A New Improved Type of Friction Connection

An Experimental Study

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*“Read not to contradict and confute;
nor to believe and take for granted;
nor to find talk and discourse;
but to weigh and consider”*

-Francis Bacon 1561-1626

Preface

This thesis is the final step of my Master of Science in Engineering with the specialization Structural Engineering, at Luleå University of Technology. During my time in Luleå I have had the opportunities of meeting professors, corporations and friends who made me grow as a Civil engineer but also as an individual.

The thesis was initiated by Luleå University of Technology and Ramböll Sweden as a part of their European research and Development project PROLIFE and was written between October 2015 and April 2016.

I would like to thank all involved in this thesis for the inspiration and support. Without you it would not have been possible.

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Abstract

This master thesis is a part of a European research project named “*Prolonging life time of old steel and steel-concrete bridges*” or simply “PROLIFE”. As the name indicates the purpose is to strengthen existing bridges. The part of the project that this thesis deals with is developing and testing a new innovative friction grip connection being used to connect a truss with the bottom flanges of beams in order to make two or more beams to act like a steel box girder. The thesis also evaluates the friction grip connection experimentally at the research facility Complab, the largest laboratory at Luleå University of Technology. By increasing the effective friction coefficient the entire friction connection can withstand larger shear forces.

The hypothesis is to increase the friction coefficient between the plates in a lap connection by inserting small indenters between the plates. In this thesis three types of indenters have been used, aluminium oxide, stainless steel and tungsten carbide. All types of indenters have been tested in a compression test between two plates. The aluminium oxide and stainless steel indenter types have also been loaded in shear in a friction connection after compression of the two plates.

The necessary pre-load to impress an indenter of 2.5 mm in diameter 2.3 mm into two plates is 11 kN. The remaining 0.2 mm gap between the plates is due to the crater developed at an impression point. The protruding edges of craters on opposite plates are brought into contact with each other at 2.3 mm indentation.

The aluminium oxide increases friction but the effect is inconsistent and uncertain, since the indenters are crushed and indenting is not identical between different tests. With small indenters made of stainless steel the effective friction coefficient (μ) of the friction connection can also be improved. How much (μ) can be increased by small indenters in a friction connection depends on the existing surface condition or treatment of the plates in a joint. The tests indicate that the effect of the indenters almost equals that of a shot or grit blasted surface with loose rust removed and no pitting. A connection with indenters provides further a certain safety against overloading and the definition of the procedure is unambiguous. In a reference test with just as rolled surface conditions and no indenters, the slip resistance was 54.5 kN and the effective friction coefficient $\mu=0.09$. With 58 indenters and an M30 bolt with a pre-loading force of 320 kN the slip resistance can be expected to slightly exceed 250 kN and the effective friction coefficient to be somewhat smaller than $\mu=0.40$, at the acceptable slip of 0.15 mm, according to the current Eurocode standard. The slip resistance from the tests with different number of indenters and without indenters is shown in Figure 36 and Figure 37. Without the slip criterion the expected slip resistance for the same configuration can be 310 kN and the effective friction coefficient somewhat smaller than $\mu=0.50$. See Figure 39.

Keywords: Friction connection, hardness, indenter, slip resistance, pre-loading.

Sammanfattning

Detta examensarbete är en del av ett europeiskt forskningsprojekt vid namn "Prolonging life time of old steel and steel-concrete bridges" eller förkortat "PROLIFE". Som namnet antyder är syftet att förstärka befintliga broar. Den del av projektet som detta examensarbete omfattar är att utveckla och testa en ny och innovativ typ av friktionsförband som ska kunna användas för att fästa ett fackverk i stålbalkars underflänsar. En konstruktion med två I-balkar (eller flera) får på så sätt delvis en lådbalks funktion, där två I-balkar delar mer eller mindre broderligt även på laster som står rakt över den ena balken. Examensarbetet innefattar även en utvärdering av friktionsförbandet genom en serie försök i Complab, Luleå tekniska universitets största laboratorium. Genom att förbättra friktionskoefficienten, så kommer hela friktionsförbandet att förbättras med avseende på tvärkraftsupptagning.

Hypotesen är att kunna öka friktionskoefficienten mellan plåtarna i ett förband genom att placera små kulor mellan plåtarna. I detta examensarbete används tre olika typer av kulmaterial, aluminiumoxid, rostfritt stål och tungstenskarbid. Kulorna har pressats samman mellan två plåtar i ett kompressionstest. Kulor av aluminiumoxid och av rostfritt stål har även blivit utsatta för skjuvning i ett friktionsförband efter att de komprimerats mellan två plåtar.

Den förspänningskraft som behövs för att pressa en kula med 2,5 mm i diameter 2,3 mm in i plåtarna är 11 kN. Glipan på 0,2 mm mellan plåtarna kommer från kratern som bildas vid varje kulintryck. De uppstickande kanterna runt kulintrycket i vardera plåten kommer i kontakt vid ett intryck 2,3 mm.

Aluminiumoxiden visar sig ge ett positivt resultat i testerna. Men resultaten är inkonsekventa och därför osäkra. Detta för att aluminiumoxidkulorna krossas mellan plåtarna och inte ger samma intryck varje gång. Ett friktionsförband med små kulor av rostfritt stål kan också förbättra den effektiva friktionskoefficienten (μ). Hur mycket (μ) kan förbättras i ett friktionsförband beror på plåtarnas befintliga yta. Försöken visar att effekten av att använda kulorna är nästan lika effektiv som ytbehandlingen blästrade ytor utan lös rost och gropar. Men förbandet med kulor ger även säkerhet mot överlaster och definitionen av förfarandet är otvetydig. I ett referenstest med ren valshud och inga kulor är friktionskraften 54,5 kN och friktionskoefficienten $\mu=0,09$. Med 58 kulor och en M30 skruv med en förspänningskraft 320 kN kan friktionskraften förväntas uppnå drygt 250 kN och friktionskoefficienten uppnå knappt $\mu=0,40$, vid den enligt nuvarande Eurokodstandarden acceptabla glidningen 0,15 mm. Friktionskraften från testerna med olika antal kulor samt utan kulor visas i Figure 36 och Figure 37. Utan en begränsning av glidningen från standarden kan friktionskraften förväntas uppnå 310 kN och friktionskoefficienten uppnå knappt $\mu=0,50$. Se Figure 39.

Nyckelord: Friktionsförband, friktionskraft, förspänning, hårdhet, kula.

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Nomenclature

Roman upper letters

A	Area	[mm ²]
A _{cap}	Surface area of a spherical cap	[mm ²]
A _s	Stress area	[mm ²]
D	Diameter of indenter	[mm]
D _k	Mean diameter of hole and the outer diameter	[mm]
E	Modulus of elasticity	[MPa]
F	Force	[kN]
F ₁	Sum of friction force and contact force	[kN]
F _N	Normal force	[kN]
F _{p,C}	Pre-loading force	[kN]
F _{p,C,Rd}	Required pre-loading force	[kN]
F _{per}	Tangential force	[kN]
F _{s,Rd}	Slip resistance (friction force between surfaces)	[kN]
F _{μ,g}	Friction force between threads	[kN]
F _{μ,u}	Friction force between bolt head and plate	[kN]
K0	k-class 0	[-]
K1	k-class 1	[-]
K2	k-class 2	[-]
L ₀	Original plate length	[mm]
M _g	Tightening moment threads	[Nm]
M _{pi}	Tightening moment	[Nm]
M _{r,1}	Reference value for tightening moment 1	[Nm]
M _{r,2}	Reference value for tightening moment 2	[Nm]
M _u	Tightening moment bolt head and plate	[Nm]
O	Perimeter of a circle	
P	Pitch	[mm]
V _k	Coefficient of variation	[-]
R	Radius of indenter	[mm]

Roman lower letters

d	Bolt diameter	[mm]
d ₀	Bolt hole diameter or the diameter of the hole in the washer	[mm]
d ₁	Smallest thread diameter	[mm]
d _i	Indentation diameter	[mm]
d _m	Mean diameter	[mm]
d _w	Outer diameter of the clamping force	[mm]
e ₁	End distance from the center of a hole to the end, in the force direction	[mm]
e ₂	End distance from the center of a hole to the end, perpendicular to the force direction	[mm]
f _{ub}	Ultimate strength of the bolt	[MPa]
f _y	Yield stress of steel	[MPa]
f _{yb}	Yield stress of the bolt	[MPa]
h	Indentation	[mm]
k _i	Relation factor between the moment and the pre-loading	[-]
k _m	Mean value for either K1 or K2	[-]
k _s	Configuration of the bolt hole	[-]

n	Number of friction surfaces	[pcs]
$n_{\text{bolt},r}$	Number of bolt radii	[pcs]
p_1	The spacing between bolt holes in a line, in the force direction	[mm]
p_2	The spacing between bolt holes in a line, perpendicular to the force direction	[mm]
t	Clamping length	[mm]
t_p	Plate thickness	[mm]
x	Distance calculated with Pythagorean theorem	[mm]

Greek letters

α	Thread angle	[°]
γ_{M3}	Partial safety factor	[-]
ΔD	Diameter change in the indenter	[mm]
ΔL	Change in plate length	[mm]
ΔL_1	Change in plate length for point 1	[mm]
ΔL_2	Change in plate length for point 2	[mm]
ε	Engineering strain	[-]
ε_r	Radial strain	[-]
μ	Friction coefficient, slip factor	[-]
μ_g	Friction coefficient between threads	[-]
μ_u	Friction coefficient between bolt head and plate	[-]
ν	Poisson's ratio	[-]
π	Constant	[-]
ρ	Friction angle	[°]
σ	Stress	[MPa]
σ_1	Principal stress 1	[MPa]
σ_2	Principal stress 2	[MPa]
σ_3	Principal stress 3	[MPa]
σ_r	Radial stress	[MPa]
σ_x	Stress in x-direction	[MPa]
σ_y	Stress in y-direction	[MPa]
σ_z	Stress in z-direction	[MPa]
$\tau=\tau_{xy}$	Shear stress	[MPa]
φ	Pitch angle	[°]

Abbreviations

AISI	American Society for Testing and Materials
BHN	Brinell Hardness Number
BSK	Boverkets handbook om stålkonstruktioner
COD	Crack Opening Displacement
DTI	Direct Tension Indicator Method
FEA	Finite Element Analysis with FEM
FEM	Finite Element Method
HR	Rockwell Hardness
HRB	Rockwell Hardness B Scale
HRC	Rockwell Hardness C Scale
HRC	A tightening method used in Eurocode
ISO	International Organization for Standardization
PROLIFE	Prolonging Life Time of Old Steel and Steel-concrete Bridges
UCS	Ultimate Compressive Strength
ULS	Ultimate Limit State
UTS	Ultimate Tensile Strength

1 Introduction

1.1 Background

Since Eurocode replaced the Swedish standards in bridge design in the early 2010s, new load models and safety factors were introduced for fatigue calculations. Because of the code replacement many older composite I-profile steel bridges came not to fulfil the new requirements. Prior to Eurocode the ultimate limit state (ULS) was the design value, but after its introduction fatigue stresses in the bottom flanges of beams became critical. This because the new load models prescribed larger eccentric traffic loads, a type of loading which constitute the main source of fatigue.

Certain types of steel bridges are less affected by the new standard than I-profiles. One type that does not require stronger bottom flanges because of current fatigue loadings is the steel box girder, which is capable of smoothing out eccentric traffic loads, unlike the I-profile which carries the entire load in one beam. The smoothing effect of the steel box girder arises because its cross section is stiffer than the I-profile, a feature to prefer. Another type of bridges that are not so sensitive to fatigue are girder bridges with very large spans, where the bending moments from eigenweight will be predominant, also compared to the moments from fatigue loading.

The wish to allow heavier road traffic and the new regulations, offer Trafikverket two different options. Either replace an entire I-profile bridge in order to allow increased traffic loads (which is expensive), or strengthen the existing bridge. One way of strengthening an I-profile bridge is to connect the bottom flanges with trusses, so that the resulting structure carries loads like a steel box girder. This way of strengthening can be read more about in *“Improvement of Fatigue Resistance Through Box Action for I-girder Composite Bridges”* (Vestman, 2016).

