolts.

2.2 General Bolt Theory

A bolt behaves similar to that of a stiff spring. When a bolt is tightened, energy is put into the bolt in the form of torque. This elongates the bolt, ever so slightly. Similar to the force in a spring trying pulling it back, the bolt creates a clamping force on the joint, holding it together.

When a bolt is tightened, the interaction between the bolt & nut and the joint it connects can be described with the following formula:

$$M = k * F * d \tag{1}$$

Where M = Torque, k = "nut factor", F = Preload and d = Diameter.
This is shown underneath in figure 3.

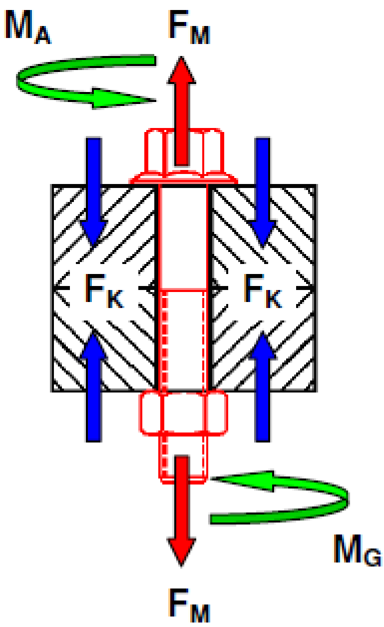

Figure 3- The torque-preload relationship (Drumheller 2008)

The figure shows, when torque is added, either at the bolt head or nut and using something to hold the other side you will get a clamping force, F_K on the joint and a preload F_m in the bolt.

Equation 1 is a simplified version of what is really happening, but it serves a good purpose in showing the physical effect: When torque (energy) is provided, some level of clamping force (F) is put on the joint. The uncertainty in the above equation is the “nut factor”. This factor acts as a “sum” of everything affecting a given connection. For a standard bolted connection, it is the sum of friction and the pitch of the bolt threads.

In equation 1, the bolt diameter was a factor. This is a practical equation and show that the relationship between torque and preload is affected by a change in diameter. More in-depth analysis often has the diameter inserted into the formula. The in-depth analysis consists of two parts, the first part is regarding the threads, given as (VDI-2230 2003):

$$\mathrm{M_g} = F_m * \frac{d_2}{2} * \tan(\varphi + \rho`) \qquad 2$$

Where M_g is the thread torque, F_m is the assembly preload, d_2 is the pitch diameter of the bolt thread, φ is the helix angle of the thread and $\rho`$ is the angle of friction in the threads.

For a standard metric bolt, it follows that:

$$\tan(\varphi) = \frac{P}{\pi * d_2} \qquad 3$$

$$\tan(\rho`) = \mu_g` = \frac{\mu_g}{\cos\left(\frac{\alpha}{2}\right)} \qquad 4$$

With P = Pitch, μ_g = coefficient of friction in the thread and α = flank angle.
With $\alpha = 60^o$, it can be simplified:

$$\tan(\varphi + \rho`) \approx \tan(\varphi) + \tan(\rho`) = \frac{P}{\pi * d_2} + 1{,}155\mu_g \qquad 5$$

This gives the total thread torque formula as:

$$M_g = F_m * \left(\frac{P}{2\pi} + 0{,}58 * d_2 * \mu_g\right) \qquad 6$$

Explanatory illustrations to follow the deriving of these equations are hard to find. In figure 4, the axial forces working on a bolt are shown:

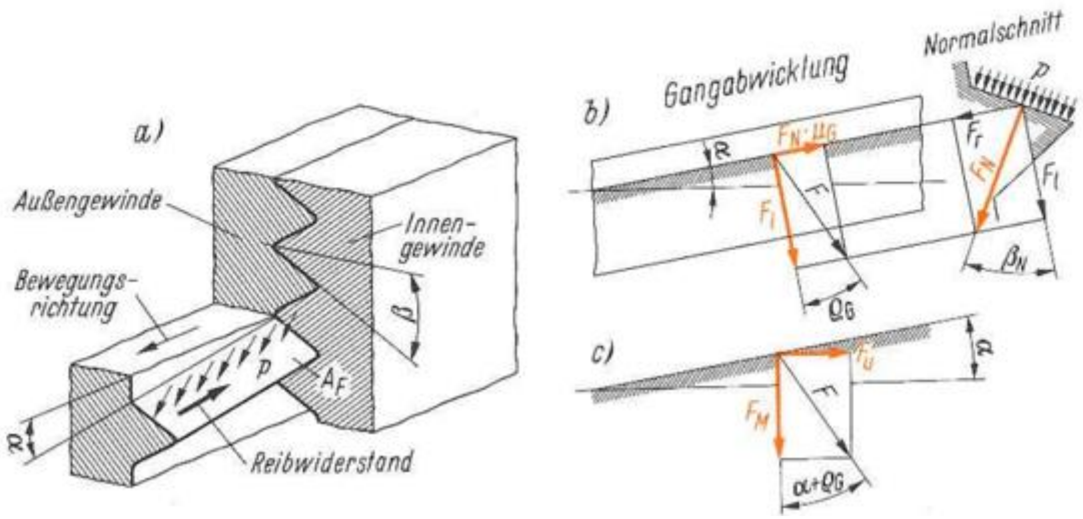

Figure 4 - Axial forces on a bolt leading to the bolt equation (Decker 2002)

Compared to the VDI annotation: $\varphi = \alpha, \rho` = \varrho_g, \mu_g` = F_n * \mu_g$

The second part is the torque required to overcome the friction between the bearing area and the head or nut. To do this the effective friction diameter needs to be known and can be calculated by

$$D_{Km} = \frac{d_w + D_{Ki}}{2} \quad (7)$$

Where D_{Km} is the effective diameter for the friction torque and d_w is the outside diameter of the plane head bearing surface of the bolt. D_{Ki} is given by:

$$D_{Ki} = \max(D_a, d_{ha}, d_h, d_a) \quad (8)$$

D_{Ki} is the inside diameter of the bearing surface at the head or nut defined by which diameter is the largest of the chamfer diameter of the nut D_a, the chamfer diameter of the clamped parts d_{ha}, the hole diameter d_h or the inside diameter of the plane bearing head area d_a.

This is shown in figure 5 below, which shows a KvK M20 bolt with a flange nut. When tightened on the nut, the red line shows D_{Ki} and the yellow line shows the $d_{w.}$ This visualizes the effect of flanges on nuts (and bolts) have on the tightening torque.
If these distances are measured, D_{Km} can be calculated.

Figure 5 – Friction diameter on a KvK flange nut

The total torque to overcome the friction in the surface area is given by:

$$M_k = F_m * \frac{D_{Km}}{2} * \mu_k \qquad 9$$

Where M_k is the surface friction torque required, and μ_k is the coefficient of friction in the bearing area. The total equation for tightening a bolt will therefore be:

$$M_a = F_m \left(\frac{P}{2\pi} + 0{,}58 * d_2 * \mu_g + \frac{D_{Km}}{2} * \mu_k \right) \qquad 10$$

With M_a as the assembly torque, equal to $M_g + M_k$.

This is related to equation 1, but the nut factor and diameter are replaced with more in-depth variables. For a standard bolt with friction values of 0.15, only roughly 10% of the added torque actually provides clamping force. This is given by the incline of the thread pitch; $\frac{P}{2\pi}$. Roughly 90% of the energy goes to overcoming friction.

One might therefore suggest decreasing the friction, to improve the situation, but the friction is what is holding the bolt in placed once it is installed. In fact, the thread pitch actually works to loosen the bolt after installment. This is why the loosening torque, barring no other influences, is always lower than the tightening torque. The equation is given as:

$$M_L = F_m\left(-\frac{P}{2\pi} + 0{,}58 * d_2 * \mu_g + \frac{D_{Km}}{2} * \mu_k\right) \qquad 11$$

Where M_L is the torque required to loosen the bolt.

In the next chapter, the variables in this equation will be discussed and changes proposed.

2.3 Variables Considered

Through equation 9, a list of variables can be made. Changes to these variables will affect how the bolted connection works. Here a brief discussion will be made regarding all of these variables.

Torque, Ma

The torque could easily be changed for a bolted connection, the only possible requirement is a bigger tool capable of delivering more torque to the bolt. One very seldom decide to change the torque alone. This is due to the relation between torque and preload, as show in equation 1 & 10. This means that changes in torque should come because a change to the preload or other changes, such as friction coefficients, are changed. This does however also work the other way around, if a change to any variable is to be made, this must be taken into account on the torque.

A second point regarding torque is not only the actual amount, but also how it is applied. A wide variety of tools exist, each with some benefits and drawbacks. In Appendix D, the VDI has a table of guide values for tightening factors (a function of preload scatter for a given tightening method). In addition, if the aim is to use more torque to get a higher preload, higher accuracy will be required to avoid failure of the bolt during the tightening process.

Assembly Preload, Fm

Similar to the torque, the preload can be easily changed. Generally, a standard is followed which gives the desired preload for a standard bolt. Most standards base this suggested preload by calculating a certain percentage of the yield strength of the bolt. Different standards exist which give different clamp forces. As noted for torque, a high percentage utilization of the yield strength must imply the use of a better, more accurate tightening method, as to ensure not to overtighten the bolt.

Preload is often listed as, if not the most, an important factor for a bolted connection.

Pitch, P

Pitch is the distance between the threads on a bolt. For a metric M20 coarse bolt, this distance is 2.5mm. For metric bolts, this can be changed from going from coarse threads to fine threads. For the M20 bolt, this would mean a change from 2.5mm to 1.5mm pitch. Alternatively, a change away from metric bolts could be implemented. This is highly unlikely to be done at KvK, and will not be considered in this thesis.

Pitch diameter of the bolt thread, d_2

This is table value for a given bolt and can be viewed in Appendix E. This variable is therefore only changeable with a change in bolt size. A change in bolt size would again affect clamping force, torque, bearing area and so on.

Coefficient of friction, μ_g & μ_k

The coefficients of friction for the thread and the bearing surface, μ_g and μ_k, can theoretically be changed easily. Today, suppliers are combining corrosion protection with coatings that give certain friction characteristics. As was shown in Appendix B, Geomet coating has a wide variety of friction coatings that influence these coefficients. In the real world, it can be more difficult. (Bickford 2008) notes that between 30 and 40 variables are identified to affect the friction seen in a fastener.

Friction is a very interesting variable, because when a bolt is tightened, the friction is what has to be overcome (unless hydraulic extension tightening is used). When the bolt is installed, the friction in the thread and under the bolt head/nut is holding the bolt in place. The VDI 2230 suggest friction class B, from the table in Appendix F, which gives coefficient values between 0.08-0.16 for both μ_g and μ_k.

Effective diameter of the friction torque, D_{Km}

The effective diameter can be changed by altering the surface area of the bolt or nut. Reducing this variable leads to less friction force which means a bolt requires less tightening torque. Increasing it requires more torque, and will have more friction to keep it in place once installed. Increasing this variable can be done with the use of washers, flanges or by reducing the maximal chamfer diameters or hole diameter of the connection.

D_{Km} is a factor that, barring introduction of flanges, will not fluctuate greatly. However, due to the KvK alteration of the M20 bolt, changing to for instance standard size could actually provide a change in loosening and tightening torque.

Other variables

There are also other factors which affects how the bolt behaves.
For instance Hooke`s law tells us the following:

$$\Delta L = \frac{F * L_c}{E * A_s} \qquad 12$$

Where ΔL is the elongation of the bolt, L_c is the clamp length, E is the elastic modulus and A is the stress area. If you have more elongation, any set loss due to relaxation or similar would mean a lower percentage of the total elongation. Therefore, more elongation is positive, which means a longer clamp length is desirable.

In the next chapter, where the effect of preload is discussed, a point will also be made of the bolt-to-joint stiffness ratio. This will also not be altered or further discussed. Other factors include using conical shaped holes. This will increase the surface area and introduce a wedge effect.

Altering the way a bolted connection work can also be done with any one of countless "special solutions". These generally aim at either to reduce stress concentrations or increase locking between either threads or surface area and bolt. However, as was noted in the introduction, this thesis will not evaluate suppliers, "special solutions" or design changes on the plough etc.

2.4 Proposed Changes

Preload

The choice of examining the preload as the first variable was due to the amount of literature claiming preload as the most important factor in a bolted joint. Just a few example are listed below:

"Improper preload can be a contributing factor in almost every situation " (Bickford 2008), in the context of bolt failure.
"Preload is the most economical way of preventing vibration loosening" (Ravinder 2013).
*"the following list can help construct a secure connection: ... *Increase the preload"* (Lange 2009).
"*The most important condition for a good and safe screw connection is correct preload"* (Tingstad 2018)

The bolting philosophy at KvK is heavily based on "Fonas Skruehåndbok" from the 1970s. This book summarized bolting theory and Norwegian Standards (NS) and was a renowned Norwegian book regarding bolt technology. One of the main principles that was adopted at KvK was the amount of preload a bolt should provide. The aim was to utilize 70 % of the yield strength of the bolts. With the stress areas given in Appendix E, the preload goals for class 10.9 M16, M20 and M24 bolts follows as:

$$M16 = 0{,}7 * 900\ MPa * 157mm^2 * 10^{-3}\frac{kN}{N} = 99\ kN \tag{13}$$

$$M20 = 0{,}7 * 900\ MPa * 245mm^2 * \ 10^{-3}\frac{kN}{N} = 154\ kN \tag{14}$$

$$M24 = 0{,}7 * 900\ MPa * 353mm^2 * \ 10^{-3}\frac{kN}{N} = 222\ kN \tag{15}$$

When any of these bolts are torqued in production, these are the desired results.

Preload has been shown to greatly increase vibration resistance, as seen below in figure 6. The figure shows the amount of cycles to failure as a function of the preload (given as a percentage of the ultimate strength).

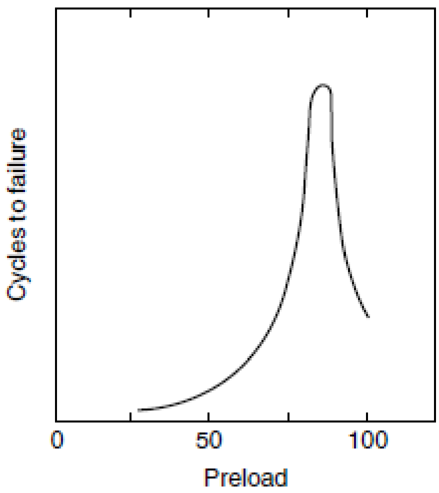

Figure 6 - Cycles to failure as a function of preload (Bickford 2008)

As can be seen by the sharp incline of the line, a small change in preload can drastically increase the amount of cycles. This can also be shown by the formula for calculating marginal slip (Lange 2009). The “marginal slip” is the displacement under the head of the bolt caused by the motion of the two clamped plates. This is then given as the displacement, in mm, that has to be exceeded in order to loosen the bolt.

$$a = \frac{F * \mu * L_c^3}{12 * EI} \qquad 16$$

Where a = marginal slip, F = preload, μ = friction coefficient under the bolt head, L_C = clamp length and EI = stiffness of the bolt.

This shows that increasing preload directly increases the resistance of the bolted joint against failure. This formula also highlights the importance of clamp length (notice that the term is to the power ^{3}!).

One way of illustrating how the preload increases resistance to failure is shown in figure 7 and 8 below. Figure 7 shows a joint diagram from (Bickford 2008), modified with axis lines. Change in length is shown on the x-axis and force on the y-axis. When a bolt is tightened, it is stretched and this creates the bolt preload and the clamping force on the joint. The first triangle, starting in O_B, shows the elongation of the bolt (ΔL) as a function of the force applied and goes up to the bolt preload (F_P). The rightmost triangle is the joint, starting in point O_J. The joint experience compression (negative elongation) and the clamping force is equal to the preload.

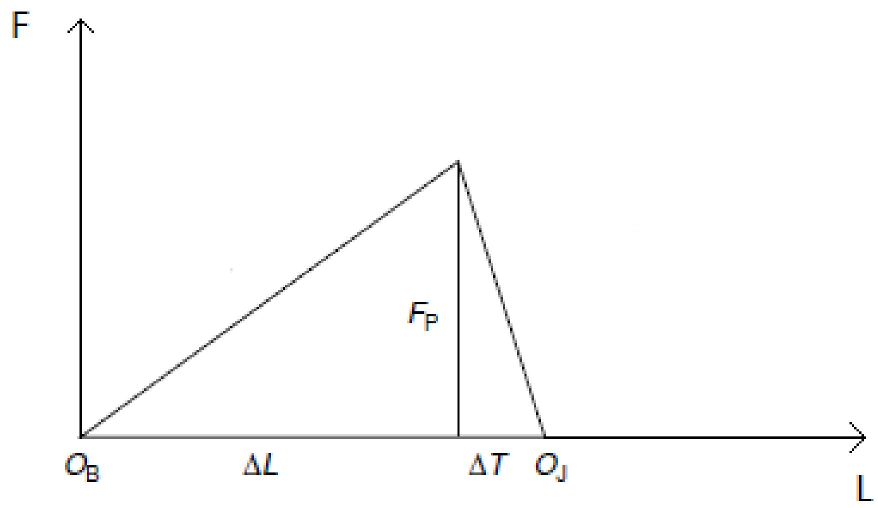

Figure 7 – Modified joint diagram (Bickford 2008)

In figure 8 below, a modified joint diagram is shown. An external load is being applied. This load will increase the elongation of the bolt, but it will also reduce the compression in the joint. The increased force on the bolt is shown with a big red line. The increase in load on the bolt is only a fraction of the total applied external load. This is due to much of the external load working against the joint compression. This is the principle that demonstrate how bolt preload can protect the bolt from external forces.

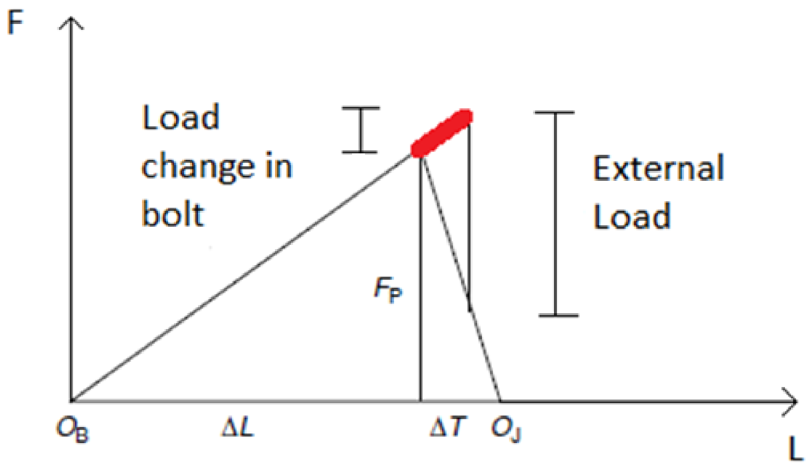

Figure 8 – Modified joint diagram with external load (Bickford 2008)

Figure 7 and 8 also highlight an important aspect, which is the ratio in stiffness between the bolt and the joint. The lower the stiffness of the bolt is compared to the joint, the less is the portion of an external load the bolt will have to endure. This is another possible variable for testing, but will not be further evaluated in this thesis.

As was noted in the beginning of this discussion, preload theory from Fonas Håndbok is being used at KvK today. The question then becomes what preload goals to test. Both my supervisor Karl Grote and contacts at VASI (Verktøy Industri AS – a supplier of high-end torque tools) suggested the VDI 2230 standard. This standard employs a much higher usage of the yield strength of any given bolt (90% compared to 70% in Fonas). A possible reasoning behind this is that Fonas is much older and so the tools available at the time most likely gave much bigger preload scatter compared to modern tools. In appendix G, the preload and torque values for metric shank bolts from VDI 2230 are shown.

One thing to remember is that increasing the preload does increase the mean stresses the bolt is experiencing. Preload is great for reducing the stress variations in a bolt, but caution must be taken not to exceed the endurance limit of the bolt.

If the test results shows a positive impact from increasing the preload goals, an evaluation of torque tools and procedures will be required at KvK to examine how this can best be implemented in production.

Overall, this thesis proposes the following hypothesis:

"Increasing the preload at KvK from their current standard, to the VDI 2230 values, will reduce loosening and increase the lifetime of their bolts".

The definition of "*increased lifetime*" will be solely based on the amount of cycles in the testing. This is not the most groundbreaking hypothesis, but according to literature, this is most likely the single variable that will improve bolting at KvK the most. As describe in chapter 2.2, this change in preload will be accompanied with a change in torque.

Effective diameter of the friction torque, D_{Km}

The effective diameter of the friction torque D_{Km}, often referred to as the surface bearing area, was the second variable chosen. The reasoning behind selecting this variable is the changes that KvK has done to its M16 & M20 bolts and nuts, referred to in chapter 2.1. Due to time limitations, this variable will not be examined, but a theoretical analysis was done. This is due to it being of great interest to KvK, as well for the general bolt literature as this reduced wrench size is an uncommon occurrence. This also highlights possibilities for further work when testing bolts at KvK.

This thesis analyzed three KvK bolts, three KvK nuts, three KvK flange nuts, three standard sized bolts and three standard size nuts were measured. If the measured D_{Ki} value was below 20.5 mm, which is the hole diameter at KvK, the hole diameter was used as measurement. The bolts are seen in figure 9 and average values are shown in table 1.

Figure 9 - Standard and KvK M20 bolts & nuts measured at KvK

KvK bolt	Outside diameter of bearing area = 26,05 mm Inside diameter of bearing area = 20,5 mm
KvK nut	Outside diameter of bearing area = 26,02 mm Inside diameter of bearing area = 20,83 mm
KvK flange nut	Outside diameter of bearing area = 33,58 mm Inside diameter of bearing area =20,5 mm
Standard bolt	Outside diameter of bearing area = 28,62 mm Inside diameter of bearing area = 20,84 mm
Standard nut	Outside diameter of bearing area = 29,63 mm Inside diameter of bearing area = 20,84 mm

Table 1 - Average values of the bearing area

Using equation 6, with the values from table 1 gives:

$$\text{KvK bolt } -\frac{26{,}05+20{,}5}{2} = 23{,}275\ mm \qquad 17$$

$$\text{KvK nut } -\frac{26{,}02+20{,}83}{2} = 23{,}43\ mm \qquad 18$$

$$\text{KvK flange nut } -\frac{33{,}58+20{,}5}{2} = 27{,}04\ mm \qquad 19$$

$$\text{Standard bolt } -\frac{28{,}62+20{,}84}{2} = 24{,}73\ mm \qquad 20$$

$$\text{Standard Nut } -\frac{29{,}63+20{,}84}{2} = 25{,}24\ mm \qquad 21$$

These are not monumental differences, but the use of a flange nut compared to a KvK standard nut increases the friction diameter with roughly 15%.
A point to notice is that if a flange nut is being used, but the bolt head is being tightened, the friction diameter of the bolt head will be the factor used.

Average values for both the bolt head and nut would give:
KvK bolt & nut – 23,4 mm
KvK bolt and flange nut – 25,2 mm
Standard bolt and nut – 24,99 mm

These values do not immediately tell much, but a point can be argued that the KvK bolt used in combination with a flange nut are more affected if an operator tightens the bolt head rather than the nut. The standard size has a much lesser deviation from the average when considering tightening from both sides. In most cases, the tightening happens on the nut, but exceptions exist. Either a change to make the sides more equally sized or a better tightening procedure, which ensures that the nut is tightened each time can be evaluated.

This could also be done with the introduction of a flanged bolt head. This is a quite interesting prospect, but due to KvK not using a standard size, this is not something that sits on the shelfs of suppliers. The cost might therefore be quite high compared to standard bolts.
With standard sized bolts and nuts, a bigger pool of suppliers could become available which potentially could decrease cost without decreasing quality.

The reduction in size has also led to problems with the M20 nut deforming during tightening, as seen below in figure 10 (Furre 2018). This deformation has often been seen during laboratory testing, but not much in production. This may point towards the torque tools in production not providing adequate torque to the nut, which is a statement several workers at KvK supported.

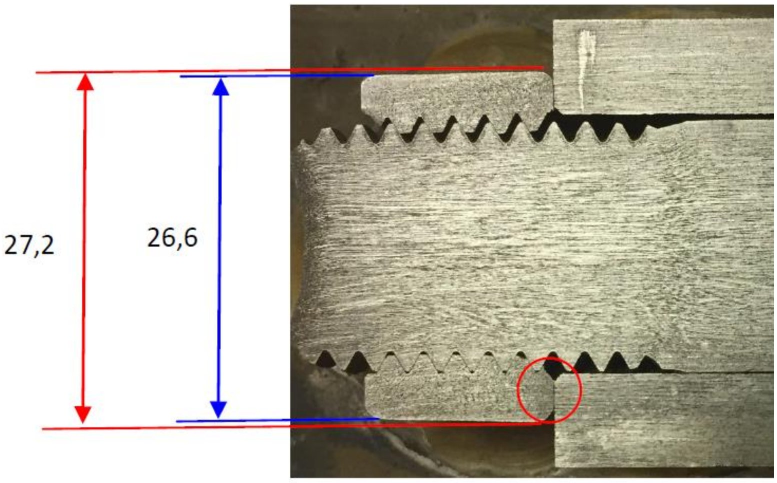

Figure 10 - Deformation of KvK M20 nut

Another issue is the surface pressure under both the bolt and the nut. From the measurements of the bearing areas taken and with a preload of 154 kN, the resulting pressure under the bolt head and nut can be calculated.