This master thesis is a part of a European research project named *“Prolonging life time of old steel and steel-concrete bridges”* or simply “PROLIFE”. As the name indicates the purpose is to strengthening existing bridges. The part of the project that this thesis addresses is a) the development of a new innovative friction grip connection that can be used to connect a truss with the bottom flange of beams and b) evaluate the friction grip connection with experiments.

Figure 1 Friction grip connection of K-brace to bottom flange, acting as a box girder. Source: PROLIFE.

1.2 Aims and Scopes

The first aim of this work is to perform a literature survey of friction connections in order to understand their function and briefly study the difference between such connections and bolted connections. Finally also investigate advantages and disadvantages in particular of the friction connection.

The second aim is to explore whether attempts have been made to improve friction connections in a manner similar to the present or if other innovative solutions regarding friction connections already exist around the world.

The main aim of the thesis is to find out if small indenters can improve the effective friction coefficient (μ) and if so, also to determine how much the effective friction coefficient in a friction connection can be increased. By increasing the effective friction coefficient the friction connection can withstand larger shear forces.

1.3 Method

To accomplish a literature survey the necessary information is collected in different ways. In the beginning searching internet for simple facts about bolted connections and friction connections give after compilation and analysis an overall picture and an initial understanding of the mechanics of the connection. Thereafter reading technical magazines, scientific papers, literature in previous courses and other theses by previous students give a deeper understanding that is necessary for the present thesis.

To investigate whether other persons successfully has tried similar methods to improve friction connections or other innovative solutions, the material found in the literature was also forwarded to the supervisor for evaluation of the ideas found.

To fulfil the main aim of the thesis and to understand the effect of ceramic-, stainless steel- or tungsten carbide indenters in a friction connection several tests were made in Luleå University of Technology largest laboratory, Complab.

1.4 Limitations

In the thesis both bolted connections and friction connections are studied in order to get a deeper understanding of a connection. The different failure modes of a bolted connection are however neglected because of lack of interest in the present contest. On the other hand, slip resistance in a friction connection is thoroughly reviewed. Also different ways of tightening the bolts are included since the friction between the plates in contact depends upon the tensile force in the tightened bolt.

Because the main purpose of the connection is transferring load between the bottom flanges of different beams, without external forces applied to the bolts, just connections subjected to in plane shear force only are studied.

Welds and rivets are not included since welds are sensitive to fatigue loading and rivets are outdated and rarely used in structures nowadays, especially in high-strength connections.

No optimization is performed regarding the diameter of the ball indenter or their positioning around the bolt hole. Because of the limited time allotted to a master thesis, long term effects on the indenter or the connections itself are not possible to consider. Thereby relaxation, fatigue or other potential long term aspects of the connection has not been taken into account.

The test configuration used in the thesis is a double covered symmetrical butt joint with one preloaded high strength bolt. The detail category of the joint is C=112, which means that the fatigue strength at $N=2 \cdot 10^6$ cycles is 112 MPa (Swedish Standards Institute, 2005). All testing was performed in the nominal elastic region and plastic deformation of the connection will therefore not be considered in the thesis.

Figure 2 Double covered symmetrical butt joint, C=112. Source: (Swedish Standards Institute, 2005).

2 Connections

2.1 Bolted Connection

In steel structures bolted connections are used to transfer loads between members in a mechanical way (Kumar & Kumar, u.d.). A connection is obtained by assembling different members against each other and by letting bolts pass through pre-drilled aligned holes which are then secured with nuts. Thereby results a connection, capable of transferring loads and holding all parts in place. In a bolted connection the nut is only lightly tightened, since the load is thought to be transferred mainly via the bolt shank (Kumar & Kumar, u.d.).

Bolt connections are convenient in smaller constructions where the loads are moderately intense and where the failure is not through fatigue. Bolts are cheaper than other joint types and the more simple to mount than friction connections.

As mentioned above, in a bolted connection the load is transferred through the bolt shank. If the connection is subjected to a shear force there are two different failure modes. Either the connection fails in the shear plane(s) of the bolt, which mode is called shear failure, or the connection fails by deforming the bolt or the connected plates, which mode is called bearing failure (Kumar & Kumar, u.d.).

If the bolt shank is weaker than the plates then failure occurs through the shear mode. In Figure 3 a clean cut of the bolt is shown as a result from the displacement of the plates. The internal force acting in the connection are also shown.

Figure 3 Shear failure in a bolt and a free body diagram of a bolted connection where the internal forces are displayed. Source: Authors picture and (Trahair, 1977).

Bearing failure of a bolted connection occurs when the plates are weaker than the bolt, or vice versa. In the first case, the bolt may be thought of as made of a high strength steel or the plates as relatively thin in relation to the diameter of the bolt. The plate cannot resist the bolt and will fail. The failure is characterized by an elongation of the plate, as shown in Figure 4. On the other hand, when the bolt is weaker than the plates, bearing failure may occur when instead the connected plates are made of high strength steel. In that case the plates deform the bolt more or less in bending, but shear failure does not occur (Kumar & Kumar, u.d.).

Figure 4 Elongation as a result of bearing. Source: Authors picture.

2.2 Friction Connection

Transferring loads by friction between members in a connection is possible when plates are clamped together under a certain pressure. In a friction connection this feature is utilized by using pre-loaded bolts in order to transfer loads between members. All friction arises from tension in the pre-loaded bolt transferred by the tightening process (Norlin, et al., 2015). The pre-loaded force applied by the tightening process increases the tension in the bolt. This force also increases the pressure holding the plates together at the same time (Colly Company AB, 1995). The only type of bolt used in a friction connection is high strength bolt of grade 8.8 or 10.9 because of higher fatigue strength and limited deformation characteristics (Norlin, et al., 2015). Figure 5 shows a friction connection.

Figure 5 Free body diagram of a friction connection where the internal forces are displayed. Source: (Trahair, 1977).

In a friction connection bolts should be tightened to at least 70 % of their ultimate tensile strength, in order to transfer loads through friction rather than shearing of the bolt (as in a bolted connection) (Nair, et al., 1974).

As indicated by the notation of the connection (*Friction Connection*) friction is of great significance, both in the tightening process where the threads between the bolt and the nut on the one hand and the bolt head and the plate on the other are in contact with each other (Neander, 2015). Friction is also included between the plates when sliding is prevented. When tightening the bolt, only 10 % of the applied moment is effective in generating clamping force. 50 % of the applied moment is required to overcome the friction between the bolt head and the adjoining plate and 40 % is required for overcoming thread friction. So a total of 90 % of the applied work is lost in heat (Swedish Fasteners Network, 2014).

In contrast to plain bolted connections, friction connections are used in larger structures such as bridges, where fatigue loading is of concern (Chu, 2010). Friction connections are in fact also common in the car industry and similar areas with small components subjected to fatigue, but such applications will not be mentioned in further detail. When it is crucial to prevent any slight slip in a structure that can result in collapse, it is important to use the friction connection (Chu, 2010). The advantages of using friction connections in fatigue loaded structures are that

the pre-loaded high strength bolts reduce the effective stress range in contrast to a non-pre-loaded bolt. The clamping force between the plates does not induce any stress concentration around the bolt hole. This in turn leads to an increased fatigue strength in comparison to an ordinary bolted connection. Due to bolt pre-loading, the connection will also prevent bolts from loosening when subjected to vibrations (Duggal, 2000). A friction connection with high strength bolts can be lighter than a bolted connection since bolts in a bolted connection are weaker and the connection requires a greater number of bolts to obtain the same strength as in a friction connection. This leads to smaller and lighter connections which in turn give less expensive connections (Carlunger, et al., 1999). A disadvantage of a pre-loaded friction grip joint is the special procedures and tightening methods required to ensure that the necessary clamping action is achieved.

Because friction transfer in-plane loads between the members of a joint, the resistance of the bolt shank to shear failure is irrelevant in theory (Kuzmanović & Willems, 1983). The tension in the bolt shank ensures that a shear failure mode does not occur, since the tension generated in the bolt now clamp the plates together and transfer the loads between the faying surfaces with friction. This means that the failure mode typical of a bolted connection cannot occur in a friction connection. In a friction connection basically only the slip-resistance is of interest. This is however theoretically correct only because in theory the bolt is perfectly centred in centred plate holes so that only the bolt head and the nut are in contact with the plates. In reality however the bolt shank may be in contact with the plates due to the difficulty in centring plate holes and bolt. Therefore the bearing resistance should also be checked (Moore, et al., 2003).

Figure 6 Bolt shank in contact with the plate leads to bearing failure. Source: (SteelConstruction.info, 2012).

3 Tensions in Bolt

To be able to calculate the tightening moment (M_{pi}) yielding a given pre-loading force ($F_{p,C}$), an understanding of the geometry of a bolt is needed.

A bolt can be seen as a cylinder with threads and a head. The thread angle, or half the flank angle, is denoted by the Greek letter alpha (α). The end of the bolt opposite the head is free. To mount a nut to the bolt, the nut must be correspondingly threaded. Definition of the pitch angle is seen in Figure 7.

Figure 7 Geometry of a bolt and the pitch angel. Source: (PENCOM, u.d.).

3.1 Pre-loading Force

When a pre-loading force is applied to a bolt the flank pressure between the threads of the bolt and of the nut results in a normal force (F_N), perpendicular to the flank surface at the mean diameter (d_m), see Figure 8 (Eriksson, et al., 1991).

Figure 8 Normal force perpendicular to the flank surface at d_m and a free body diagram. Source: Inspired from (Eriksson, et al., 1991).

The force F_N can be resolved into components, see Figure 8, where the component $F_N \sin \alpha$ acts radially in the bolt and therefore does not enter in the equilibrium equations below. Non-radial forces are shown in Figure 9. The pitch angle (φ) of the thread is the angle between the longitudinal axis of the bolt and the component $F_N \cos \alpha$. $F_{p,C}$ is the axial pre-loading force of the bolt. $F_{\mu,g}$ is a friction force acting tangentially along the flank surface in a direction opposite to the applied rotation. The circumferential force (F_{per}), acting at the mean diameter (d_m), is equivalent to the tightening moment (M_g) (Eriksson, et al., 1991).

Figure 9 Acting forces in the axial plane of the bolt considering the pitch angel and a free body diagram. Source: (Eriksson, et al., 1991).

From Figure 9 equilibrium equations are obtained axially and circumferentially:

$$\uparrow: F_N \cos \alpha \cos \varphi - F_{\mu,g} \sin \varphi - F_{p,C} = 0 \quad (1)$$

$$\rightarrow: -F_N \cos \alpha \sin \varphi - F_{\mu,g} \cos \varphi + F_{per} = 0 \quad (2)$$

The friction force ($F_{\mu,g}$) is related to the flank surface normal force (F_N) through,

$$F_{\mu,g} = \mu_g F_N \quad (3)$$

where μ_g is the friction coefficient, and the circumferential force (F_{per}), to the tightening moment (M_g) through,

$$F_{per} = \frac{2M_g}{d_m} \quad (4)$$

Inserting Equation (3) in (1) and (2) and Equation (4) in (2) and simplifying yields,

$$F_N (\cos \alpha \cos \varphi - \mu_g \sin \varphi) = F_{p,C} \quad (5)$$

$$F_N (\cos \alpha \sin \varphi + \mu_g \cos \varphi) = \frac{2M_g}{d_m} \quad (6)$$

Now dividing Equation (6) by (5) and solving M_g yields,

$$M_g = F_{p,c} \frac{d_m}{2} \frac{\cos \alpha \sin \varphi + \mu_g \cos \varphi}{\cos \alpha \cos \varphi - \mu_g \sin \varphi} \quad (7)$$

Multiplying both numerator and denominator with the factor $\frac{1}{\cos \alpha \cos \varphi}$ on the right side of the equation and simplifying yields further,

$$M_g = F_{p,c} \frac{d_m}{2} \frac{\frac{\cos \alpha \sin \varphi}{\cos \alpha \cos \varphi} + \frac{\mu_g \cos \varphi}{\cos \alpha \cos \varphi}}{\frac{\cos \alpha \cos \varphi}{\cos \alpha \cos \varphi} - \frac{\mu_g \sin \varphi}{\cos \alpha \cos \varphi}} = F_{p,c} \frac{d_m}{2} \frac{\tan \varphi + \mu_g \frac{1}{\cos \alpha}}{1 - \mu_g \frac{\tan \varphi}{\cos \alpha}} \quad (8)$$

By extracting $\tan \varphi$ from the numerator of the equation is obtained.

$$M_g = F_{p,c} \frac{d_m \tan \varphi}{2} \frac{1 + \mu_g \frac{1}{\cos \alpha \tan \varphi}}{1 - \mu_g \frac{\tan \varphi}{\cos \alpha}} \quad (9)$$

The distance along the thread corresponding to one revolution, at the diameter (d_m), is,

$$O = \pi d_m \quad (10)$$

By developing one revolution of the thread, see Figure 10 the thread pitch (P) is obtained from (Swedish Fasteners Network, 2014).