The equation is:

$$Pressure = \frac{F}{A} = \frac{154\,000}{\frac{\pi}{4}(d_w^2 - d_{Ki}^2)} \qquad 22$$

The pressures under the bolt head and above the nut will then be:

$$\text{Bolt head } - \frac{154\,000}{\frac{\pi}{4}(26{,}05^2 - 20{,}5^2)} = 759\ MPa \tag{23}$$

$$\text{KvK nut} - \frac{154\,000}{\frac{\pi}{4}(26{,}02^2 - 20{,}83^2)} = 807\ MPa \tag{24}$$

$$\text{KvK flange nut} - \frac{154\,000}{\frac{\pi}{4}(33{,}58^2 - 20{,}5^2)} = 277\ MPa \tag{25}$$

This shows an incredible effect of the flange to reduce surface pressure. The problem is that under the bolt head the surface pressure remains high. With an increase in preload, as is to be tested in this thesis, this pressure would naturally rise further. The goal for an M20 10.9 bolted connection is an increase in preload of roughly 16%, which would give a corresponding 16% higher surface pressure.

The high surface pressure was the reasoning behind the introduction of the flange nut at KvK. This was due to practical problems rather than an evaluation of the bolted connection. This is most likely the reason why it is only flange on the nut and only on certain bolts. KvK employees estimated the use of flange nuts was about 25% of the total usage of M20 nuts.

As a suggestion for further work, this thesis suggest evaluating standard size M20 bolts and nuts against the KvK M20 with reduced wrench size. A hypothesis could be:

"A standard size M20 bolt and nut will outperform a KvK M20 bolt & nut, in regards to loosening and failure"

As priorization went to testing the increased preload, this is mentioned in the conclusion as a suggestion for the further work at KvK.

3. Investigation Methods

This thesis attempt to validate the improvements from the proposed change in preload through two channels. The change is first simulated through a Finite Element Method (FEM) software. The test setup is a simulated copy of a bolt test setup at KvK, and can be seen in chapter 3.1.

In chapter 3.2, the same change in preload will is tested through parts of a full-scale life cycle test at KvK. A standard life cycle test was performed at KvK on one of the large ploughs, with clear bolt loosening problems. This plough failed during testing and new parts were ordered. Certain bolts were then chosen and then tightened to a higher preload and the test was restarted.

3.1 FEM Simulation

One year ago, a physical test setup was built for testing bolts at KvK. The test is a combination of two hardened L-shaped metal pieces. Two bolts tighten the components together, with a hardened washer on each side (to prevent surface damage). One side is locked to the test bench and the other is connected to a cylinder. The cylinder provides a cyclic force, which provides shear and bending stresses for the bolts.

In cooperation with KvK, this thesis chose to imitate this test through FEM. Technical drawings can be seen in appendix H. The physical setup is shown underneath in figure 11 and the simulated setup can be seen in figure 12. In figure 13 the simulated figure 13 the force working all through the middle hole on the leftmost piece is shown, as well as the fully locked constraint in the middle hole on the rightmost piece. The force being applied is 50 kN and it is applied linearly, meaning it goes starts at zero and works up to 50 kN.

Figure 11 – Physical test setup at KvK

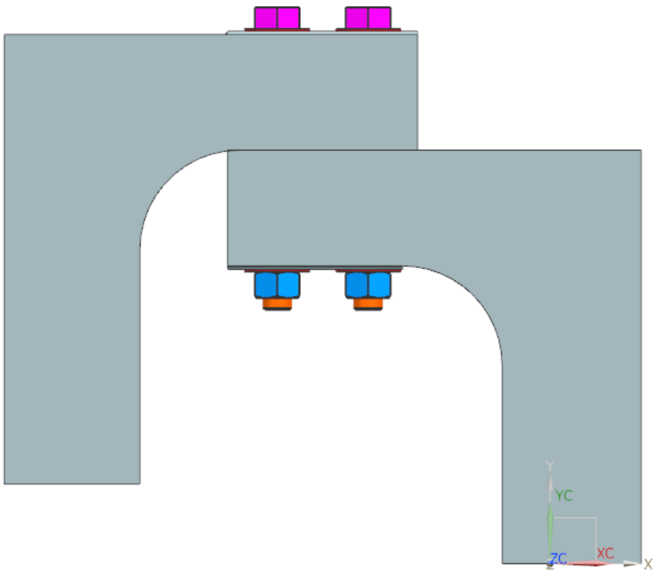

Figure 12 – Simulated setup

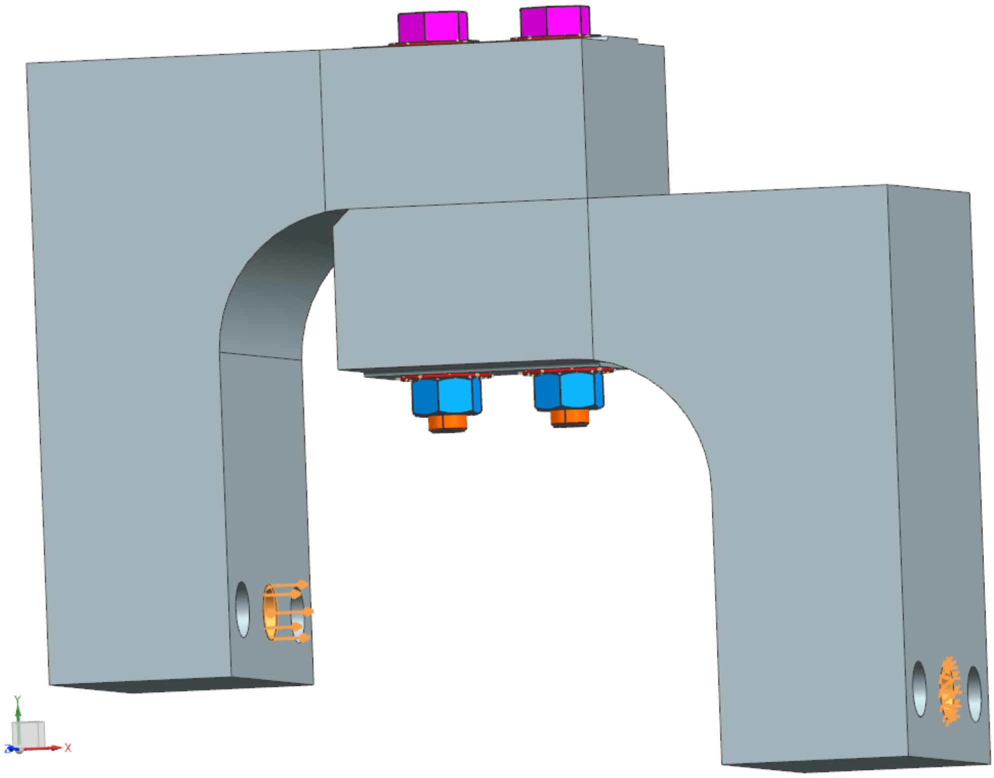

Figure 13 – Simulated set up with force and constraint

The simulated model uses 206,8 GPa as the Elastic Modulus and 0,29 as the Poisson ratio. The threads on the bolts were not simulated (common practice at KvK), instead the pitch diameter was used.

A mesh size of five is used in the whole mid-section of the simulation and a mesh size of forty for the rest (to reduce the computation time). The mesh sizes can be seen underneath in figure 14.

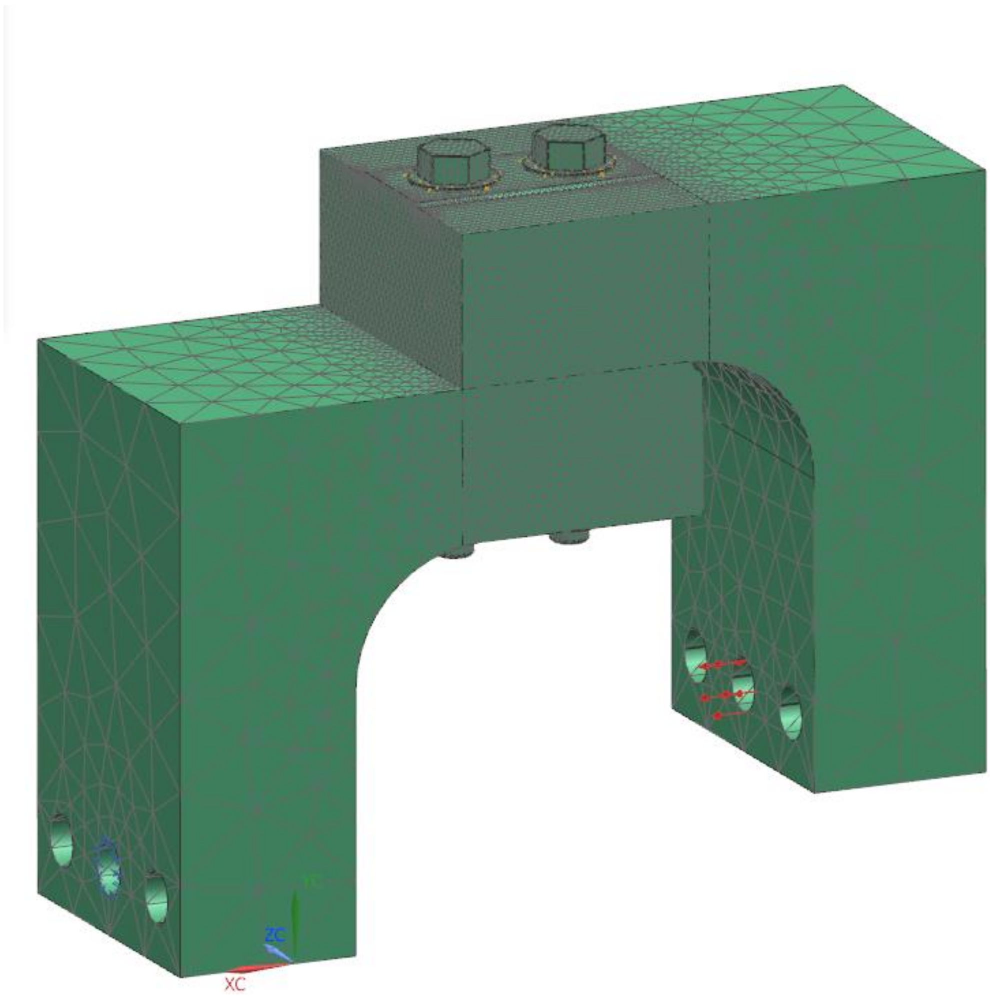

Figure 14 – Meshed simulated set up

This test is more practical minded than more standardized tests, as the bolts experience both shear and bending compared to the more used tensile testing. Validation might therefore be more difficult than if a standardized test was use. The reasoning was that this better imitates the real life working conditions for KvK bolts. The bending this test imitates is generally unwanted and should be minimized via design, but this thesis will not analyze this aspect.

Two simulations are attempted, one with 154 kN preload and one with 178 kN preload. Both tests had a force of 50 kN applied through the hole of the upmost piece. In the physical test, a hydraulic cylinder would provide this. The other piece was locked in place. The stresses were then reviewed and can be seen in chapter 4.

3.2 Physical Testing

The physical testing of the bolts could not be done through the physical test setup at KvK. This was due to it being upgraded and the test centre was waiting on parts from suppliers.

Instead, a full life cycle test was elected for the bolt testing. The testing would not go through an entire life cycle (due to time limitations), but go through roughly 15%. This first period of the life cycle test is, based on experience of the KvK staff, the period with the biggest bolting challenges.

A 7-furrow plough was lined up at KvK Test Centre. When connected, it is turned and exposed to the same forces that it receives during usage. One cycle in the full-scale test is equal to the turning motion and forces in usage. The plough can be shown below in figure 15 (name markings on the plough have been crossed over). As the forces used during this test is a company secret, this thesis can only state that the plough is exposed to dynamical forces equal to those measured in normal plough usage.