Figure 10 The definition of a pitch (P). Source: (Sandvik Coromant, 2010).

$$\tan \varphi = \frac{P}{\pi d_m} \quad (11)$$

Inserting Equation (11) in (9) and simplifying results in,

$$M_g = F_{p,c} \frac{P}{2\pi} \frac{1 + \mu_g \frac{1}{\cos \alpha \tan \varphi}}{1 - \mu_g \frac{\tan \varphi}{\cos \alpha}} \quad (12)$$

With some approximations and further derivations, a somewhat simpler expression for the tightening moment, resulting in less than 5 % deviation from Equation (12), is obtained (Broberg & Gustafsson, 1983).

$$M_g = F_{p,c} \left(\frac{P}{2\pi} + \frac{\mu_g d_m}{2 \cos \alpha} \right) \quad (13)$$

Equation (13) is fully derived in Appendix A.

There is an alternative way to derive Equation (13) which more clearly reveals the deviations of the equation, arising from the approximations made which are dependent upon the choice of the friction coefficient (μ_g). Define a friction angle (ρ) as the ratio between the friction force ($F_{\mu,g}$), Equation (3), and the contact force component ($F_N \cos \alpha$), Equation (14), (Eriksson, et al., 1991). The force (F_1) in Figure 11 is the vector sum of $F_{\mu,g}$ and $F_N \cos \alpha$.

Figure 11 Free body diagram of the acting forces in the axial plane with the defined friction angle (ρ). Source: (Eriksson, et al., 1991).

$$\tan \rho = \frac{\mu_g F_N}{F_N \cos \alpha} = \frac{\mu_g}{\cos \alpha} \quad (14)$$

With F_1 replacing $(F_{\mu,g})$ and $(F_N \cos \alpha)$ the following equilibrium equations are obtained from Figure 11.

$$\uparrow: F_1 \cos(\rho + \varphi) - F_{p,c} = 0 \quad (15)$$

$$\rightarrow: -F_1 \sin(\rho + \varphi) + F_{per} = 0 \quad (16)$$

First solving for $F_{p,c}$ and F_{per} , then dividing the resulting equations and inserting (4) and finally solving for M_g , yields,

$$M_g = F_{p,c} \frac{d_m}{2} \frac{\sin(\rho + \varphi)}{\cos(\rho + \varphi)} = F_{p,c} \frac{d_m}{2} \tan(\rho + \varphi) \quad (17)$$

The tangent theorem of the sum of two angles is,

$$\tan(\rho + \varphi) = \frac{\tan \rho + \tan \varphi}{1 - \tan \rho \tan \varphi} \quad (18)$$

Inserting Equation (18) in (17) and then Equation (11) and (14) in the resulting numerator.

$$M_g = F_{p,c} \frac{d_m}{2} \frac{\frac{\mu_g}{\cos \alpha} + \frac{P}{\pi d_m}}{1 - \tan \rho \tan \varphi} \quad (19)$$

In Equation (19), the term (in the denominator) $\tan \rho \tan \varphi \ll 1$, see Table 1 below. Thus the term can therefore in practice be neglected with only a minor difference in the result. In Table 1 the deviation due to the approximation is shown for different friction coefficients and bolt sizes for the ISO-profile MC6S. All Table 1 calculations are given in Appendix B.

Table 1 Percent deviation in the result due to the approximation $\tan \rho \tan \varphi = 0$.

μ_g	M12	M14	M16	M18	M20	M22	M24	M27	M30	M33	M36
0.1	0.6	0.6	0.5	0.6	0.5	0.5	0.5	0.4	0.5	0.4	0.4
0.2	1.2	1.2	1.0	1.1	1.0	0.9	1.0	0.9	0.9	0.8	0.9
0.3	1.8	1.8	1.5	1.7	1.5	1.4	1.5	1.3	1.4	1.3	1.3
0.4	2.4	2.4	2.0	2.3	2.0	1.8	2.0	1.8	1.9	1.7	1.8
0.5	3.1	3.0	2.6	2.9	2.6	2.3	2.6	2.3	2.4	2.1	2.3
0.6	3.7	3.6	3.1	3.5	3.1	2.8	3.1	2.7	2.9	2.6	2.7
0.7	4.3	4.2	3.6	4.1	3.6	3.3	3.6	3.2	3.4	3.0	3.2
0.8	5.0	4.9	4.2	4.7	4.2	3.7	4.2	3.6	3.9	3.5	3.6
0.9	5.6	5.5	4.7	5.3	4.7	4.2	4.7	4.1	4.4	3.9	4.1
1.0	6.3	6.1	5.3	5.9	5.3	4.7	5.3	4.6	4.9	4.4	4.6

Since the deviation due to the approximation is not larger than 6.3 % it is a reasonable assumption that the term “ $\tan \rho \tan \varphi$ ” can be neglected in Equation (19). This assumption is also implicit in Equation (13), repeated below.

$$M_g = F_{p,c} \left(\frac{P}{2\pi} + \frac{\mu_g d_m}{2 \cos \alpha} \right)$$

Equation (13) is the final equation in practice of the tightening moment needed to achieve a given pre-loading force in a clamped bolt. However, the equation contains only the friction coefficient of the threads (that is, relating to the contact surface between bolt and nut), while in a bolt friction also develops in the contact surface between the bolt head and the underlying plate and likewise as regards the nut. The corresponding friction force ($F_{\mu,u} = \mu_u F_{p,c}$) is assumed acting tangentially at the mid-radius of a ring-shaped friction surface, defined as the active clamping surface between the bolt head/nut and the underlying plate. If a washer is inserted between the bolt head and the plate the surface is defined as the washer area (Swedish Fasteners Network, 2014).

Figure 12 The moment contribution from the friction between the bolt head and the plate. Source: (Eriksson, et al., 1991).

The contribution from the bolt head friction to the total required tightening moment can be derived from the relation between the tightening moment and the friction force ($F_{\mu,u}$).

Moment equilibrium around the longitudinal axis of the bolt yields,

$$\sum x: \quad M_u - \mu_u F_{p,c} \left(\frac{d_0 + d_w}{4} \right) = 0 \Leftrightarrow M_u = \mu_u F_{p,c} \left(\frac{d_0 + d_w}{4} \right) \quad (20)$$

The mid-diameter of the ring-shaped friction surface is,

$$D_k = \frac{d_0 + d_w}{2} \quad (21)$$

where,

d_0 is the diameter of the plate hole or of the washer hole.

d_w is the outer diameter of the ring-shaped clamping surface.

Insert Equation (21) in (20).

$$M_u = \mu_u F_{p,c} \frac{D_k}{2} \quad (22)$$

The total tightening moment for a given pre-loading force is obtained by adding the Equations (13) and (22).

$$M_{pi} = M_g + M_u = F_{p,c} \left(\frac{P}{2\pi} + \frac{\mu_g d_m}{2 \cos \alpha} + \frac{\mu_u D_k}{2} \right) \quad (23)$$

The first of the three terms inside the bracket (multiplied with $F_{p,c}$) represents the moment needed to obtain a specific pre-loading force and the other two terms the proportions of the applied moment which are needed to overcome the thread friction and the bolt head-plate friction, respectively.

By testing bolts of different sizes and different friction coefficients, the proportions between the respective terms can be obtained. By taking mean values from the testing results it can be seen that the proportions is ~10 % for the first term, ~40 % for the second and ~50 % for the third term. These proportions agree with other different sources, for example (Malmqvist, 2010). However it is shown by calculation in this thesis that the proportions are valid only for friction coefficients in the range 0.14–0.20. For friction coefficients outside of this range the first term may vary significantly. For friction coefficients smaller than 0.14 the first term can increase up

to 25 % for $\mu_g = \mu_u = 0.06$, while for coefficients above 0.20 the first term may decrease to 5 % for $\mu_g = \mu_u = 0.3$. From the testing results it can also be seen that the bolt size has only a minor influence on the proportions. Further calculations on this subject can be found in Appendix C.

3.2 Slip Resistance

The relation between the normal force and the friction force is used to derive the slip resistance in Equation (24). Where the normal force is equal to the pre-loading force and the friction force only has a new name (the slip resistance). The coefficient (μ) is the friction coefficient between the plates in the slip surface.

$$F_{s,Rd} = \mu F_N \quad (24)$$

This equation describes the slip resistance per slip surface.

3.3 Eurocode

Eurocode prescribes that only bolts of classes 8.8 and 10.9 are allowed in friction connections, corresponding to the general theory. A friction connection subjected to shear load at ultimate limit state belongs to category C. Slip-resistance, bearing resistance and the plastic resistance of the net cross-section at the bolt holes are therefore failure modes that in addition should be verified separately according to Eurocode (Swedish Standards Institute, 2005).

For a friction connection, Eurocode also prescribes the placing of bolt holes and their size. In the figures below, parts from Eurocode are included to illustrate the procedure (Swedish Standards Institute, 2005).

Distances and spacings, see Figure 3.1	Minimum	Maximum ^{1) 2) 3)}		
		Structures made from steels conforming to EN 10025 except steels conforming to EN 10025-5		Structures made from steels conforming to EN 10025-5
		Steel exposed to the weather or other corrosive influences	Steel not exposed to the weather or other corrosive influences	Steel used unprotected
End distance e_1	$1,2d_0$	$4t + 40$ mm		The larger of $8t$ or 125 mm
Edge distance e_2	$1,2d_0$	$4t + 40$ mm		The larger of $8t$ or 125 mm
Distance e_3 in slotted holes	$1,5d_0$ ⁴⁾			
Distance e_4 in slotted holes	$1,5d_0$ ⁴⁾			
Spacing p_1	$2,2d_0$	The smaller of $14t$ or 200 mm	The smaller of $14t$ or 200 mm	The smaller of $14t_{\min}$ or 175 mm
Spacing $p_{1,0}$		The smaller of $14t$ or 200 mm		
Spacing $p_{1,i}$		The smaller of $28t$ or 400 mm		
Spacing p_2 ⁵⁾	$2,4d_0$	The smaller of $14t$ or 200 mm	The smaller of $14t$ or 200 mm	The smaller of $14t_{\min}$ or 175 mm
¹⁾ Maximum values for spacings, edge and end distances are unlimited, except in the following cases: <ul style="list-style-type: none"> – for compression members in order to avoid local buckling and to prevent corrosion in exposed members and; – for exposed tension members to prevent corrosion. ²⁾ The local buckling resistance of the plate in compression between the fasteners should be calculated according to EN 1993-1-1 using $0,6 p_1$ as buckling length. Local buckling between the fasteners need not to be checked if p_1/t is smaller than 9ε . The edge distance should not exceed the local buckling requirements for an outstand element in the compression members, see EN 1993-1-1. The end distance is not affected by this requirement. ³⁾ t is the thickness of the thinner outer connected part. ⁴⁾ The dimensional limits for slotted holes are given in 1.2.7 Reference Standards: Group 7. ⁵⁾ For staggered rows of fasteners a minimum line spacing of $p_2 = 1,2d_0$ may be used, provided that the minimum distance, L , between any two fasteners is greater or equal than $2,4d_0$, see Figure 3.1b).				

Figure 13 Placing of bolt holes. Source: (Swedish Standards Institute, 2005).

Figure 14 Visualization of bolt hole distances. Source: (Swedish Standards Institute, 2005).

Nominal bolt or pin diameter d (mm)	12	14	16	18	20	22	24	27 and over
Normal round holes ^a	1 ^{b,c}		2					3
Oversize round holes	3		4				6	8
Short slotted holes (on the length) ^d	4		6				8	10
Long slotted holes (on the length) ^d	1,5 d							

^a For applications such as towers and masts the nominal clearance for normal round holes shall be reduced by 0,5 mm unless otherwise specified.

^b For coated fasteners, 1 mm nominal clearance can be increased by the coating thickness of the fastener.

^c Bolts with nominal diameter 12 and 14 mm, or countersunk bolts may also be used in 2 mm clearance holes under conditions given in EN 1993-1-8.

^d For bolts in slotted holes the nominal clearances across the width shall be the same as the clearances on diameter specified for normal round holes.

Figure 15 Bolt hole size. Source: (Swedish Standards Institute, 2011).

The pre-loading force of a bolt is calculated according to,

$$F_{p,c} = 0.7f_{ub}A_s \quad (25)$$

The constant 0.7 means that a bolt should be tightened to only 70 % of the ultimate tensile strength. This means the Eurocode indeed permits just the minimum tightening needed to obtain a fully functioning friction connection (Swedish Standards Institute, 2005).

The yield strength (f_{yb}) and the ultimate tensile strength (f_{ub}) for different bolt classes are shown in Table 2 (Swedish Standards Institute, 2005).

Table 2 Yield strength and ultimate tensile strength of bolt classes 8.8 and 10.9.

Bolt class	8.8	10.9
f_{yb} (N/mm ²)	640	900
f_{ub} (N/mm ²)	800	1000

The stress area of a bolt (A_s) is calculated according to the expression,

$$A_s = \frac{\pi \left(\frac{d_1 + d_m}{2} \right)^2}{4} \quad (26)$$

where $(d_1 + d_m)/2$ is the mean value of the mean and the smallest thread diameter.

Bolt diameter and stress area are shown in Table 3.

Table 3 Bolt diameters and stress area.

Bolt diameter	12	14	16	18	20	22	24	27	30	33	36
d_1	9.85	11.55	13.55	14.93	16.93	18.93	20.32	23.32	25.71	28.71	31.09
d_m	10.86	12.70	14.70	16.38	18.38	20.38	22.05	25.05	27.73	30.73	33.40
A_s [mm ²]	84.2	115	157	192	245	303	352	459	561	694	817

In Table 4 the pre-loading force for a given bolt class and diameter is shown.

Table 4 Pre-loading force in kN for a given bolt class and bolt diameter.

Bolt class	Bolt diameter [mm]							
	12	16	20	22	24	27	30	36
8.8	47	88	137	170	198	257	314	458
10.9	59	110	172	212	247	321	393	572

The slip resistance is calculated according to the theoretical expression adjusted with regard to safety,

$$F_{s,Rd} = \frac{k_s n \mu}{\gamma_{M3}} F_{p,C} \quad (27)$$

where k_s is bolt hole configuration factor, Table 5, n the number of friction surfaces, μ friction coefficient, Table 6, and $\gamma_{M3}=1.25$ partial safety factor (Swedish Standards Institute, 2005).

Eurocode defines the slip resistance as the force at 0.15 mm slip relative each other of two adjacent plates (Swedish Standards Institute, 2011).

Table 5 Bolt hole configuration safety factor. Source: (Swedish Standards Institute, 2005).

Description	k_s
Bolts in normal holes.	1.0
Bolts in either oversized 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 6 Friction coefficient for different surface treatment according to Eurocode. Source: (Swedish Standards Institute, 2011).

Surface treatment	Class	Slip factor μ
Surfaces blasted with shot or grit with loose rust removed, not pitted.	A	0.50
Surfaces blasted with shot or grit: a) spray-metallized with an aluminium or zinc based product; b) with alkali-zinc silicate paint with a thickness of 50 μm to 80 μm .	B	0.40
Surfaces cleaned by wire-brushing or flame cleaning, with loose rust removed.	C	0.30
Surfaces as rolled.	D	0.20

4 Tightening Methods

As shown in the previous chapter, the resulting friction force of a joint, due to a given applied bolt moment, may vary considerably depending on the bolt head and thread friction coefficients. This means that the applied moment in general is not the same for all kinds of connections, but must specifically be calculated in each case so that the desired clamping force is obtained. In order to save time in the designing process, Eurocode provides simplified methods giving approximate solutions. The friction coefficients are included in the tightening methods in terms of k -classes (Swedish Standards Institute, 2011), which are provided by the manufacturer. Depending on the type of tightening tool available, different bolt systems with different k -classes are required. Or, if a particular bolt system is available, the tightening procedure is chosen accordingly.

Since the introduction of Eurocode the accuracy of the moment applied to a bolt has become of a greater importance than before. In the earlier Swedish standard BSK 07 an approved method for use in practice was just to select the lever arm length according to the bolt size and then use “manskraft” (hand power) applied to a torque wrench (Boverket, 2007). Accordingly, it is not surprising that bolt failures previously have occurred due to thread stripping or to insufficient bolt preloading. The current Eurocode offers however four different optional methods in order to achieve accurate pre-loading force economically (Swedish Standards Institute, 2011).

Figure 16 Bolt failure due to thread stripping. Source: (Auto Repair Instructions, u.d.).

The desired pre-loading force is at least 70 % of the ultimate tensile force of the bolt according to Equation (25).

All tightening according to Eurocode shall be performed by rotating the nut.

The four methods of pre-loading are summarized below. The different k -classes are denoted by K0, K1 or K2.

4.1 The Torque Method

The necessary k -class for the Torque method is K2. In this class the manufacturer has tested several bolt systems in order to provide an effective value (k_m) for a specific type of bolt system. The parameter k_m is the mean value of several k_i s and as such an effective summary of all factors affecting the relationship between the tightening moment and the desired given pre-loading. The k -class is approved as K2 if $0.10 \leq k_m \leq 0.23$ and if the coefficient of variation (V_k) is $V_k \leq 0.06$. For this method the torque reference value ($M_{r,2}$) is calculated as,

$$M_{r,2} = k_m d F_{p,C} \quad (28)$$

The Torque method comprises two tightening steps.

- a) The wrench is first set to 75 % of the torque reference value. When all bolts in the connection are tightened to this level the next step is initiated.
- b) The wrench is now set to 110 % of the torque reference value and all bolts in the connection are tightened to this level.

Eurocode mentions that effects of relaxation are already considered in the different tightening methods, but still notes that the Torque method requires retightening after a couple of days. This sounds contradictory.

4.2 The Combined Method

The necessary k -classes for the Combined method are K1 or K2. K2 is determined in the same way as in the torque method. The k -class is K1 if $0.10 \leq k_i \leq 0.16$. k_i is calculated by the manufactures in the following way after testing the bolt system.

$$k_i = \frac{M_{pi}}{d F_{p,C}} \quad (29)$$

where

d is the bolt diameter, $F_{p,C}$ the pre-loaded force and M_{pi} the moment corresponding to the pre-loading force ($F_{p,C}$) according to Figure 17.

Figure 17 Bolt force versus torque curve. Source: (Swedish Standards Institute, 2015).

For this method the torque reference value $M_{r,2}$ is calculated according to Equation (28) and $M_{r,1}$ is calculated as shown below.

$$M_{r,1} = k_m d F_{p,c} \quad (30)$$

where the k_m is the mean value for either K1 or K2.

The Combined method comprises two tightening steps.

- a) The wrench is first set to 75 % of the torque reference value. When all bolts in the connection are tightened to this level the next step is initiated.
- b) The angular position of the nut is marked and the bolt is then further rotated an angle given in Table 7. The angle depends on the clamping length including all sealing details and washers.

Table 7 The Combined method: additional rotation.

Clamping length (t), bolt diameter (d)	Angle [Degrees, °]
$t < 2d$	60
$2d \leq t < 6d$	90
$6d \leq t \leq 10d$	120

4.3 HRC- Method

The necessary k -class for the HRC- method is K0, but if a tightening tool (wrench) cannot be used, for example because of limited space, the Torque control method should be for the HRC bolt system with the k -class K2. K0 means that there are no requirements for k -factors.

The HRC- method uses a certain shear wrench equipped with two sockets, one attached to the spline end and the other to the nut, but rotated in opposite directions. The resulting moment applied to the bolt ensures that the necessary pre-loading is obtained given the geometrical and torsional characteristics of the bolt together with the lubrication conditions. An overview of the two tightening steps is shown in Figure 18 below.

The HRC- method comprises two tightening steps.

- a) The outer socket is rotated clockwise while the inner socket holding the spline does not rotate. When the outer socket stops turning the first step is done. When all bolts in the connection are fastened the next step is initiated.
- b) The inner socket is now rotated counter-clockwise until the spline end shears off at the break- neck.

Figure 18 An overview of the tightening steps of the HRC- method. Source: (Berenbak, 2012).

4.4 Direct Tension Indicator Method (DTI- Method)

The DTI-method relies on a compressible washer, designed with indicator protrusions (elevated areas on the washer). Depending on the bolt grade and the placing of the DTI-washer, a bolt face washer is placed between the bolt head and the elevated areas of the DTI-washer. For bolts of grade 10.9 there is a need of such a washer. Markings on the washer indicate whether the washer is intended for an 8.8 bolt or a 10.9 bolt. Usually the washer is placed between the bolt head and the plate, where no rotation occurs, but can alternatively be placed under the nut if for example space is limited. If placed under the nut, a nut face washer is also needed between the nut and the elevated areas of the DTI- washer.

This method is valid for all k -classes because the compressible washer itself indicates when the required pre-load has been achieved. A feeler gauge helps determine when the washer is compressed enough. Depending on the number of indicator protrusions and the placement of the DTI- washer, a certain thickness of the feeler gauge is required, see Table 8. A similar condition applies to the acceptable number of feeler gauge refusals, see Table 9.

Figure 19 DTI- washer with indicator protrusions visible. Source: (Swedish Standards Institute, 2011).

The DTI- method comprises two tightening steps.

- a) The nut is tightened until all indicator protrusions begin to compress. When all bolts in the connection are fastened accordingly, the next step is initiated.
- b) The elevated areas on the DTI- washer are then successively compressed until the acceptable number of feeler gauge refusals is reached.

Table 8 The DTI- method: Thickness of feeler gauge.

Indicator positions	Thickness of feeler gauge [mm]
Under bolt head, nut rotated	0.40
Under nut, bolt rotated	
Under nut, nut rotated	0.25
Under bolt head, bolt rotated	

This table applies to DTI- washers suitable for both bolt grade 8.8 and 10.9.

Table 9 The DTI- method: Feeler gauge refusals.

Number of indicator protrusions	Minimum number of feeler gauge refusals
4	3
5	3
6	4
7	4
8	5
9	5

At most 10 % of the indicator protrusions can be fully compressed.

5 Innovative Friction Connection

A literature survey regarding similar innovative methods have been conducted. The closest solution found in relation to existing methods is the 3M™ Friction Shims method, which includes a steel foil with a nickel diamond coating. It is mainly used in vehicles to increase the static friction coefficient between rotating bolt mounted engine parts, but also in friction connections in wind energy applications. The friction coefficient may attain $\mu=0.7$ according to their sales brochure (3M, 2016).

Figure 20 Illustration of the Friction Shims. Source: (3M, 2016).

Otherwise mainly improvements of connections in which composite elements are tightened together, using metal inserts in the bolt hole, were found.

In particular, one work of interest, claims that appropriate connections are necessary when fibre reinforced polymer materials are used in bridges (Mara, et al., 2015). In this work a double covered symmetrical butt joint is used with a pre-loaded bolt. To the bolt hole was added a steel insert, some untreated and other covered with a thin thermally sprayed layer containing tungsten carbide particles of dimension 0.1 ± 0.03 mm. The friction coefficient (μ) of the untreated insert varies between 0.1 and 0.2, and that of the treated insert between 0.41 and 0.58. This result confirms that the surface modification effectively increases the friction coefficient (Mara, et al., 2015).

Figure 21 Configuration of the steel insert. Source: (Mara, et al., 2015).

Figure 22 Steel inserts in a connection with the load transfer. Source: (Mara, et al., 2015).

6 Properties of the Spherical Indenter

The choice of using indenters to increase the effective friction of a friction joint seems to be a good idea and worth exploring. Especially after the literature survey was conducted, since it promoted a further investigation of the subject.