Figure 15 - Full plough

During the initial testing, bolts connected to the `Connecting Rod` and the `Rear Frame Support` had big bolt challenges and were chosen for testing. The bolts on the rear frame support goes through the frame and orvel plates on the other side. The bolts are all M20 and can be seen below in figures 16, 17 and 18, with the bolts being marked with a yellow star being the ones chosen:

Figure 16 - Bolts rear frame support back

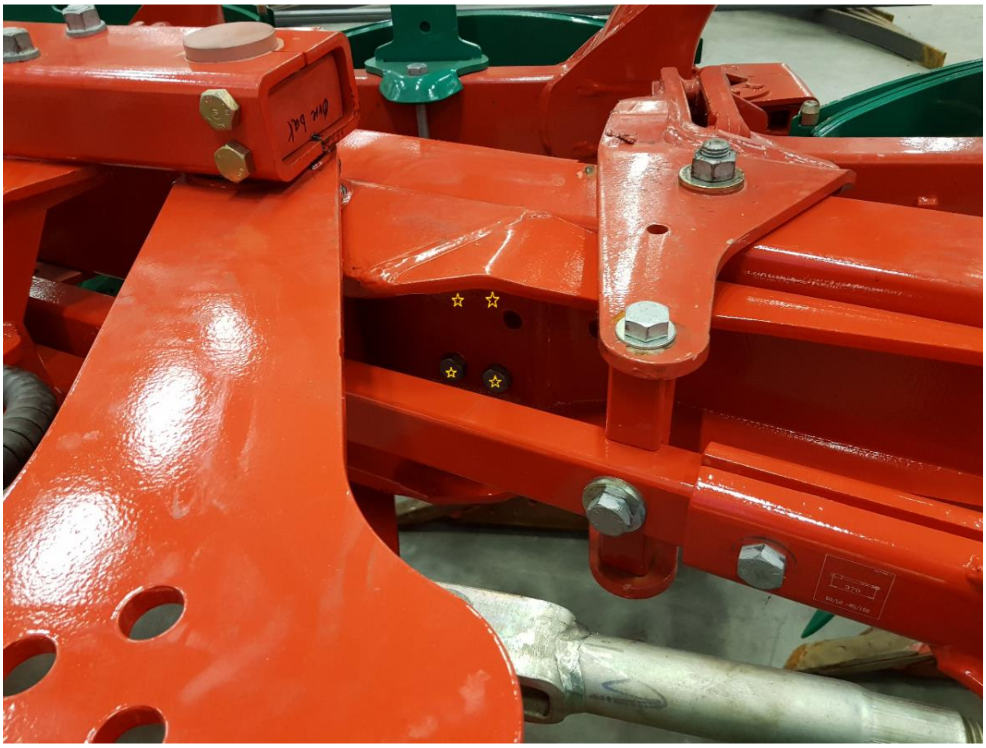

Figure 17 - Bolts rear frame support front

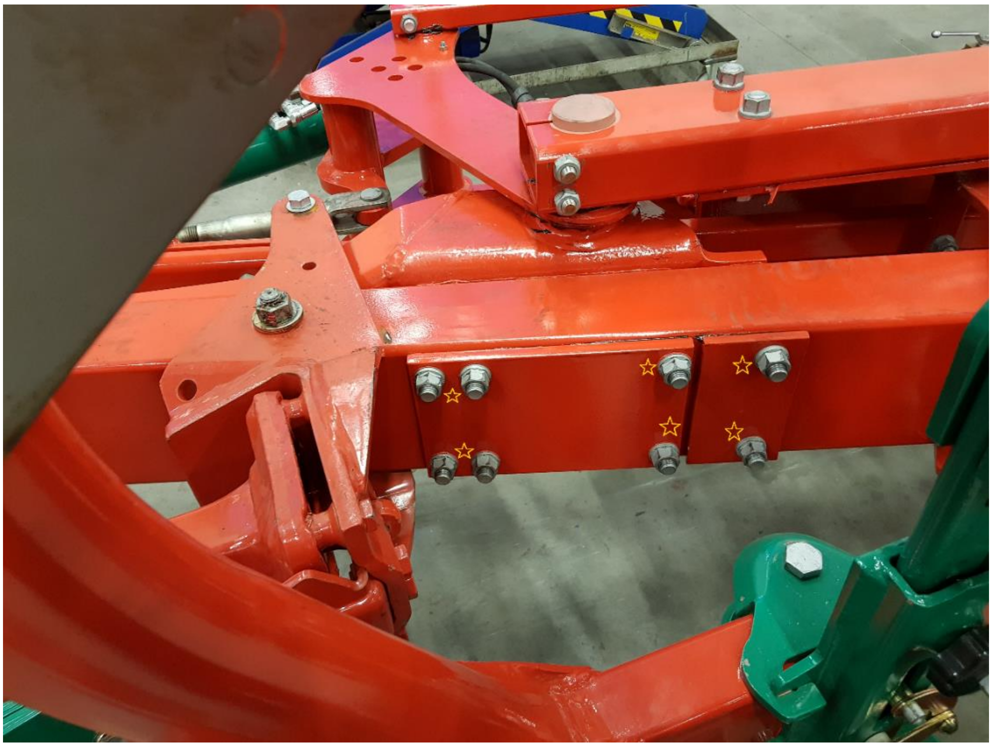

Figure 18 - Nuts rear frame support

There is one Connecting Rod on each side of the plough, which means double the amount of bolts. Through each Connecting Rod there are six bolts going through vertically and four bolts going through horizontally, two on each side. The horizontal bolts are M16 and the other are M20. These bolts are shown in figures 19 and 20:

Figure 19 - Vertical bolts connecting rod

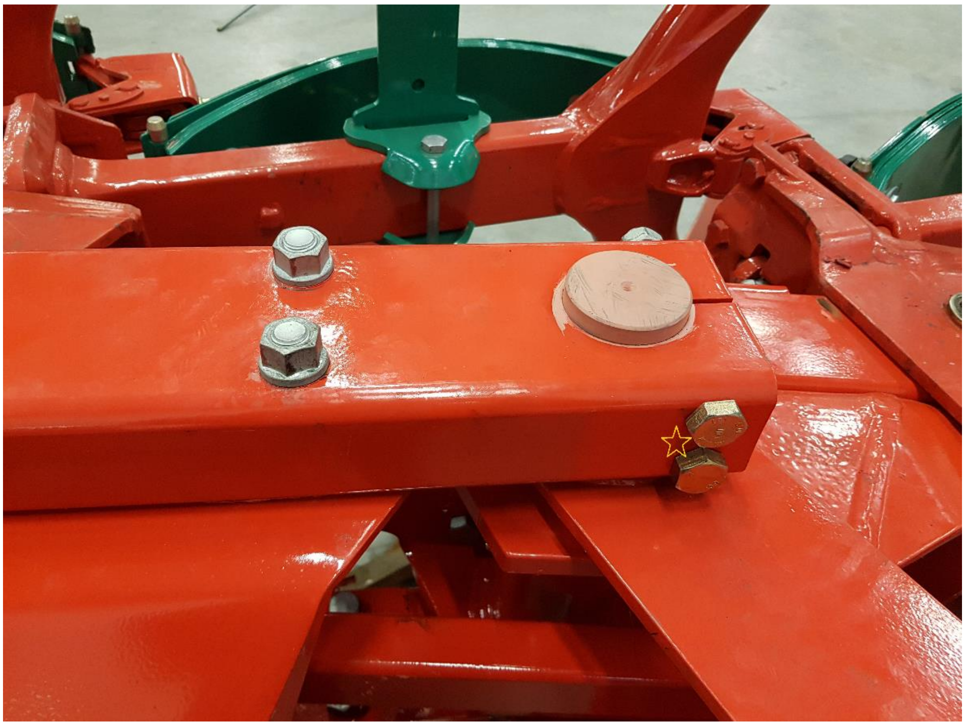

Figure 20 - Horizontal bolts connecting rod

All bolts are of grade 10.9, the original torque for these M16 and M20 bolts at KvK is:

M16: 279,5-304 Nm

M20: 544,5-598,5 Nm

The VDI 2230 proposes the following torques for the same bolts (interpolated between 0,14 & 0,16 friction coefficient):
M16: 354 Nm
M20: 693 Nm

A note should be made here, that the bolts going vertically through the connecting rods originally used standard KvK nuts (not flange nuts). Using the increased torque caused the first two nuts to fail during tightening. When analyzing the KvK nut, as is done in chapter 2, this was not unexpected. The nuts were changed to flange nuts to be able to carry on with the test.

A second note is that a new paint was used for the second test. KvK believes that the new paint will cause less loss of preload for the bolts (as well as other benefits). When examining the effects of different variables, generally only one variable should be changed at a time to see the effect. This was not done here due to the cost of a full-scale life cycle test.

When examining the bolts throughout the test, the same range of "measured loosening" as the original test was used. There were three categories listed as 1, 2 and 3. These are shown below in table 2 with the corresponding action taken.

Category	**Measured**	**Action**
1	Less than 200 Nm lost	The bolts remain as is
2	200-400 Nm lost	The bolts are retightened to original torque
3	Over 400 Nm lost	The bolts are retightened to original torque

Table 2 - Torque measurement categories

Analyzing the equation 1, torque and preload have a linear relationship. It can then be inferred that if torque is retained, then the preload is also retained (barring other influences). The bolts are measured and retightened with a dial wrench. The wrench can be seen in appendix I. The results during the tests were collected and can be seen in chapter 4. The test counts cycles, which is when a plough is turned from ploughing one way to the other. Measurements have been made to simulate the forces that a plough receives in usage and to repeat them during this test. It was decided to test plough up to roughly 5000 cycles, or 15% of the total lifetime, as time limitations at the test centre was at play.

4. Results

In this chapter, the results from the simulation and the physical testing are shown. It ends with a discussion regarding the results and the validity.

4.1 Simulation

Below in figure 21 through 23 the results of the first simulation are shown. Figure 21 shows the stress on the entire set-up and figure 22 shows the same forces, but with half of the setup hidden (as to see the bolts, referred to as sectional view).

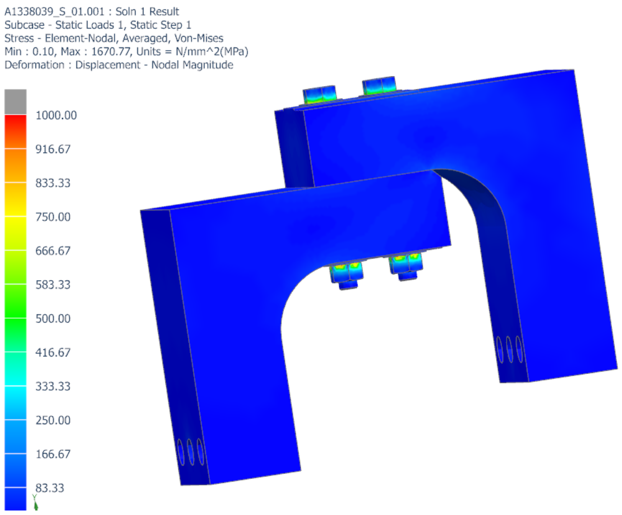

Figure 21 – Full scale simulated model with original preload

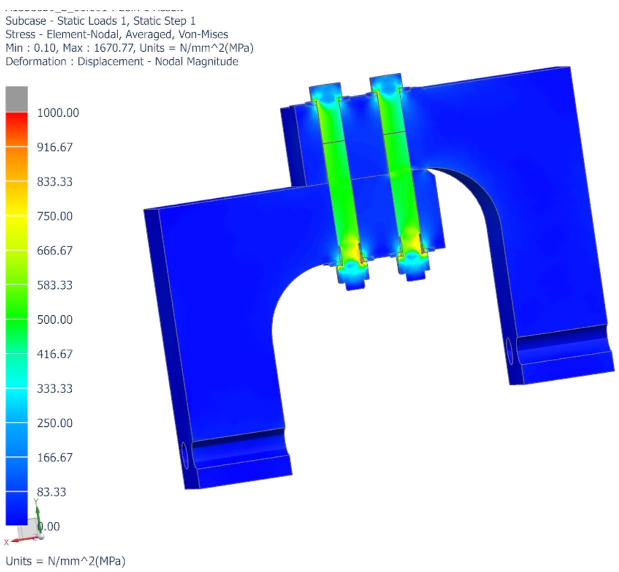

Figure 22 – Sectional view of the simulated model with original preload

Figures 23 shows a close up of only the two bolts. Earlier physical tests at KvK had always ended in failure just above the nut. The stresses in this area were therefore analyzed. Just above the nut a yellow area can be seen, which indicate the maximal stress was about 1200 MPa.

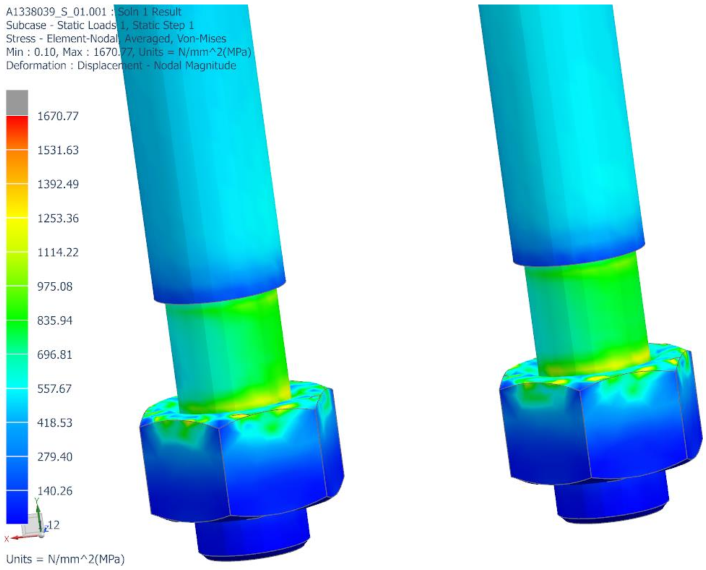

Figure 23 – Stresses over the nut at original preload

An identical test was then gone through, but the bolt preload was changed from 154 kN to 178,5 kN. The figure 24 shows the sectional view of the simulation and figure 25 shows the stresses over the nut with the increased preload.