When choosing indenter material, it is important to understand material behaviour when loaded. The crystal structure, inter-atomic bonding forces and grain size are three characteristics that affect the mechanical properties of a material (Singh & Dwivedi, 2014) & (Dieter Jr., 1961).

6.1 Brittleness and Ductility

Whether a material is brittle or ductile depends on its ability to deform plastically when subjected to a load (Dieter Jr., 1961). A brittle material fractures suddenly without any previous yielding and only absorbs a small amount of energy before failure. In contrast to a brittle material, a ductile material deforms plastically before failure. This property allows the material to change its shape prior to failure (Dieter Jr., 1961) & (Davis, 2004). Ceramics are examples of brittle materials and modern mild and high-strength steels are ductile materials (Rösler, et al., 2007).

It is important to understand that brittleness and ductility of a material on the one hand and strength on the other are different material properties. Strength is in short the load bearing capacity of a material (Rösler, et al., 2007), while ductility is the capacity of plastic deformation before failure (and brittleness the absence thereof). This makes it important not only to consider the ultimate strength of a material, but also the material properties. Depending upon the stress state, the behaviour of a given material can vary between brittle and ductile. A three-axial stress state promotes brittleness, while uni- and bi-axial stress state promotes ductile behaviour (Dieter Jr., 1961).

If the indenter material is too ductile the indenter only deforms without penetrating the plates, therefore the hardness both of indenter and plate materials needs to be taken into account.

6.2 Hardness

The mechanical property hardness is related to the resistance of a material against scratching, abrasion or indentation (Singh & Dwivedi, 2014). Scratching is mostly of interest for mineralogists working with minerals. Abrasion hardness is usually used in softer materials and the resistance is measured by rubbing a harder material against the sample. Indentation by pressing a hard object into another material, so as to leave a permanent indenter mark, is of interest for engineers. Indentation hardness is defined as the resistance to plastic deformation of a material and is obtained from measurement of the indenter mark size for a given indenter force. Two indentation hardness tests are the Brinell hardness test and the Rockwell hardness test (Dieter Jr., 1961).

6.2.1 Brinell (BHN)

The Brinell hardness test was invented by the Swedish engineer J.A. Brinell in 1900. The hardness test is conducted by pressing a spherical steel indenter under a constant load during 30 seconds into the material of interest. The constant load is related to the diameter of the indenter. For example mild steel, the diameter of the indenter is 10 mm and the load 3000 kg, as originally stated (≈ 30 kN) and for harder materials the indenter diameter is 2.5 mm and the load 187.5 kg (≈ 18.4 kN) (Dieter Jr., 1961).

Figure 23 Brinell hardness test on the plate from the first test. Source: Authors picture.

The Brinell hardness number is defined as the load (F) divided by the surface area of a spherical cap (A_{cap}) (Dieter Jr., 1961).

Figure 24 Free body diagram of a Brinell hardness test. Source: Authors picture.

The surface area of the spherical cap is calculated as,

$$A_{cap} = \pi D h \quad (31)$$

where the indentation (h) is the radius of the sphere minus the distance (x) calculated with the Pythagorean Theorem.

$$h = \frac{D}{2} - x = \frac{D}{2} - \sqrt{\left(\frac{D}{2}\right)^2 - \left(\frac{d_i}{2}\right)^2} = \frac{D}{2} \left(1 - \sqrt{1 - \left(\frac{d_i}{D}\right)^2}\right) \quad (32)$$

By inserting Equation (32) into Equation (31), and then dividing the load (F) with the resulting expression, the Brinell hardness number is given by,

$$BHN = \frac{F}{A_{cap}} = \frac{2F}{\pi D^2 \left(1 - \sqrt{1 - \left(\frac{d_i}{D}\right)^2}\right)} \quad (33)$$

It is important to remember that the unit of the load (F) is kg and that of the diameter is mm.

An empirical relationship between the Brinell hardness number and the ultimate strength of a given material was found rather early for mild and medium strength structural and machine steels. The relationship is simply (Bhushan, 2013),

$$f_{ub} = 3.45BHN \quad (34)$$

where f_{ub} is the ultimate tensile strength.

6.2.2 Rockwell (HRC)

The Rockwell hardness scale was invented in 1919 by Stanley P. Rockwell. There are 15 different regular Rockwell scales which varies depending on the type of indenter, the pre-load and the total load applied to the test material (Low, 2001). The Rockwell hardness test is commonly used because of its speed, no human errors can occur, the ability to distinguish small hardness differences in hardened steel and the small size of indentation (Dieter Jr., 1961).

Figure 25 Rockwell hardness test on the plate from the first test. Source: Authors picture.

The hardness test is conducted in two steps. The material to be tested is first subjected to a small pre-load a certain prescribed time in order to minimize the need of surface preparation. Second, the major load is applied and left for its prescribed time. Finally, the difference in indent depths at the loads in the first and second step are measured. The Rockwell hardness test apparatus when started does all the work itself and then displays the result on its screen (Dieter Jr., 1961).

Rockwell hardness number 100, HR 100, is defined by the indenter depth 0.00008 in, ~0.002 mm and is assumed to be a maximum hardness. The hardness scale goes from 0–100 and increases with hardness. The Rockwell hardness number is dimensionless unlike the Brinell hardness (Dieter Jr., 1961). The Rockwell hardness number is given (Low, 2001),

$$HR = 100 - \frac{h}{0.002} \quad (35)$$

where h is the indentation in mm.

The Rockwell hardness is a suitable measure for materials of hardness HRC 20–70. For smaller values the Rockwell scale HRB is preferred. HRC features a 120° conical diamond indenter with a spherical tip, also called a “Brale indenter”. The HRC pre-load is 10 kg and the final load 150 kg (Dieter Jr., 1961).

6.3 Fatigue

The fatigue strength is the maximum stress range a material is capable of withstanding under a certain large number of load cycles (Bathias & Pineau, 2010). In a bridge the traffic load subject the indenters in a friction connection to a number of sliding cycles along plate grooves. Indenter fatigue may be crucial regarding the function of the connection, but this subject, due to its complexity, falls outside the scope of this work. Two important questions in this context are: How is the effective joint friction coefficient affected by sliding cycles and how are the indenters affected by fatigue loading?

6.4 Deformation

6.4.1 Elastic Deformation

A material is elastic if a body of the material returns to its original shape upon removal of an applied load. The full recovery of the original shape can either be instant or delayed. If the recovery is instant the elastic deformation is time independent, otherwise time dependent. If the elastic deformation is time dependent the material has properties that best can be described as viscoelastic properties. Mainly this property is observed in polymers (Rösler, et al., 2007). A material can either be elastic or not, since it is impossible for a material to be “more” elastic than another material (Roylance, 2008).

The elastic deformation in the neighbourhood of an indentation is described by the “Hertz theory of elastic contact” (Heinrich Hertz, 1882). Hertz wanted to measure the hardness of a solid material (Johnson, 1985), but his theory only applies to purely elastic materials (Adams & Nosonovsky, 2000). Because of that it is impossible to predict the indentation depth in the material beyond the elastic region with the theory of elasticity.

When a small particle-shaped indenter is encapsulated between two plates pressed against each other the stress distribution is close to hydrostatic in the particle and the elastic deformation of the indenter can be described with Hooke’s law in three dimensions (Heyden, et al., 2008).

To calculate the elastic deformation of the indenter, Navier's formula first needs to be defined, which is an equilibrium equation,

$$\sigma = \frac{F}{A} \quad (36)$$

where F is the force in kN and A is the area in mm.

The radial strain in a spherical indenter as a function of stress is,

$$\varepsilon_r = \frac{1}{E} [\sigma_r - \nu(\sigma_y + \sigma_z)] \quad (37)$$

where ν is the Poisson's ratio and σ_r , σ_y and σ_z are stresses in different directions. Because of the hydrostatic stress state the strain ε_r is a constant.

Finally, the reduction of the diameter of the indenter is obtained by inserting Equation (36) into Equation (37) and then multiplying with the original diameter of the indenter.

$$\Delta D = D\varepsilon_r \quad (38)$$

Even if the stress exceeds the uniaxial yield strength, the indenter still remains elastic due to the hydrostatic stress state (Odenö & Klarbring, 1984).

6.4.2 Plastic Deformation

Plastic deformation is the permanent deformation that remains after unloading of a body. To deform a material plastically the flow stress of the material must be exceeded (Singh & Dwivedi, 2014). Plastic deformation can also, in the same way as elastic deformation, either be time independent or time dependent. Time dependent deformation also is known as creep, while time independent deformation is "regular" plasticity where time does not affect the deformation. Elastic deformation is superimposed when the material is undergoing plastic deformation. The total deformation of a body is the sum of its elastic and plastic deformations. Upon unloading the elastic deformation reverses but the plastic deformation remains (Rösler, et al., 2007).

Plastic deformation at indentations.

The plastic deformation in a plate due to a permanent indenter mark is difficult to describe analytically and therefore the topic is relatively unexplored (Choi, et al., 2008). The available knowledge of indentation plasticity comes mostly from finite element analysis (FEA), for rather shallow indentations ($h/R < 0.05$) (Kurapati, 2008).

6.5 Stress

6.5.1 Shear stress

If the indenter is not fully encased between the plates, it is not only subjected to homogeneous compression when the plates are subjected to shear force and therefore no hydrostatic stress state appear. If the plates are stiff enough to shear the indenter a value of the shear strength of the indenter therefore can be derived from the tensile strength according to Von Mises yield theory (Nam-Ho & Bhavani, 2009). The material yields when the Von Mises stress exceeds the uniaxial yield stress. Assume a situation where only shear stress exist ($\sigma_x=\sigma_y=0$ and $\tau_{xy}=\tau$). Then this is true for principal stresses, $\sigma_1=-\sigma_2=\tau$ and $\sigma_3=0$. Von Mises equation for principal stresses is defined as (Nam-Ho & Bhavani, 2009).

$$f_y = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \quad (39)$$

Insert $\sigma_3=0$ in Equation (39) and simplify.

$$f_y = \sqrt{\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2} \quad (40)$$

Insert $\sigma_1=-\sigma_2$ in Equation (40) and simplify.

$$f_y^2 = 3\sigma_1^2 \quad (41)$$

Insert $\sigma_1=\tau$ in Equation (41) and derive the shear stress, τ , which now only depends on the yield stress.

$$\tau = \frac{f_y}{\sqrt{3}} \approx 0.577f_y \quad (42)$$

6.6 Indenter Materials

6.6.1 Aluminium Oxide

The 2.5 mm diameter sphere of aluminium oxide, Al_2O_3 , can be used as a ceramic indenter. Aluminium oxide is a very hard material, with the hardness number 1356 Vickers (HRC ~70), but also brittle with low compressive strength in uni- and biaxial loading. Besides, aluminium oxide is a very temperature resistant material. The indenters used in this work were provided by Aliaxis Utilities & Industry in Stockholm, Sweden.

Figure 26 Aluminium oxide indenter. Source: Authors picture.

6.6.2 Stainless Steel

The material in the 2.5 mm diameter stainless steel spherical indenter used in this work is the high strength steel X46Cr13 according to Eurocode nomenclature (Swedish Standards Institute, 2005), also called AISI 420C by the American Society for Testing and Materials (ASTM). The ultimate tensile strength is 1700–1900 MPa and the hardness HRC 52–60. This corrosion resistant steel is a very ductile material, as the brittle fracture mechanism cleavage does not exist in this material. Young's modulus is 205 GPa and Poisson's ratio 0.235. The indenters were provided by Specuma, Stockholm, Sweden.

Figure 27 Stainless steel indenter. Source: Authors picture.

6.6.3 Tungsten Carbide

The 2.5 mm diameter tungsten carbide spherical indenter is also a high strength indenter, like the stainless steel indenter. The ISO-standard grade of the indenter material is K20. The ultimate compressive strength is considerable, around 4600–5800 MPa and a hardness HRC 75–78, which is greater than for aluminium oxide. The tungsten carbide indenter is rather expensive in comparison with the stainless steel indenter. Young's modulus is 650 GPa and Poisson's ratio 0.22. The indenters were also provided by the company Specuma in Stockholm, Sweden.

Figure 28 Tungsten carbide indenter. Source: Authors picture.

7 Experimental Testing

7.1 First Test Set

At the outset of the work, aluminium oxide was selected as candidate indenter material. In a preliminary test, a ceramic ball of aluminium oxide was compressed in a hand-held plier tool. The ball was crushed at a certain load and the failure was brittle. This feature shows that hardness is not the sole important parameter when selecting a suitable indenter material. After further market search, as alternatives both stainless steel and tungsten carbide indenters were acquired from Specuma.

The purpose of the first test was to investigate whether the spherical indenters inserted between two steel plates is capable of penetrating the plates when pressed together and to determine the force necessary to press an indenter a certain distance into the steel plate. The test was conducted such that the indenters did not interfere with each other.

The plates were made of cold formed structural steel grade S235JRC+C. Ten plates, each of dimensions 100x100x10 mm, are used for the test. The steel grade was selected considering availability and ordering time, since the time allotted to a thesis is rather limited. The original aim was to use steel grade S355 which is most common in existing older composite I-profile steel bridges (de Matteis, et al., 2010), but steel grade S235 may still indicate the behaviour and show if the concept of using indenters is viable. As indenters 50 spherical particles made of stainless steel and 50 made of tungsten carbide were used. Both material types have a dimension of 2.5 mm in diameter, but different material properties. The ceramic indenter was not used in this test because of the result of the plier pre-test. However, the curiosity of knowing whether particles of aluminium oxide in fact would increase the effective friction coefficient, even after crushing brought the aluminium oxide particles back to the second test, in the belief that the crushed material might behave like a coated abrasive.

The indenters were fixed to a plate with a thin double-sided adhesive tape, the effect of which can be disregarded from in calculations because its compressive strength is negligible. A pair of plates with indenters in between was compressed in an electro-hydraulic 600 kN Dartec testing machine, Figure 29. During a test the cross-head displacement increased linearly with time. Depth of indentation and further compression was registered with three circumferentially equidistant displacement gauges attached to sides of the circular end blocks of the loading device. Cross-head displacement, depth of indentation and the corresponding compressive force, registered during a test, were stored in a computer using the in-house software Catman. Drawings of the test specimens in the first test are presented in Appendix D.

Figure 29 Dartec hydraulic rig and the second test with one tungsten carbide indenter. Source: Authors picture.

The first test set comprised five different individual tests sorted in three categories a) No. 1, a reference test in which just two plates were compressed, b) No. 2 and 3, tests with one single indenter of tungsten carbide and of stainless steel, respectively, and c) No. 4 and 5, tests with 49 indenters of the two materials, respectively. The position of the indenters is shown in Appendix E.

The purpose of the reference test was to obtain a reference stiffness of the testing machine and the plates themselves, indicating the limit stiffness at full indentation for plates with indenters. Note that the limit stiffness is ideally independent of the number of indenters. The reference stiffness is simply the slope of the reference test force-displacement curve. After testing, Brinell and Rockwell hardness of just one of the plates used was measured, since all plates were taken from the same batch, to compare the hardness of indenters, given in HRC and the plates. The result of the hardness tests, compared with the indenters is shown in Table 14.

The Brinell hardness test is performed with an indenter 2.5 mm in diameter applied under 30 seconds with a load of 187.5 kg and an indenter 10 mm in diameter applied under 30 seconds with a load of 3000 kg. The Rockwell hardness test is performed accordingly to the C scale with a conical diamond indenter with a spherical tip.

The ultimate tensile strength of the plates and the elastic deformations of the indenters are also calculated and shown in the results of the first test set.

Table 10 Conditions of the first test set.

Test	Number of indenters	Material of indenter	Velocity [mm/s]
1	0	–	0.00083
2	1	Tungsten carbide	0.00083
3	1	Stainless steel	0.0083
4	49	Tungsten carbide	0.0083
5	49	Stainless steel	0.0083

7.2 Results of the First Test Set

The real-time progression of a test was visualized by keeping the software Catman running to show certain selected test parameters graphically on a computer screen during the test. For each displacement the sensors are detecting, a corresponding load from the rig at a certain time is displayed. All test results were also stored in Excel document format to allow further analysis using suitable software.

The load measured together with a mean value calculated with the data from the three sensors a load-deformation diagram is produced with the results from all five tests, with the load [kN] on the y-axis and the deformation [mm] on the x-axis. The x-axis of the diagram is chosen according to the size of the indenter showing the deflection of the indenter between minus 2.5 mm and 0 mm, thereafter the deformation of the plates up to 0.25 mm. At a displacement of minus 2.5 mm the indenter has not penetrate the plates yet. At the displacement 0 mm the indenter is fully impressed into the plates. After 0 mm the deformation of the plates pressed together is visualized. In this way both the reference test and the other four tests could be visualized in the same diagram.

Figure 30 Load-Deformation diagram with all five tests. Source: Authors picture.

By dividing the force in test four and in test five by the number of indenters, the result of all tests in terms of force per indenter can be compared directly against each other. In Figure 31 the tests with just one indenter and with 49 indenters almost coincide for a gap between the plates from 2.5 mm and down to 0.25 mm. Thereafter the inclination of the force-deformation curve increases significantly for the tests with one indenter in comparison with the tests with 49 indenters. It is also seen that the test with 49 stainless steel indenters the gap does not fully close and that the tests with 49 indenters do not completely align with the reference line.

Figure 31 Load-Deformation diagram with all five tests, where test 4 & 5 are divided with 49 indenters. Source: Authors picture.

The indentation markings are visible on the plates after testing. The depth of the markings can be compared with the depth from the diagrams, to be able to ensure that the results are indicating one and the same thing. Photos of the indentation from all tests are shown in Appendix F.

Figure 32 Test 5 with 49 stainless steel indenters after compression. Source: Authors picture.

The force needed to impress one indenter is taken from test five. The reason is that tests four and five with 49 indenters are thought to be more accurate in load determination when the two plates are close to each other than are tests two and three with just one indenter each. This point will be discussed further in the conclusions. Also the choice of stainless steel as an indenter for determining the force needed to impress one indenter will be explained later. Since there are no data for a full compression of test five (i.e. at zero gap between the plates), an assumption is made that an indentation of 2.30 mm is good enough to be considered as fully effective in a joint subjected to shear force. Why the assumption of 2.30 mm is used will be explained in the conclusions as well. If an indentation around 2.30 mm, corresponding to a gap of 0.20 mm between the plates, is a correct assumption for a stainless steel indenter, a mean value is taken from some points near that value. By studying the curve of test five in Figure 31, some points around 2.30 mm can be chosen for the calculation of the mean value.

Table 11 Mean value of force needed to compress 49 indenters 2.30 mm.

Point	Indentation [mm]	Force [kN]
1	2.314	522.9
2	2.321	524.8
3	2.294	518.9
4	2.296	519.3
5	2.284	517.3
6	2.301	520.4
Mean value	2.302	520.6

$$F_{p,C,Rd} = \frac{F_{c,49}}{49} \quad (43)$$

$$F_{p,C,Rd} = 10.62 \text{ kN} \approx 11 \text{ kN}$$

The theoretical stress value can be calculated and then plotted against the deformation. In this way it is possible to investigate whether the theoretical modulus of elasticity for steel is reached. The modulus of elasticity for structural steel has a theoretical value of 210 GPa (Åstedt, 2009). For calculating the theoretical stress value, Hooke's law is applied,

$$\sigma = E\varepsilon \quad (44)$$

Engineering strain, ε , is defined as the change in length divided with the original length.

$$\varepsilon = \frac{\Delta L}{L_0} \quad (45)$$

By combining the two Equations (44) and (45) the theoretical stress is given,

$$\sigma = \frac{E}{L_0} \Delta L \quad (46)$$

The original length, L_0 , is known and constant for the two reference plates.

$L_0 = 20 \text{ mm}$

By trying different values for the modulus of elasticity after plotting the theoretical stresses against the deformation the modulus of elasticity was found.

$E = 6.5 \text{ GPa}$

Figure 33 Reference line and the theoretical line. Source: Authors picture.

Another way to determine the modulus of elasticity is to numerically calculate it by comparing two points from the reference line. The original length of the two plates is still known and constant (20 mm). One point is fixed at the maximum measured value, and two other points are then compared to the fixed value. The two other points are taken from the end of the reference line but not at the very end where the fixed value is. The equation of Young's modulus is defined as,

$$E = \frac{\Delta\sigma}{\Delta\varepsilon} \quad (47)$$

Insert Equation (45) in Equation (47).

$$E = \frac{\sigma_2 - \sigma_1}{\frac{\Delta L_2 - \Delta L_1}{L_0}} \quad (48)$$

Table 12 Numerical calculated modulus of elasticity, two different comparison with one fixed point.

	σ_1 [MPa]	ΔL_1 [mm]	σ_2 [MPa]	ΔL_2 [mm]	E [MPa]
First comparison	21.026785714286	0.10228087377860	25.01488095238	0.11465963511984	6443
Second comparison	23.000992063492	0.10850040558178	25.01488095238	0.11465963511984	6539

The Rockwell hardness of a plate was measured three times. The individual results and their mean value are shown in Table 13.

Table 13 Results from the Rockwell hardness test for the S235JRC+C plate.

Attempt	Rockwell (HRC) result
1	11.9
2	11.0
3	11.5
Mean value	11.5

The Brinell hardness test with a 2.5 mm diameter indenter was performed five times with identical results. The diameter of the resulting indenter impression was 1.07 mm. For the second Brinell hardness test with a 10 mm diameter indenter, the test was performed twice. The results were identical in the two attempts and the resulting impression diameter was 4.4 mm. By using Equation (33) a Brinell hardness value (BHN) was calculated from the applied load and the measured impression diameters. In Table 14 the hardness of the different materials is shown together with the Rockwell hardness of the plate and of the indenter materials, for comparison.

Table 14 Hardness test result for the S235JRC+C plate and comparison with the indenters.

Material	Brinell 2.5 mm	Brinell 10 mm	Rockwell HRC
Tungsten carbide	-	-	77.0-79.0
Stainless steel	-	-	52-60
S235JRC+C	198	187	11.5

For the Brinell hardness 187, which is the lowest value of the two Brinell hardness tests made, an ultimate tensile strength of 645 MPa is calculated according to Equation (34) for the material used. It can be compared with the ultimate tensile strength for a steel plate with steel grade S235 according to Eurocode which is 360 MPa (Swedish Standards Institute, 2005).

The compression of the spherical indenters, as expressed by the reduction of the diameter of an indenter, assuming linear elastic behaviour and embedment in a hydrostatic stress field, have been calculated using Equation (36)-(38) for the stainless steel and tungsten indenters. The compressive stress σ is calculated as the normal stress in the equatorial plane of an indenter with Equation (37) where the load is $F_{p,C,Rd}=11$ kN, the compressive load required to close the gap between the plates, see Figure 31, and $A=\pi d^2/4$, the equatorial area of an indenter, where d is the diameter of the indenter. As the stress field is assumed hydrostatic, all stress components in Equation (37) equals σ . Material parameters and diameter reduction for the two indenter types are shown in Table 15.

Table 15 Elastic deformation of the indenters loaded with 11 kN.

Elastic deformation	Stainless steel	Tungsten carbide
Young's modulus, E [GPa]	205	650
Poisson's ratio, ν [-]	0.235	0.22
Indenter area, A [mm ²]	4.91	4.91
Stress, σ [MPa]	2241	2241
Strain, ϵ_r [-]	5.79e-3	1.93e-3
Change in diameter, ΔD [mm]	0.01448	0.00483

7.3 Second Test Set

A lap joint as shown in Figure 34, by 2 aligned and partly overlapping plates of equal width and thickness, joined by one pre-loaded centred bolt and with equal number of indenters in both friction planes and loaded longitudinally.

Prior to the second test set, a preparatory test was conducted in order to determine the tightening moment corresponding to the pre-loading bolt force to be applied in the second test. To determine this moment, a load cell connected to a computer is compressed by tightening a 220 mm long M30 class 10.9 bolt, using an adjustable torque wrench. The torque wrench is set to a certain moment and then tightened with hand power. Thereafter the moment was compared with the obtained pre-loading force shown on the computer screen. When the prescribed pre-loading force is reached, the corresponding torque wrench moment is noted. The prescribed pre-loading force is 320 kN, corresponding to a tightening moment of 1.62 kNm, in all seven tests.