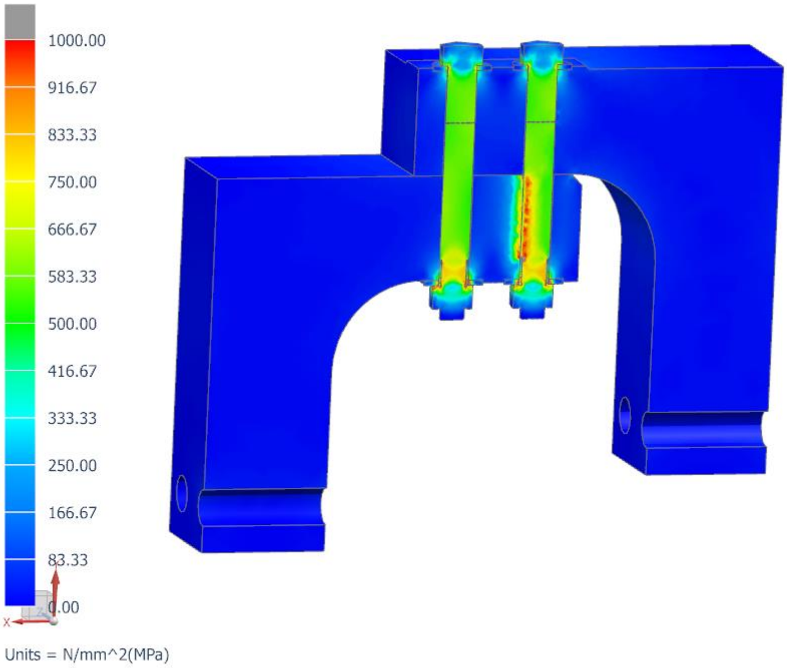

Figure 24 – Sectional view at increased preload

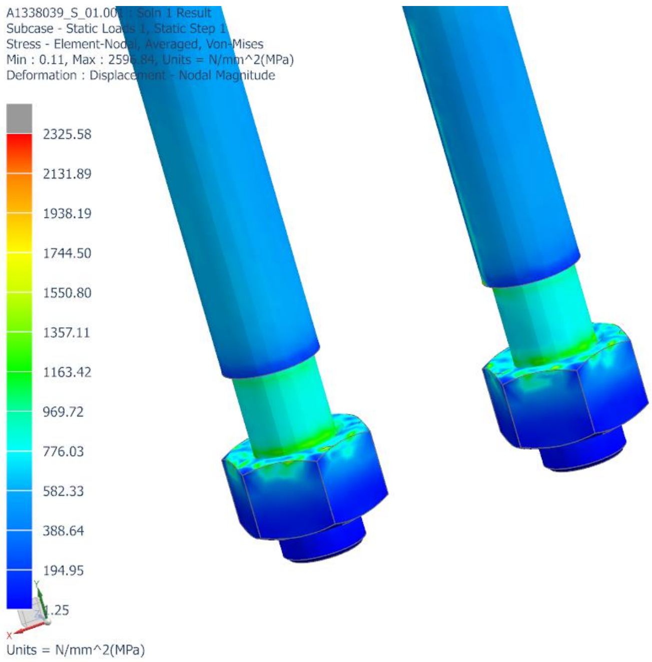

Figure 25 – Stresses just above the nut at increased preload

With the increased preload, the maximal stress did not change significantly. Roughly 1200 MPa is observed just over the nut in both simulations. A new simulation is done with only 50 kN as the bolt preload, to see if any significant changes could be seen. Below, in figure 26, the sectional view with 50 kN preload is shown and in figure 27 the stress just over the nut.

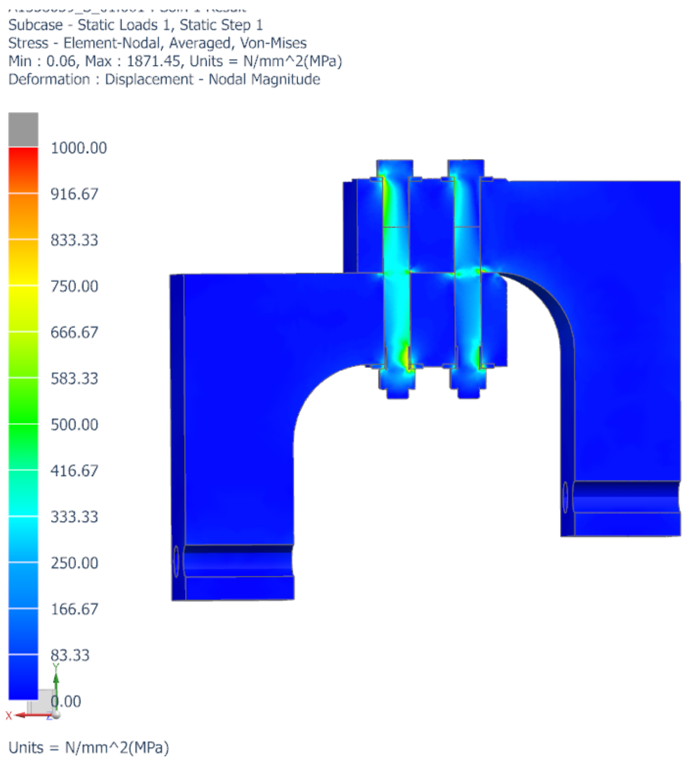

Figure 26 - Sectional view of 50 kN preload simulation

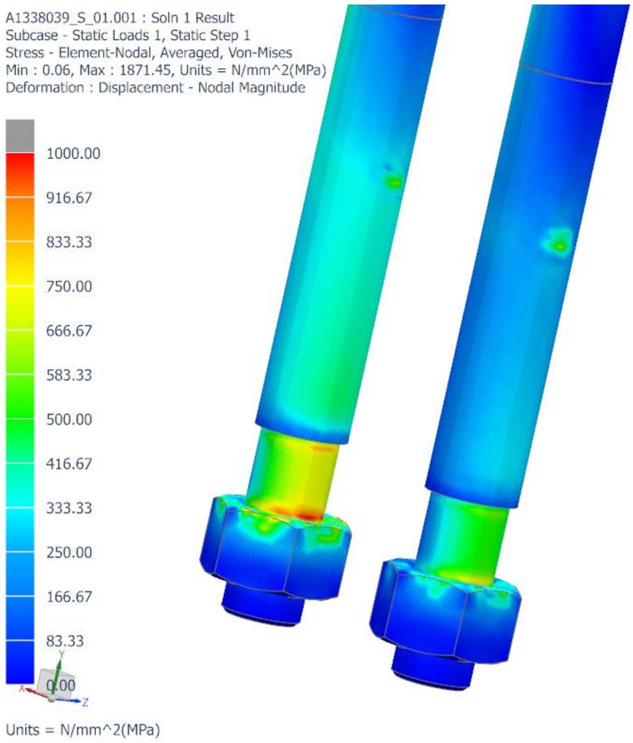

Figure 27 – Stress just above the nut at 50 kN preload

The stress just over the nut was reduced from the roughly 1200 MPa area down to roughly 900 MPa. This shows that the simulation most likely only manage to tell how the bolt preload stresses the bolt and not its reaction to the outside force.

Due to time limitations, this thesis did not try further simulations to emulate the real life effect of bolt preload, as the physical testing will take the rest of the available time.

4.2 Physical Test

Two full-scale tests were carried out to about 15% of the total lifetime of the ploughs. The test parameters were identical, and with the second test having increased preload in the bolts.

Original Preload

During the initial full-scale test, all bolts are checked for loosening. After the end of the test, the connecting rod and rear frame support had especially big loosening challenges. This is shown in table 3 below. Table 3 is a table showing the loss of torque, and through their linear relationship the preload, based on the categories presented in table 2.

Cycles	Connecting Rod	Rear Frame Support
277	Category 2	Category 2
1467	Category 1	Category 2
2024	Category 2	Category 2
3602	Category 3	Category 3
5083	Category 1	Category 3*

Table 3 – Torque losses based on table 2

**Two bolts in the rear frame support was at category 3, the other ones were at category 2.*

This was not the end of the original testing, but the test with increased preload went to 5140 cycles, this is where the comparison stopped. As can be seen from table 3, the bolts on the connecting rod struggle to retain its preload and the bolts on the rear frame support lose its preload with every checkup.

Increased Preload

The same test as before was done, but with all bolts connected to the connecting rod and rear frame support tightened to a higher preload as described in chapter 3. The results when checking the loosening of bolts on the new test with increased preload has been summed up below in table 4.

Cycles	Connecting Rod	Rear Frame Support
313	Category 1	Category 1
1562	Category 2	Category 2
3771	Category 1	Category 1
5140	Category 1	Category 2*

Table 4 – Torque losses bases on Table 2

** The front four bolts were category 2, the rest was category 1.*

After 5140 cycles, the test was terminated. Due to time limitations at the test centre, as other tests were planned. The behavior of the bolts in this test is vastly different. After only one retightening, none of the bolts on the connecting rod loses a significant amount of preload. In addition, the bolts on the rear frame support show a clear improvement in retaining its preload.

4.3 Discussion

The simulated results proved a “dead end” and did not provide validation of the hypothesis of this thesis. The only useful thing to come out of the simulation was the reminder that increasing the preload does increase the mean stress in the bolt. If further simulations were to be done, a study on how to do more dynamical FEM testing should be done as to better simulate the real life situation.

From the physical testing of the increased preload a clear improvement is observed. During the original test, the bolts on the rear frame support could never retain their preload over any amount of time, and the bolts on the connecting rod also struggled over time.

During the new test, the bolt also lost preload (and some loss is always to be expected). These losses were however much smaller and occurred over a much longer timeframe. This would point to the expected theoretical benefits from increasing the preload described in chapter 2 which are also seen during physical testing. The improvement could stem from the new paint being used. This investigation cannot separate these two as they were tested simultaneously.

In terms of validation, the results of the research done in this thesis believe that the increased preload hypothesis is valid. This is due to it being based on a thorough theoretical examination and literature study. In addition, the physical results were greatly improved with the higher preload, showing that the preload does provide resistance to loosening and a higher chance of retaining the preload. The validation did not show the “increased lifetime” part of the hypothesis, only to reduce loosening. These are however so closely linked (loosening reduces preload causing the bolt to experience more external load) that this investigation does not differ between these two

It might be hard for others to replicate the findings of this study through repeating the testing done in this thesis. This is mainly due to KvK being in a competitive industry and its full-scale test parameters are a company secret. The replication might still come through a similar changes being made to similar machinery and exposing these to dynamical loads.

5. Conclusion

In chapter 2, this thesis stated the following hypothesis:

"Increasing the preload at KvK from their current standard, to the VDI 2230 values, will reduce loosening and increase the lifetime of their bolts".

The results were clear, a big improvement was observed from the first to the second full-scale test. Based on the theoretical analysis in chapter 2, the test proposed in chapter 3 and the results presented in chapter 4, this investigation claims that this hypothesis is valid. Increasing the preload does reduce the loosening in bolts and by extension increase the bolt lifetime.

The thesis does have limitations. The improvements could be due to factors such as material qualities, bolt hardness etc. It is hard to fully validate this without more extensive research, but due to the theoretical backing of the test results, this thesis is confident in claiming a clear improvement due to the changes in preload.

As for KvK, the change is not without cost, as most likely KvK will have to remove the reduced wrench size nut (and most likely bolt head) to implement these changes. An upgrade of the assembly tools might also be necessary, but this investigation di not covered this.

5.1 Future Work

This thesis is built on an academic study of the variables at play in a bolted joint. Preload was tested but also the effective friction diameters were proposed as a suitable variable for change. This could provide a big improvement for KvK and could be of interest in general for an increased knowledge on bolted connections. The situation of a high quality bolt with reduced wrench size is very rare and would therefore be a new way of validating knowledge and formulas within the bolt literature.

Several other variables were also mentioned in chapter 2. From an academic perspective, all of the variables mentioned are of interest and can be altered. A suggestion would be a study of the friction coefficients. These greatly affect how the bolted connection work and can be influenced by much. For instance, one could build on the work of (Blom 2013) who studied the effect of tightening speed in combination with friction, with interesting results (big changes between use of 10 and 200 rpm).

To build more directly on the findings of this work, a proposed change could be to look at some of the limitations of this thesis. For instance, examining the bolt supplier and supplied bolts is a big area for further work for KvK. Variations within the bolts themselves can be a big cause for bolt loosening and failure.

Identical or similar testing could also be repeated to increase the validity and statistical significance of the results from this thesis. In fatigue testing, big variances are expected, so more data could be useful to increase the confidence in these results.