At the beginning of the work it was expected that the aluminium oxide indenter would penetrate the plates instead of crushing, and later on indenters of other materials were proven more promising. The main purpose of the second test is to determine whether the effective friction coefficient (μ) of a friction connection can be improved by indenters and thereby operate as an enhanced friction connection. The purpose is also to investigate the efficiency of an indenter as a function of the distance from the hole or bolt centre. The aluminium oxide was not used in the first test set but was included in the second test set to find out to what extent the crushed material could improve the effective friction coefficient by acting as a coated abrasive rather than a spherical indenter.

Cold formed structural steel with steel grade S235JRC+C is used as plate material in tests No. 2 to No. 7. This is the same steel grade as in the previous tests in the first test set and necessary material data are therefore known. Because of the rare plate dimension needed in test No. 1, it was not possible within reasonable time to acquire the same cold formed structural steel of grade S235JRC+C as in the first test set. Instead plates of grade S355J2+N was delivered and used in test No. 1. Hardness testing showed that the given S235 and S355 have similar hardness value, and the difference should therefore not affect the results.

Two quadratic plates, both with the dimensions 200x200x20 mm of steel grade S355J2+N are used in test No. 1. Eighteen plates, each with the dimensions 400x100x20 mm, and six quadratic plates, each with the dimensions 100x100x20 mm, are used for the remaining six tests. In addition to the different plates, six M30 class 10.9 bolts and accompanying nuts and washers are used. Two plastic cylinders are used in the tightening process in order to centre the bolt and in that way avoid a shear- or a bearing failure. As indenters 120 spherical pellets made of stainless steel and 48 made of aluminium oxide were used. The pellet diameter of both material types is 2.5 mm, but the material properties are different, in particular strength. The indenters were fixed to a plate with a thin double-sided adhesive tape, the effect of which can be disregarded from in calculations because its compressive strength is negligible.

Two pair of knife edges were welded to the plates, one pair on each side of the connection. To register the relationship between the slip force and the slip of the joint, a Crack Opening Displacement (COD) clip gauge (measuring the slip) is mounted to the edges of each pair of knives and then connected to the testing machine computer. A clip-gauge mounted to one pair of knives is seen in Figure 34.

The testing machine computer registered and stored data from the tests. In the testing machine a joint was loaded longitudinally under in time linearly increasing displacement until slip failure occurred. The drawings of the test specimens in the second tests are shown in Appendix D.

Figure 34 Pre-loading of the first test and the test setup for the remaining tests. Source: Authors picture.

In the second test set, comprising seven individual tests, the first test (No. 1), a compressive test, with 24 steel indenters, forming a cross pattern between the two S355J2+N plates, was pre-loaded with an M30 bolt. The indentation diameter of the impressions was then measured in a light microscope, in order to explore whether indenter penetration varied appreciably with distance from the centre of the joint. Decreasing indenter penetration might be thought of as indicating decreasing “efficiency” of an indenter.

After the first test a reference joint test was performed with fresh plates without any indenters for comparison of force-slip behaviour to the following tests with various indenter arrangements.

Thereafter five tests were made with different number and arrangement of indenters of either stainless steel or aluminium oxide. Indenter position in the five tests are shown in Appendix E. After all tests, Brinell (10 mm indenter) and Rockwell (HRC) hardness of two plates of different steel grade, was measured in the same way as in the first test set.

Table 16 Conditions of the second test set.

Test	Total number of indenter	Material of indenter	Velocity [mm/s]
1	24	Stainless steel	–
2	–	–	0.0083
3	16	Aluminium oxide	0.0083
4	16	Stainless steel	0.0083
5	32	Stainless steel	0.0083
6	48	Stainless steel	0.0083
7	32	Aluminium oxide	0.0083

7.4 Results of the Second Test Set

The theoretical tightening moment for the desired pre-loading force in the pre-test is calculated in order to allow comparison with the actual required moment for the pre-test. According to Equation (23) the theoretical moment is 1689 Nm, while the actual pre-test moment is 1619 Nm. The values used for calculation of the theoretical moment are shown in Table 17.

Table 17 List of values used in the moment calculation above.

M30 bolt with washers	
$F_{p,C}$	320 kN
P	3.5 mm
d_0	27.73 mm
D_k	43.5 mm
$\mu_g = \mu_u$	0.125 (Arvid Nilsson AB, 2007)

For each slip detected by the two COD clip gauges a corresponding load from the rig is displayed. This force-slip relationship for all tests are shown in Figure 36. Photos of indentations in all tests are shown in Appendix F.

The measured indentation diameters in the first test are plotted versus the distance of the indenter from the bolt centre. The indentation diameter of the first indenter, at a distance 31 mm from the bolt centre, is 2.6 mm, while that of the outermost indenter, at a distance of 91 mm from the same centre is 2.4 mm. The relationship between indentation diameter and distance from bolt centre are shown in Figure 35. Photos of the impressions of all indenters are shown in Appendix G.

Figure 35 Indentation diameter versus distance from bolt centre. Source: Authors picture.

The efficiency of the outermost indenter (in relation to the first) is calculated as the ratio of the first and outermost indentation diameters.

$$Efficiency = \frac{2.4}{2.6} = 0.92 \quad (49)$$

The ratio of the distance to the outermost indenter and the bolt radius is,

$$n_{bolt,r} = \frac{91}{15} \approx 6 \text{ pcs} \quad (50)$$

and describes the distance of the efficiency in terms of number of bolt hole radius.

The theoretical slip force is calculated for the reference test in order to compare it with the experimentally obtained value. By using Equation (27), with $\gamma_{M3}=1$, the slip force is 108.8 kN. In the Equation $k_s=0.85$ and $\mu=0.20$ are assumed.

The measured slip force together with a mean value calculated by the data from the two COD clip gauges, a slip force-slip diagram is produced with the results from all steel indenters and the reference line. With the slip force [kN] in the y-axis and the slip [mm] in the x-axis.

Figure 36 Slip resistance for respective connection with steel indenters. Source: Authors picture.

Since Eurocode has a requirement that two plates are not allowed to slip more than 0.15 mm relative to each other, it is easier to compare all tests when the value of 0.15 mm is zoomed in. The reference line indicates a constant declining slip force after the initial slip, while the tests with indenters has an increasing slip force. The more indenters inserted in the connection the more slip resistance is given.

For the connection with 16 indenters, 8 per shear plane, the slip resistance at 0.15 mm slip is 105.9 kN. For the connection with 32 indenters, 16 per shear plane, the slip resistance at 0.15 mm slip is 162.6 kN. For the connection with 48 indenters, 24 per shear plane, the slip resistance at 0.15 mm slip is 217.0 kN. This can be compared with the connection without any indenters where the slip resistance at 0.15 mm slip is 54.5 kN.

Figure 37 Slip resistance at 0.15 mm slip for respective connection with steel indenters. Source: Authors picture.

In Figure 38 the slip force, at the acceptable slip of 0.15 mm, versus the total number of indenters is shown. Thereafter a prediction of the slip force is made for the number of indenters possible to impress with the pre-loading force 320 kN used in the tests. All predictions were conducted by a second-degree polynomial. The maximum number of indenters with a pre-loading force of 320 kN is 58 indenters. The prediction indicates that the slip force can reach just above 250 kN.

Figure 38 Slip force per steel indenter at the acceptable slip 0.15 mm. Source: Authors picture.

In Figure 39 the slip force, at the slip 0.50 mm, versus the total number of indenters is shown. The prediction indicates that the slip force can reach just above 310 kN, with 58 indenters.

Figure 39 Slip force per steel indenter without the slip criterion. Source: Authors picture.

By using Equation (27), the effective friction coefficient (μ) can be calculated by taking k_s and γ_{M3} equal to unit, Equation (51). The bolt hole coefficient (k_s) and the partial safety factor (γ_{M3}) should not be included when calculating μ as these can be considered as safety factors in dimensioning.

$$\mu = \frac{F_{s,Rd}}{F_{p,C}n} \quad (51)$$

With the friction coefficient calculated for respective slip force a diagram can be produced where the effective friction coefficient versus number of indenters according to Equation (51) is shown in Figure 40. Thereafter a prediction of the friction coefficient is made for the number of indenters possible to impress with the pre-loading force 320 kN used in the tests. In the reference test where zero indenters are used the friction coefficient is $\mu=0.09$. For the connection with 16 indenters, 8 per shear plane, where the slip is 0.15 mm and the slip resistance is 105.9 kN. The friction coefficient is $\mu=0.17$. For the connection with 32 indenters, 16 per shear plane, where the slip is 0.15 mm and the slip resistance is 162.6 kN. The friction coefficient is $\mu=0.25$. For the connection with 48 indenters, 24 per shear plane, where the slip is 0.15 mm and the slip resistance is 217.0 kN. The friction coefficient is $\mu=0.34$. The prediction indicates that the friction coefficient can reach just under $\mu=0.40$, with 58 indenters and a pre-loading force of 320 kN.

Figure 40 Friction coefficient per steel indenter, with the pre-loading force 320 kN. Source: Authors picture.

If the Eurocode requirement, that two plates are not allowed to slip more than 0.15 mm relative to each other, is disregarded from, higher values of the friction coefficient may be obtained since the slip force for connections with indenters increase beyond the Eurocode boundary, as shown in Figure 36 and Figure 37.

If a slip up to 0.50 mm is allowed, then through extrapolation of the curve in Figure 41 an effective friction coefficient of almost 0.50 is obtained for the pre-loading force 320 kN and the maximum number of indenters that can be fully impressed by this force.

Figure 41 Friction coefficient per steel indenter, with the pre-loading force 320 kN and a slip up to 0.5 mm. Source: Authors picture.

In the connections with aluminium oxide indenters, the crushed material made smaller impression than metal indenters in the plates. An aluminium oxide indenter impression is illustrated in Figure 43.

In Figure 42 a diagram is shown slip force versus slip for the reference test and the aluminium oxide tests. For the connection with 16 indenters, 8 per shear plane, the slip resistance at 0.15 mm slip is 158.9 kN. For the connection with 32 indenters, 16 per shear plane, the slip resistance at 0.15 mm slip is 152.1 kN.

Figure 42 Slip resistance for respective connection with aluminium oxide indenters. Source: Authors picture.

Figure 43 Impression from test 2 with 16 aluminium oxide indenters. Source: Authors picture.

The friction coefficient is calculated with Equation (51) in the same way as for the steel indenters. In Figure 44 the effective friction coefficient versus number of indenters is shown.

For the connections with 16 indenters, 8 per shear plane, where the slip is 0.15 mm and the slip resistance is 158.9 kN. The friction coefficient is $\mu=0.25$. For the connection with 32 indenters, 16 per shear plane, where the slip is 0.15 mm and the slip resistance is 152.1 kN. The friction coefficient is $\mu=0.24$. This result is strangely, since 16 indenters gives a better effective friction coefficient than 32 indenters.

The peculiar behaviour of aluminium oxide indenters needs further investigation and seems not to invite extrapolations.

Figure 44 Friction coefficient per aluminium oxide indenter, with the pre-loading force 320 kN. Source: Authors picture.

7.5 Complementary Hardness Testing

In order to get a more accurate result of the hardness testing, a number of the Rockwell hardness test (HRC) was performed and a mean value was calculated.

Table 18 Results from the Rockwell hardness test for the S235JRC+C plate and the S355J2+N plate.

Test No.	Rockwell (HRC) result S235JRC+C	Rockwell (HRC) result S355J2+N
1	14.6	9.5
2	13.0	8.5
3	10.9	9.8
4	13.7	10.2
5	9.79	-
Mean value	12.4	9.5

For the Brinell hardness test with a 10 mm indenter, the test was performed three times. The result were identical for the three attempts with the S235JRC+C plate and the resulting diameter of the impression was 4.6 mm. For the S355J2+N plate, the diameter of the impression was 4.5 mm for two of the attempts, while the diameter of the impression was 4.6 mm for the last attempt. A mean value was then calculated for the diameter of the impression. By using Equation (33) a Brinell hardness value (BHN) of the measured diameters can be calculated. In Table 19 the hardness of the different materials are shown.

Table 19 Hardness test result for the S235JRC+C plate, the S355J2+N plate and a comparison with the indenters.