6. References

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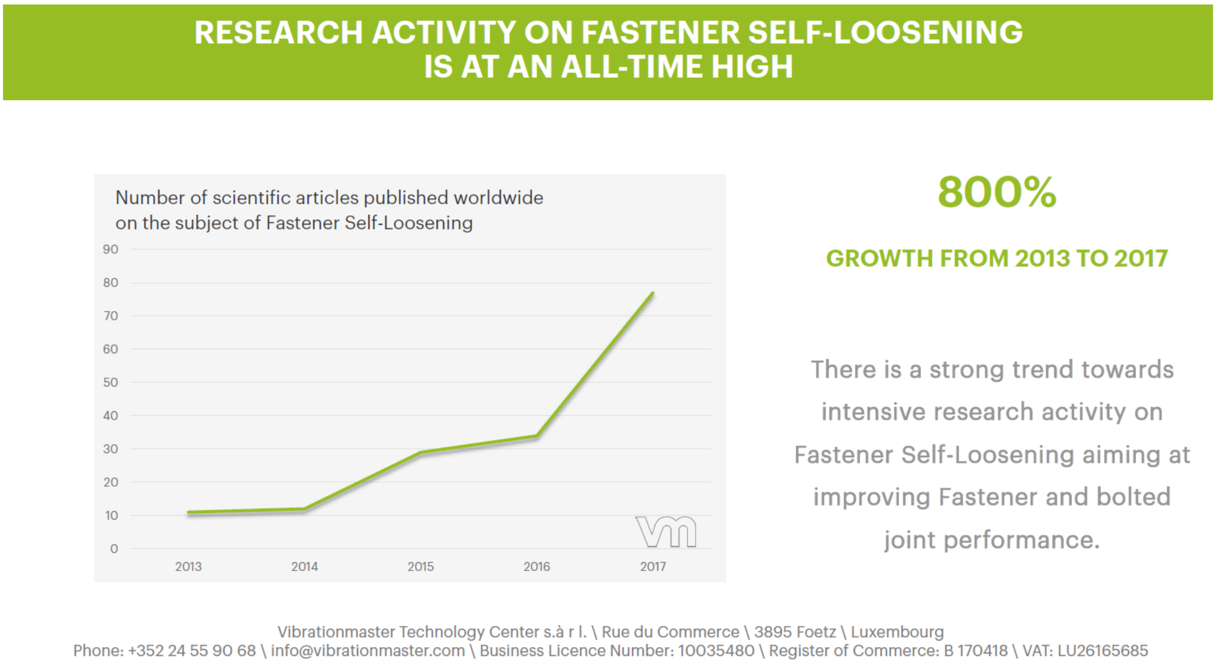
7. Appendix

Appendix A: Vibrationmaster Newsletter

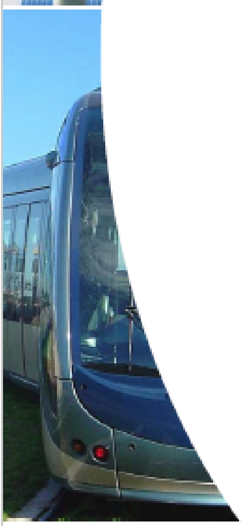
This newsletter was sent out by Vibrationmaster 22.06.2018.



Appendix B: Geomet Coating Datasheets

GEOMET® 500

A unique product for optimal friction and anticorrosion performance



GEOMET® 500 is applied to fasteners and many type of metallic parts to protect from corrosion, and it is used in many industries

- Thin dry-film, non-electrolytic, self-lubricated
- Water-based chemistry
- Passivated zinc and aluminium flakes in a binder, patented chemistry
- Chrome free alternative to DACROMET® 500
- Metallic silver appareance

Characteristics and performance*

- Coefficient of friction: 0,15 ± 0,03 (ISO 16047)
- No topcoat required
- No hydrogen embrittlement
- Excellent assembly and multi-tightening behavior
- Good mechanical damage (test method D24 1312, USCAR 32) and chemical (test VDA 621-412) resistance
- Performance maintained at elevated temperatures (up to 300°C)
- Paintable coating
- Electrical conductivity for most application
- Bimetallic compatibility with aluminum
- Application cost savings

High corrosion resistance*

Coating Weight	Salt Spray Test (ISO 9227 / ASTM B117)	Cyclic Test
Grade A > 24 g/m^2	> 240 hours without white rust > 720 hours without red rust	25 cycles APGE
Grade B > 36 g/m^2	> 240 hours without white rust > 1000 hours without red rust	6 cycles ACT 50 cycles APGE

* Results may vary depending on substrate, geometry of parts and type of application processes

NOF METAL COATINGS GROUP

GEOMET® 321

The world's choice for corrosion protection

GEOMET® 321 is applied to protect fasteners and many type of metallic parts from corrosion and is used in many industries. It can be combined with PLUS®, DACROLUB® or GEOKOTE® topcoats to provide a very broad range of friction coefficients. It is the most widely used product in zinc flake technology.

- Thin dry-film, non-electrolytic
- Water-based chemistry
- Passivated zinc and aluminium flakes in a binder, patented chemistry
- Chrome free alternative to DACROMET® 320
- Metallic silver appearance

Characterictics and performance*

- The coefficient of friction can be adjusted to targeted values ranging from 0.06 to 0.20 (ISO 16047) with NOF METAL COATING GROUP's selected topcoats
- It can be used with or without topcoat
- No hydrogen embrittlement
- Excellent assembly and multi-tightening behavior (with lubricated topcoat)
- Good mechanical damage (test method D24 1312, USCAR 32) and chemical (test VDA 621-412) resistance
- Performance maintained at elevated temperatures (up to 300°C)
- Paintable coating
- Electrical conductivity for most application processes
- Bimetallic compatibility with aluminum

High corrosion resistance*

Coating Weight	Salt Spray Test (ISO 9227/ASTM B117)	Cyclic Test
> 24 g/m^2	> 240 hours without white rust > 720 hours without red rust	
> 24 g/m^2 + topcoat	> 720 hours without red rust	4 cycles ACT 60 cycles GMW 14872 60 cycles SAE J2334
> 36 g/m^2	> 1000 hours without red rust	

* Results may vary depending on substrate, geometry of parts and type of application processes

NOF METAL COATINGS GROUP

PLUS® XL, L, VL, ML, M, 10

Worldwide trusted and proven performance

The PLUS® brand of topcoats are lubricated sealers. When applied to DACROMET® or GEOMET® base-coats, they provide controlled friction coefficient of threaded parts and increase the resistance to corrosion. The controlled sacrificial corrosion protection mechanism of DACROMET® and GEOMET® is improved further with the PLUS® topcoats.

Characteristics and performance*

- Friction coefficient on GEOMET® 321 base-coat (ISO 16047)

PLUS® XL	:	0.06 – 0.09
PLUS® L	:	0.08 – 0.14
PLUS® VL	:	0.09 – 0.14
PLUS® ML	:	0.10 – 0.16
PLUS® M	:	0.12 – 0.18
PLUS®	:	0.14 – 0.20
PLUS® 10	:	not lubricated

- Appearance: matte metallic silver (clear topcoats)
- Salt Spray Test according to ISO 9227 / ASTM B117

  GEOMET® 321/500 + PLUS® XL, L, VL, M, ML, 10 > 720 hours without red rust

  (with a GEOMET® 321/500 coating weight > 24 g/m²)

- Increased field performance of parts
- Improved contact corrosion resistance (to magnesium, rubber, aluminum, etc)
- Excellent resistance to solvents, fuels and brake fluids (VDA 621 412)
- Competitive production cost

Application process

These water-based inorganic products can be applied by Dip-Spin, Spray, Dip-Drain-Spin, using bulk or rack

* Results may vary depending on substrate, geometry of parts and type of application processes

NOF METAL COATINGS GROUP

Appendix C: Internal KvK Torque Tables

Kverneland Standard	Tiltrekningsmoment for metriske grovgjenger	KN 47.02 Side : 1 av : 1 Utgitt : Sep. -78 Revidert : Mai -84 Sign. : AS

Denne standard er utarbeidet som et hjelpemiddel
til bruk på KN.

Skrue diam.	Tiltrekkingsmoment KPm						
	Svarte skruer (oljet)			El.forsinkede skruer (tørre)			
	8,8	10,9	12,9	4,6	8,8	10,9	12,9
M8			4,5- 5,5				
M10	5,5- 6,0	7,5- 8,0	9,0-10,0	1,5- 2,0	4,5- 5,0	6,5- 7,0	8,0- 9,0
M12	9,0-10,0	13,0-14,0	15,5-17,0	3,0- 3,5	8,5- 9,0	11,5-12,5	14,0-15,0
M14	14,5-16,0	20,5-22,5	25,0-27,0	5,0- 5,5	13,0-14,5	18,5-20,0	22,0-24,0
M16	23,0-25,0	32,0-35,0	38,0-42,0	7,5- 8,5	20,0-22,0	28,5-31,0	34,0-37,5
M20	44,0-48,5	62,0-68,5	74,5-82,0	14,5-16,0	39,0-43,0	55,5-61,0	65,5-73,0
M24	76,5-84,0	107,0-118,0	129,5-142,5	25,5-28,0	68,0-75,0	95,0-105,0	115,5-127,0

1KPm ≈ 9,81 Nm

Skrue diam.	Tiltrekkingsmoment (Nm)						
	Svarte skruer (oljet)			El.forsinkede skruer (tørre)			
	8,8	10,9	12,9	4,6	8,8	10,9	12,9
M8			44-54				
M10	54,0- 59.0	73.5- 78,5	88,0- 98.0	15,0- 19,5	44,0- 49,0	64,0- 69,0	78,5- 88,0
M12	88,0- 98,0	127,5-137,0	152,0-167,0	29,0- 34,0	83,0- 88,0	113,0-122,5	137,0-147,0
M14	142,0-157,0	201,0-221,0	245,0-265,0	49,0- 54,0	127,5-142,0	181,5-196,0	216,0-235,5
M16	225,5-245,0	314,0-343,0	373,0-412,0	73,5- 83,0	196,0-216,0	279,5-304,0	333,5-368,0
M20	431,5-476,0	608,0-672,0	731,0-804,0	142,0-157,0	382,5-422,0	544,5-598,5	652,0-716,0
M24	750,5-824,0	1049,5-1157,5	1270,0-1398,0	250,0-274,5	667,0-736,0	932,0-1030,0	1133,0-1246,0

1 Nm ≈ 0,102 KPm

Appendix D: Tightening factors

Preload scatter using different tightening methods – Standard VDI 2230 A8

Table A8. Guide values for the tightening factor α_A

<table>
<tr><th>Tightening factor α_A</th><th>Scatter $\frac{\Delta F_M}{2 \cdot F_{Mm}} = \frac{\alpha_A - 1}{\alpha_A + 1}$</th><th>Tightening technique</th><th>Adjusting technique</th><th colspan="2">Remarks</th></tr>
<tr><td>1,05 to 1,2</td><td>±2 % to ±10 %</td><td>Elongation-controlled tightening with ultrasound</td><td>Echo time</td><td colspan="2">• Calibrating values necessary
• Allow for progressive increase in errors at $l_K/d < 2$
• Smaller errors with direct mechanical coupling, larger with indirect coupling</td></tr>
<tr><td>1,1 to 1,5</td><td>±5 % to ±20 %</td><td>Mechanical elongation measurement</td><td>Adjustment via longitudinal measurement</td><td colspan="2">• Exact determination of the axial elastic resilience of the bolt is necessary. The scatter depends substantially on the accuracy of the measuring technique.
• Allow for progressive increase in errors at $l_K/d < 2$</td></tr>
<tr><td>1,2 to 1,4</td><td>±9 % to ±17 %</td><td>Yield-controlled tightening, motor or manually operated</td><td>Input of the relative torque/rotation-angle coefficient</td><td colspan="2" rowspan="2">The scatter in preload is substantially determined by the scatter in the bolt yield point. Here, the bolts are dimensioned for $F_{M\,min}$; a design of the bolts for $F_{M\,max}$ with the tightening factor α_A therefore does not apply to these tightening techniques.</td></tr>
<tr><td>1,2 to 1,4</td><td>±9 % to ±17 %</td><td>Angle-controlled tightening, motor or manually operated</td><td>Experimental determination of pre-tightening torque and angle of rotation (steps)</td></tr>
<tr><td>1,2 to 1,6</td><td>±9 % to ±23 %</td><td>Hydraulic tightening</td><td>Adjustment via length or pressure measurement</td><td colspan="2">• Lower values for long bolts ($l_K/d \geq 5$)
• Higher values for short bolts ($l_K/d \leq 2$)</td></tr>
<tr><td>1,4 to 1,6</td><td>±17 % to ±23 %</td><td>Torque-controlled tightening with torque wrench, indicating wrench, or precision tightening spindle with dynamic torque measurement</td><td>Experimental determination of required tightening torques on the original bolting part, e.g. by measuring bolt elongation</td><td>Lower values:
large number of calibration or check tests (e.g. 20) required; low scatter of the transmitted torque (e.g. ±5 %) necessary</td><td>Lower values for:
• small angles of rotation, i.e. relatively stiff joints
• relatively soft mating surface[1])
• mating surfaces which are not inclined to "seize", e.g. phosphated or with sufficient lubrication</td></tr>
<tr><td>1,6 to 2,0
(friction coefficient class B)

1,7 to 2,5
(friction coefficient class A)</td><td>±23 % to ±33 %

±26 % to ±43 %</td><td>Torque-controlled tightening with torque wrench, indicating wrench, or precision tightening spindle with dynamic torque measurement</td><td>Determination of the required tightening torque by estimating the friction coefficient (surface and lubricating conditions)</td><td>Lower values for:
Measuring torque wrenches with steady tightening and for precision tightening spindles

Higher values for:
Signaling or automatic tripping torque wrenches</td><td>Higher values for:
• large angles of rotation, i.e. relatively resilient joints and fine threads
• high mating surface hardness combined with a rough surface</td></tr>
<tr><td>2,5 to 4</td><td>±43 % to ±60 %</td><td>Tightening with impact wrench or impact wrench with momentum control</td><td>Calibration of the bolt by means of re-tightening torque, made up of the required tightening torque (for the estimated friction coefficient) and an additional factor</td><td colspan="2">Lower values for:
• large number of calibration tests (re-tightening torque)
• on the horizontal segment of the bolt characteristic
• momentum transfer free from play</td></tr>
</table>

[1]) Mating surface: Clamped unit its surface contact the tightening unit of the joint (bolt head or nut).

Appendix E: Data for standard Metric bolt dimensions

Standard VDI 2230 A12

Table A12. Nominal values for pitch, minor diameter, reduced-shank diameter, reduced-shank cross section and load $F_{0,2\,min}$ **for necked-down bolts** with metric standard and fine threads (pitch and minor diameter according to DIN 13-1, -5 to -8; minimum yield point according to DIN EN ISO 898-1)

Abmessung Size	**Steigung Pitch**	**Kerndurchmesser Minor diameter**	**Taillendurchmesser Reduced-shank diameter**	**Taillenquerschnitt Reduced-shank cross section**	**Kraft an der Mindest-Streckgrenze Load at the minimum yield point** $F_{0,2min} = R_{p0,2min} \cdot \frac{\pi}{4}(0,9 \cdot d_3)^2$ Festigkeitsklasse/Strength grade		
	P	d_3	$d_T = 0,9 \cdot d_3$	$A_T = \frac{\pi}{4}(0,9 \cdot d_3)^2$	8.8	10.9	12.9
	mm	mm	mm	mm²	N	N	N
Metrisches Regelgewinde/Metric standard thread							
M 4	0,7	3,141	2,83	6,28	4 000	5 900	6 900
M 5	0,8	4,019	3,62	10,3	6 600	9 700	11 300
M 6	1	4,773	4,30	14,5	9 300	13 600	15 900
M 7	1	5,773	5,20	21,2	13 600	19 900	23 300
M 8	1,25	6,466	5,82	26,6	17 000	25 000	29 500
M 10	1,5	8,160	7,34	42,4	27 000	40 000	46 500
M 12	1,75	9,853	8,87	61,8	39 500	58 000	68 000
M 14	2	11,546	10,4	84,8	54 000	80 000	93 000
M 16	2	13,546	12,2	117	75 000	110 000	128 000
M 18	2,5	14,933	13,4	142	94 000	133 000	156 000
M 20	2,5	16,933	15,2	182	120 000	171 000	201 000
M 22	2,5	18,933	17,0	228	151 000	214 000	250 000
M 24	3	20,319	18,3	263	173 000	247 000	290 000
M 27	3	23,319	21,0	346	228 000	325 000	380 000
M 30	3,5	25,706	23,1	420	275 000	395 000	460 000
M 33	3,5	28,706	25,8	524	345 000	495 000	580 000
M 36	4	31,093	28,0	615	405 000	580 000	680 000
M 39	4	34,093	30,7	739	490 000	700 000	810 000
Metrisches Feingewinde/Metric fine thread							
M 8	1	6,773	6,10	29,2	18 700	27 500	32 000
M 9	1	7,773	7,00	38,4	24 600	36 000	42 500
M 10	1	8,773	7,90	49,0	31 500	46 000	54 000
M 10	1,25	8,466	7,62	45,6	29 000	43 000	50 000
M 12	1,25	10,466	9,42	69,7	44 500	66 000	77 000
M 12	1,5	10,160	9,14	65,7	42 000	62 000	72 000
M 14	1,5	12,160	10,94	94,1	60 000	88 000	103 000
M 16	1,5	14,160	12,74	128	82 000	120 000	140 000
M 18	1,5	16,160	14,54	166	110 000	156 000	183 000
M 18	2	15,546	13,99	154	101 000	145 000	169 000
M 20	1,5	18,160	16,34	210	138 000	197 000	231 000
M 22	1,5	20,160	18,14	259	171 000	243 000	285 000
M 24	1,5	22,160	19,94	312	206 000	295 000	345 000
M 24	2	21,546	19,39	295	195 000	280 000	325 000
M 27	1,5	25,160	22,64	403	265 000	380 000	445 000
M 27	2	24,546	22,09	383	255 000	360 000	420 000
M 30	1,5	28,160	25,34	504	335 000	475 000	550 000
M 30	2	27,546	24,79	483	320 000	455 000	530 000
M 33	1,5	31,160	28,04	618	410 000	580 000	680 000
M 33	2	30,546	27,49	594	390 000	560 000	650 000
M 36	2	33,546	30,19	716	470 000	670 000	780 000
M 36	3	32,319	29,09	664	440 000	620 000	730 000
M 39	2	36,546	32,89	850	561 000	799 000	935 000
M 39	3	35,319	31,79	794	520 000	750 000	870 000

Appendix F: Friction coefficient tables.

Standard VDI 2230 A5

Table A5. Friction coefficient classes with guide values for different materials/surfaces and lubrication states in bolted joints

Friction coefficient class	Range for μ_G and μ_K	Selection of typical examples for	
		Material/surfaces	**Lubricants**
A	0,04 to 0,10	metallically bright black oxide phosphated galvanic coatings such as Zn, Zn/Fe, Zn/Ni Zinc laminated coatings	solid lubricants, such as MoS_2, graphite, PTFE, PA, PE, PI in lubricating varnishes, as top coats or in pastes; liquefied wax wax dispersions
B	0,08 to 0,16	metallically bright black oxide phosphated galvanic coatings such as Zn, Zn/Fe, Zn/Ni Zinc laminated coatings Al and Mg alloys	solid lubricants, such as MoS_2, graphite, PTFE, PA, PE, PI in lubricating varnishes, as top coats or in pastes; liquefied wax; wax dispersions, greases; oils; delivery state
		hot-galvanized	MoS_2; graphite; wax dispersions
		organic coatings	with integrated solid lubricant or wax dispersion
		austenitic steel	solid lubricants or waxes; pastes
C	0,14 to 0,24	austenitic steel	wax dispersions, pastes
		metallically bright phosphated	delivery state (lightly oiled)
		galvanic coatings such as Zn, Zn/Fe, Zn/Ni Zinc laminated coatings adhesive	none
D	0,20 to 0,35	austenitic steel	oil
		galvanic coatings such as Zn, Zn/Fe; hot-galvanized	none
E	≥ 0,30	galvanic coatings such as Zn/Fe, Zn/Ni austenitic steel Al, Mg alloys	none

The aim is to **achieve** coefficients of friction which fit into the **friction coefficient class B** in order to apply as high a preload as possible with low scatter. This does not automatically mean using the smallest values and that the friction coefficient scatter present corresponds to the class spread. The tables apply at room temperature.

Appendix G: Preload & Torque Table for Metric Shank Bolts

Standard VDI 2230 A1

Anhang A Tabellen zur Berechnung

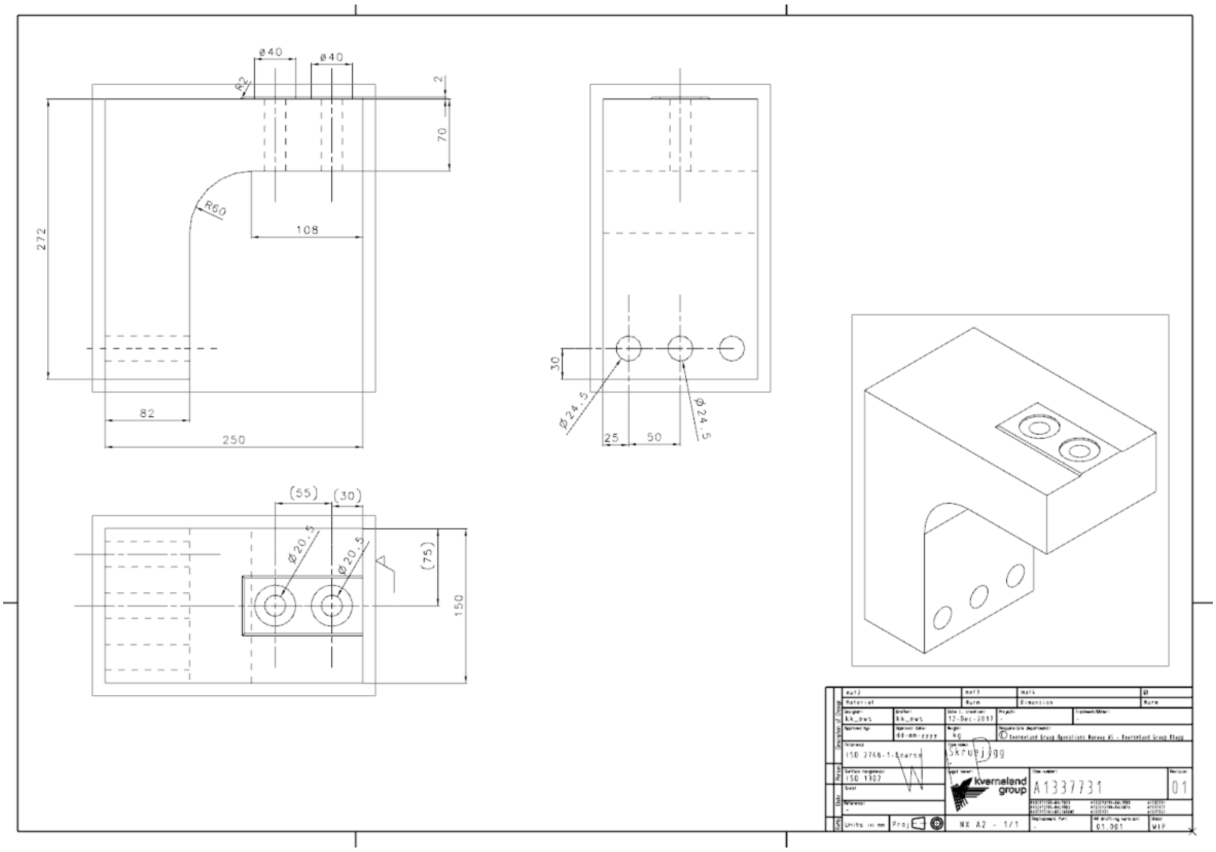
Tabelle A1. Montagevorspannkräfte $F_{M\,Tab}$ und Anziehdrehmomente M_A bei ν = 0,9 für **Schaftschrauben** mit metrischem Regelgewinde nach DIN ISO 262; Kopfabmessungen von Sechskantschrauben nach DIN EN ISO 4014 bis 4018, Schrauben mit Außensechsrund nach DIN 34 800 bzw. Zylinderschrauben nach DIN EN ISO 4762 und Bohrung „mittel" nach DIN EN 20 273

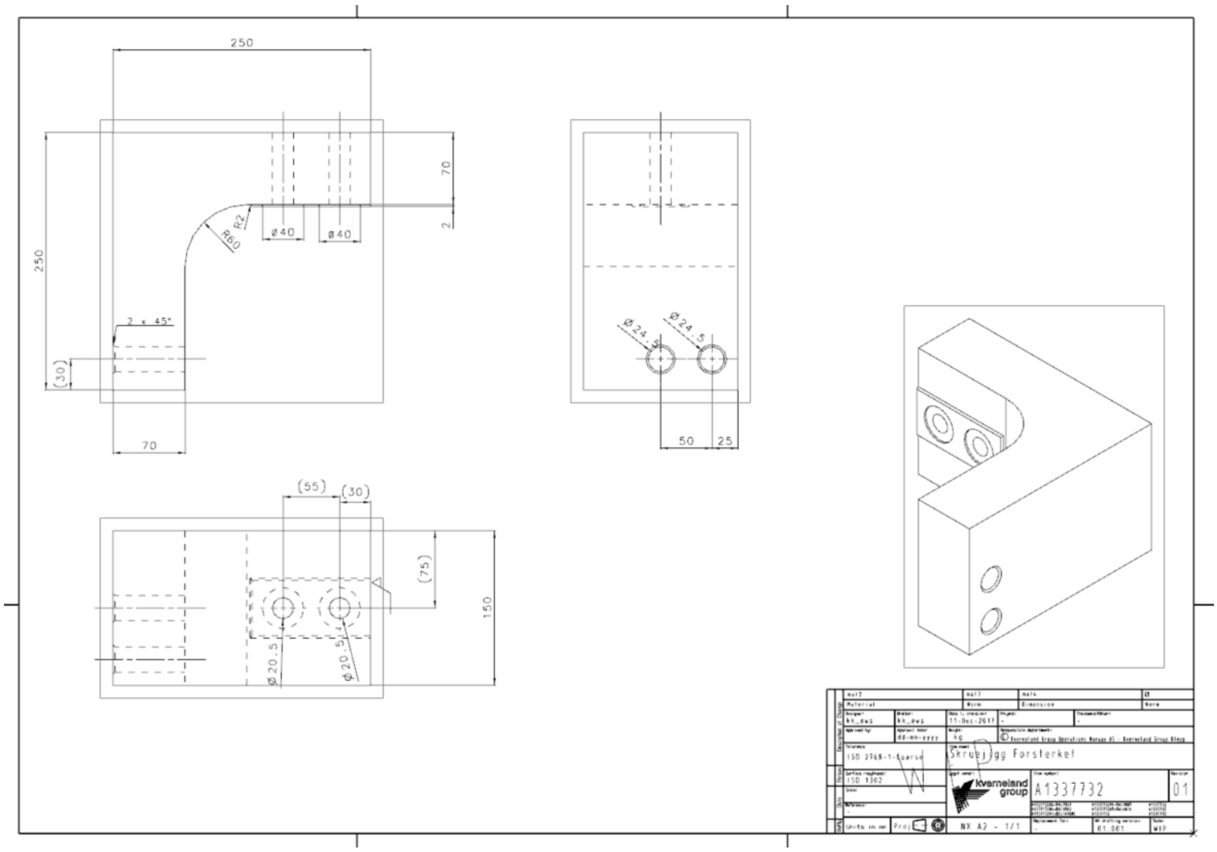
Annex A Calculation tables

Table A1. Assembly preload $F_{M\,Tab}$ and tightening torque M_A with ν = 0,9 for **shank bolts** with metric standard thread according to DIN ISO 262; head dimensions of hexagonal bolts according to DIN EN ISO 4014 to 4018, hexalobular external driving head bolts according to DIN 34 800 or cylindrical bolts according to DIN EN ISO 4762 and hole "medium" according to DIN EN 20 273

Abm. Size	Fest.-Klasse Strength Grade	Montagevorspannkräfte/Assembly preload $F_{M\,Tab}$ in kN für μ_G =							Anziehdrehmomente/Tightening torque M_A in Nm für $\mu_K = \mu_G$ =						
		0,08	0,10	0,12	0,14	0,16	0,20	0,24	0,08	0,10	0,12	0,14	0,16	0,20	0,24
M 4	8.8	4,6	4,5	4,4	4,3	4,2	3,9	3,7	2,3	2,6	3,0	3,3	3,6	4,1	4,5
	10.9	6,8	6,7	6,5	6,3	6,1	5,7	5,4	3,3	3,9	4,6	4,8	5,3	6,0	6,6
	12.9	8,0	7,8	7,6	7,4	7,1	6,7	6,3	3,9	4,5	5,1	5,6	6,2	7,0	7,8
M 5	8.8	7,6	7,4	7,2	7,0	6,8	6,4	6,0	4,4	5,2	5,9	6,5	7,1	8,1	9,0
	10.9	11,1	10,8	10,6	10,3	10,0	9,4	8,8	6,5	7,6	8,6	9,5	10,4	11,9	13,2
	12.9	13,0	12,7	12,4	12,0	11,7	11,0	10,3	7,6	8,9	10,0	11,2	12,2	14,0	15,5
M 6	8.8	10,7	10,4	10,2	9,9	9,6	9,0	8,4	7,7	9,0	10,1	11,3	12,3	14,1	15,6
	10.9	15,7	15,3	14,9	14,5	14,1	13,2	12,4	11,3	13,2	14,9	16,5	18,0	20,7	22,9
	12.9	18,4	17,9	17,5	17,0	16,5	15,5	14,5	13,2	15,4	17,4	19,3	21,1	24,2	26,8
M 7	8.8	15,5	15,1	14,8	14,4	14,0	13,1	12,3	12,6	14,8	16,8	18,7	20,5	23,6	26,2
	10.9	22,7	22,5	21,7	21,1	20,5	19,3	18,1	18,5	21,7	24,7	27,5	30,1	34,7	38,5
	12.9	26,6	26,0	25,4	24,7	24,0	22,6	21,2	21,6	25,4	28,9	32,2	35,2	40,6	45,1
M 8	8.8	19,5	19,1	18,6	18,1	17,6	16,5	15,5	18,5	21,6	24,6	27,3	29,8	34,3	38,0
	10.9	28,7	28,0	27,3	26,6	25,8	24,3	22,7	27,2	31,8	36,1	40,1	43,8	50,3	55,8
	12.9	33,6	32,8	32,0	31,1	30,2	28,4	26,6	31,8	37,2	42,2	46,9	51,2	58,9	65,3
M 10	8.8	31,0	30,3	29,6	28,8	27,9	26,3	24,7	36	43	48	54	59	68	75
	10.9	45,6	44,5	43,4	42,2	41,0	38,6	36,2	53	63	71	79	87	100	110
	12.9	53,3	52,1	50,8	49,4	48,0	45,2	42,4	62	73	83	93	101	116	129
M 12	8.8	45,2	44,1	43,0	41,9	40,7	38,3	35,9	63	73	84	93	102	117	130
	10.9	66,3	64,8	63,2	61,5	59,8	56,3	52,8	92	108	123	137	149	172	191
	12.9	77,6	75,9	74,0	72,0	70,0	65,8	61,8	108	126	144	160	175	201	223
M 14	8.8	62,0	60,6	59,1	57,5	55,9	52,6	49,3	100	117	133	148	162	187	207
	10.9	91,0	88,9	86,7	84,4	82,1	77,2	72,5	146	172	195	218	238	274	304
	12.9	106,5	104,1	101,5	98,8	96,0	90,4	84,8	171	201	229	255	279	321	356
M 16	8.8	84,7	82,9	80,9	78,8	76,6	72,2	67,8	153	180	206	230	252	291	325
	10.9	124,4	121,7	118,8	115,7	112,6	106,1	99,6	224	264	302	338	370	428	477
	12.9	145,5	142,4	139,0	135,4	131,7	124,1	116,6	262	309	354	395	433	501	558
M 18	8.8	107	104	102	99	96	91	85	220	259	295	329	360	415	462
	10.9	152	149	145	141	137	129	121	314	369	421	469	513	592	657
	12.9	178	174	170	165	160	151	142	367	432	492	549	601	692	769
M 20	8.8	136	134	130	127	123	116	109	308	363	415	464	509	588	655
	10.9	194	190	186	181	176	166	156	438	517	592	661	725	838	933
	12.9	227	223	217	212	206	194	182	513	605	692	773	848	980	1092
M 22	8.8	170	166	162	158	154	145	137	417	495	567	634	697	808	901
	10.9	242	237	231	225	219	207	194	595	704	807	904	993	1151	1284
	12.9	283	277	271	264	257	242	228	696	824	945	1057	1162	1347	1502
M 24	8.8	196	192	188	183	178	168	157	529	625	714	798	875	1011	1126
	10.9	280	274	267	260	253	239	224	754	890	1017	1136	1246	1440	1604
	12.9	327	320	313	305	296	279	262	882	1041	1190	1329	1458	1685	1877
M 27	8.8	257	252	246	240	234	220	207	772	915	1050	1176	1292	1498	1672
	10.9	367	359	351	342	333	314	295	1100	1304	1496	1674	1840	2134	2381
	12.9	429	420	410	400	389	367	345	1287	1526	1750	1959	2153	2497	2787
M 30	8.8	313	307	300	292	284	268	252	1053	1246	1428	1597	1754	2931	2265
	10.9	446	437	427	416	405	382	359	1500	1775	2033	2274	2498	2893	3226
	12.9	522	511	499	487	474	447	420	1755	2077	2380	2662	2923	3386	3775
M 33	8.8	389	381	373	363	354	334	314	1415	1679	1928	2161	2377	2759	3081
	10.9	554	543	531	517	504	475	447	2015	2392	2747	3078	3385	3930	4388
	12.9	649	635	621	605	589	556	523	2358	2799	3214	3601	3961	4598	5135
M 36	8.8	458	448	438	427	415	392	368	1825	2164	2482	2778	3054	3541	3951
	10.9	652	638	623	608	591	558	524	2600	3082	3535	3957	4349	5043	5627
	12.9	763	747	729	711	692	653	614	3042	3607	4136	4631	5089	5902	6585
M 39	8.8	548	537	525	512	498	470	443	2348	2791	3208	3597	3958	4598	5137
	10.9	781	765	748	729	710	670	630	3345	3975	4569	5123	5637	6549	7317
	12.9	914	895	875	853	831	784	738	3914	4652	5346	5994	6596	7664	8562

Appendix H: Technical Drawings of bolt test setup

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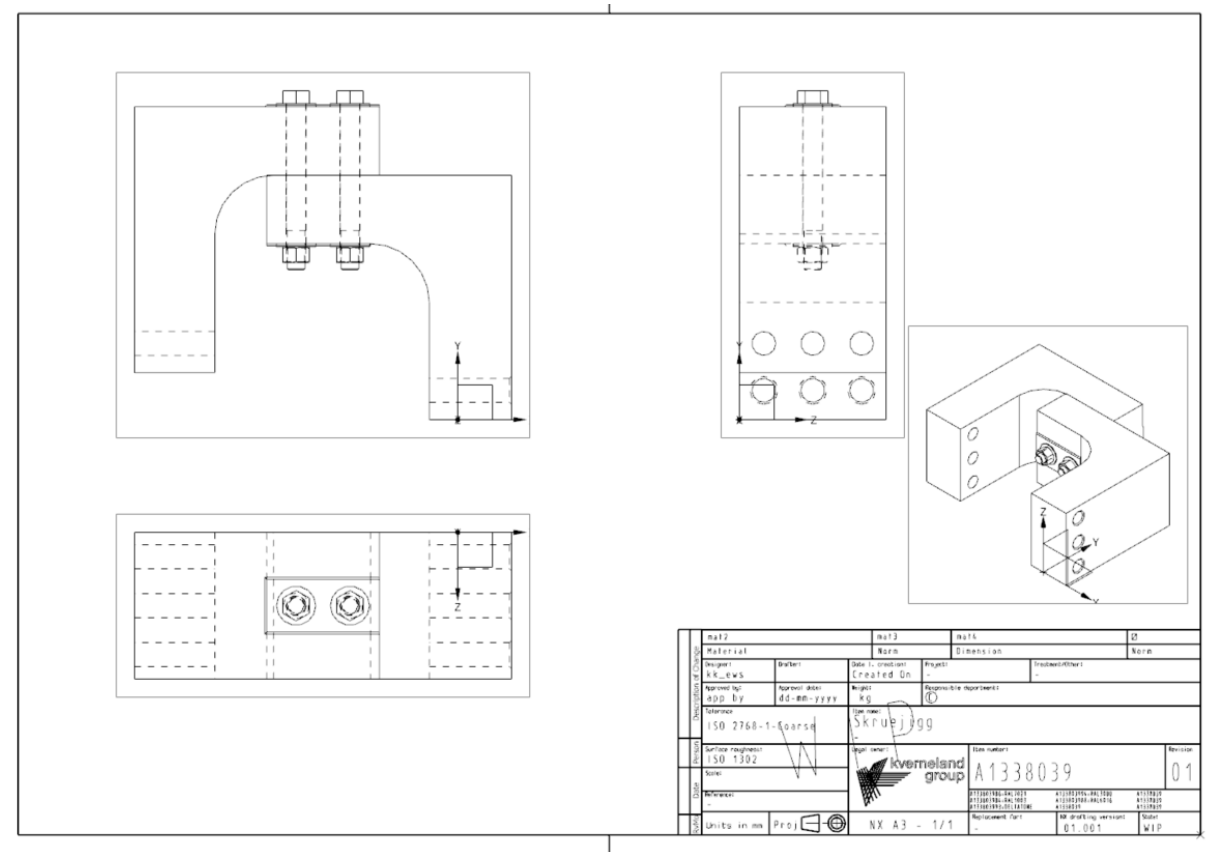
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Appendix I: Dial Wrench used for torque measurements