Material	Brinell 10 mm	Rockwell HRC
Stainless steel	-	52-60
Aluminium oxide	-	~70
S235JRC+C	170	12.4
S355J2+N	176	9.5

For the Brinell hardness number 170 and 176, an ultimate tensile strength of 587 MPa and 607 MPa are calculated according to Equation (34). It can be compared with the ultimate tensile strength for a steel plate with steel grade S235 and S355 according to Eurocode which is 360 MPa and 510 MPa (Swedish Standards Institute, 2005).

8 Discussion and Conclusions

In the first test set the results shown that to impress the 49 indenters of test No. 4 and 5 fully a compressive force greater than $49 \times 11 = 540$ kN is required, but a force of this magnitude is beyond the testing machine capacity. If the rig had had a greater capacity, all the lines in Figure 31 would have approached the reference line. Figure 33 shows that the modulus of elasticity needed to get the same inclination for the theoretical line as the reference line is 6.5 GPa. This indicates that the plates without any indenters are still elastic and not fully compressed together in order to reach the right value of 210 GPa for steel. In Table 12 it can also be seen that the closer the value is taken relative the fixed end point, the higher modulus of elasticity is given. This also indicates that the plates are still elastic and not fully compressed yet. The reference line can however be seen as the searched value needed in order to consider the plates fully compressed.

In Figure 31 it is seen that the force–deformation curves for 49 indenters divided by 49 and for one indenter almost coincide. This agreement means that the distance between neighbouring indenters is sufficient for the indenters to act independently of each other. Both stainless steel and tungsten carbide indenters were used in the first test set and produced similar results. Tungsten carbide indenters are however much more expensive and for several reasons only stainless steel indenters were used in the remaining tests.

The calculations of the elastic deformation of respective indenter shows that tungsten carbide has a smaller deformation than stainless steel. This results in a deeper indentation for the tungsten carbide indenter, which partly can be seen in Figure 31. But since the difference in impression between the two kinds of indenters in Figure 31 is greater than the difference in the elastic deformations calculated, the difference also depends of something else. It may be a small plastic deformation that could not be seen with the eyes. The reason for the smaller deformation derives from the larger modulus of elasticity in tungsten carbide.

When two plates with one indenter in between are compressed, there is just one single point of pressure on each plate. The pressure point is situated at the centre of the plate, and on loading the two plates tend to cup slightly around the indenter. At a certain load the plates are themselves brought in contact with each other. The area of the contact surface increase with further compression and the required load therefore increases more rapidly with the applied displacement than before contact. This happens when the measured gap between the plates is less than 0.25 mm. In the test with 49 indenters the plates are not in such contact with each other and the load does not increase more rapidly than before.

It is assumed that the force needed to impress one indenter (of many) is best given by test No. 5, where the indenters are distributed over the whole plate area and the plates do not interact with each other. When the two plates, with indenters in between, are compressed, a crater develops around each impression. The crater material is pressed up around the edge of the craters. The elevated material on different plate tend to meet when the measured plate gap is smaller than around 0.2 mm. That is why the force needed to impress one indenter is calculated from when the two elevated edges interact. Since the assumption is that the indenters are fully impressed when the plate material comes in contact, which they do at a 0.2 mm gap. The calculated value for when one indenter is impressed is used in the second test set.

The irregularities of the load-deformation curve for test No. 2 in Figure 31 between the plate gap 2 mm and 1 mm are associated with the deformation velocity, which was only 10 % of that in tests No. 3 to 5. When the indenter settle into the plates, a rapid load reduction is needed to maintain a constant deformation velocity. In test 3 to 5 the deformation velocity was sufficient in order to maintain a linearly increasing deformation, without any visible load reductions in the diagram.

Indentation diameter varies very little with distance from the bolt centre. Thus penetration can be assumed to vary little, and so also “indenter efficiency” at various distances from the bolt.

According to the tests with stainless steel indenters in the second test set, the slip resistance force and the effective friction coefficient at an acceptable slip of 0.15 mm increases linearly with the number of indenters, see Figure 38 and Figure 40. Without any slip criterion the slip resistance force and the effective friction coefficient increases more rapidly with the number of indenters, see Figure 39 and Figure 41. For an assumed number of 58 stainless steel indenters and a pre-loading force of 320 kN, several predictions can be made. The slip resistance can reach 250 kN at 0.15 mm slip while it may reach 310 kN at 0.50 mm slip. The prediction also indicates that the friction coefficient with the slip criterion can reach just under $\mu=0.40$, and $\mu=0.50$ without the slip criterion. For the aluminium oxide connection with 16 indenters, 8 per shear plane, no prediction was made. However the slip resistance is 158.9 kN and the friction coefficient is $\mu=0.25$ with the slip criterion.

The slip resistance force of the reference test is 54.5 kN and the friction coefficient is $\mu=0.085$. In comparison, the theoretical value of the slip resistance force for the reference test, with the friction coefficient $\mu=0.20$ for a rolled surface (Swedish Standards Institute, 2011) is 109 kN. This considerable difference raises the question: What is the definition of a rolled surface concerning the reliability of the Eurocode friction coefficients in the present case and others, and in the present case in particular, whether the surface condition “rolled surface” might simply be too wide.

Figure 36 shows that friction force at fully developed slip decreases slowly with increasing slip from the initial slip force. It is noted that the slip force at 0.15 mm slip is in fact somewhat lower than the initial slip force and slowly decreasing with further slip. For a joint characterized by a decreasing slip force, an (accidental) overload entails inevitably unbounded further slip, or, in other terms, such a joint provides no safety against overloading. In contrast, the friction force of joints with indenters increases monotonically with slip. A joint with indenters thus provides safety against overloading as an overload would simply be met by increased resistance and a small slip increment.

To conclude whether indenters have a positive effect and can contribute to an enhanced friction connection the friction coefficients obtained are compared with those given in Eurocode, see Table 6 for different surface treatments. Since connections with oversized holes were used in this work the Eurocode coefficients should be multiplied with the factor $k_s=0.85$. It is now possible to compare the greatest value in Eurocode $\mu=0.43$ (for surfaces blasted with shot or grit with loose rust removed, not pitted) and the greatest value in the second test set $\mu=0.40$. The comparison shows that the connection with indenters is almost as effective as a

surface blasted with shot or grit with loose rust removed, not pitted. But the connection with the indenters gives a safety against overloading and the effective friction coefficient is certain for a given amount of indenters, while the Eurocode definition of the surface condition is perhaps too wide. If the slip criterion is disregarded from the friction coefficient increases with 16 % in comparison with Eurocode.

The results from the Rockwell hardness test is uncertain since the measured values are outside of the recommended interval (HRC 20–70). But it is still possible to understand the huge difference between the plates and the indenters. The stainless steel indenter had a hardness number HRC 52–60, while the value of the plates are around HRC 10. The Brinell hardness tests give a more consistent result for the same sample, when using structural steel. The first discovery made for structural steel was that the measured hardness, and in that way the calculated ultimate tensile strength, varies a lot for the same steel grade. It is also discovered that a steel plate of lower steel grade can have a higher ultimate tensile strength than a steel plate with higher steel grade. This might be an effect of that the lower steel grade were cold formed (which increases the strength), while the higher steel grade were not. Even if the ultimate tensile strength varies, it is well above the minimum requirements.

The calculated ultimate tensile strength for the plates with steel grade S235 are 587 MPa and 645 MPa, when it should be 360 MPa according to Eurocode. The calculated strength of the plate with steel grade S355 is 607 MPa, when it should be 510 MPa. This differences are great and may depend on much higher steel quality produced and delivered than ordered, since the factory only ensure a minimum steel grade and not a maximum. The differences may also depend on a poorly done approximation in Equation (34).

Calculations of the distribution of the proportions of the total bolt tightening moment attributed to the clamping force, to the thread friction and to the friction between bolt head and plate, is shown in Appendix C. The proportions [10 %, 40 %, 50 %] the theories refer to for example (Malmqvist, 2010), are proven valid only for friction coefficients around 0.14–0.20. Outside of this friction coefficient interval the first term varies greatly, at the expense of the other two terms.

Eurocode offers simplified tightening methods which provide approximate solutions where the friction coefficients are included in terms of k -classes. In the k -classes the relaxation also is included, but Eurocode however still mentions that if the torque method is used, a retightening after a couple of days is necessary. This sounds contradictory.

9 Reflections and Further Research

In this thesis the assumption that a 0.2 mm gap between the plates, is good enough to be able to define the indenter as impressed, was made. If a wider gap gives the same slip force, then the effective friction coefficient will increase since a lower pre-loading force is needed. Therefore it would be interesting to see an optimization of the gap in order to get the highest friction coefficient with the lowest pre-loading force. Even the size of the indenter should be optimized to give the most effective result.

The way the indenters were placed onto the plates and then secured with a double-sided tape was time consuming and tricky. It is therefore of practical interest to develop simple and reliable methods of application of indenters to a plate surface, for example using pre-manufactured fabrics of standardized sizes and shapes, with indenters arranged in a certain pattern, or optionally to be cut into the desired shape for a particular connection, or as in the example with the 3MTM Friction Shims case, a steel foil with steel indenters attached instead of a nickel diamond coating.

It would also be of interest to experimentally show the effects of the 3MTM Friction Shims method, as well as plates with different surface conditions, particularly for surfaces blasted with shot or grit with loose rust removed, not pitted.

Finally the questions raised earlier about the friction coefficient in Eurocode are relevant to ask, because the observed friction coefficient is so far apart from the prescribed friction coefficient! What is the definition of a rolled surface? Is the friction coefficient in Eurocode reliable? If not, are the other friction coefficients reliable? How can we be sure that a designer knows the Eurocode definition of a rolled surface? Is the defined surface conditions in Eurocode too wide? This has a serious consequence since the slip resistance only has half the resistance the designer believes in the reference test in the second test set. It is also interesting to investigate in which condition the surface of a structure steel is delivered and which routines the contractor has to ensure the right friction coefficient.

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Appendixes

Appendix A- Derivation of the tightening moment.

Derivation of the tightening moment, proceeding from Equation (12) to the finale Equation (13), based on theory from (Eriksson, et al., 1991) and approximations from (Broberg & Gustafsson, 1983).

$$\begin{aligned}
 M_{pi} &= F_{p,c} \frac{P}{2\pi} \frac{1 + \mu_g \frac{1}{\cos \alpha \tan \varphi}}{1 - \mu_g \frac{\tan \varphi}{\cos \alpha}} = F_{p,c} \frac{P}{2\pi} \frac{1 + \mu_g \frac{1}{\frac{P}{\pi d_m \cos \alpha}}}{1 - \mu_g \frac{P}{\pi d_m \cos \alpha}} = \\
 &= F_{p,c} \frac{P}{2\pi} \frac{1 + \mu_g \frac{\pi d_m}{P \cos \alpha}}{1 - \mu_g \frac{P}{\pi d_m \cos \alpha}} = F_{p,c} \frac{P}{2\pi} \frac{\frac{P \cos \alpha + \mu_g \pi d_m}{P \cos \alpha}}{\frac{\pi d_m \cos \alpha - \mu_g P}{\pi d_m \cos \alpha}} = \\
 &= F_{p,c} \frac{P}{2\pi} \frac{(P \cos \alpha + \mu_g \pi d_m) \pi d_m \cos \alpha}{(\pi d_m \cos \alpha - \mu_g P) P \cos \alpha} = F_{p,c} \frac{(P \cos \alpha + \mu_g \pi d_m) d_m}{2(\pi d_m \cos \alpha - \mu_g P)} = \\
 &= F_{p,c} \frac{P d_m \cos \alpha + \mu_g \pi d_m^2}{2\pi d_m \cos \alpha - 2\mu_g P} = F_{p,c} \frac{(P d_m \cos \alpha + \mu_g \pi d_m^2)(2\pi d_m \cos \alpha + 2\mu_g P)}{(2\pi d_m \cos \alpha - 2\mu_g P)(2\pi d_m \cos \alpha + 2\mu_g P)} = \\
 &= F_{p,c} \frac{P d_m^2 2\pi \cos^2 \alpha + 2P^2 d_m \mu_g \cos \alpha + \mu_g 2\pi^2 d_m^3 \cos \alpha + 2P \mu_g^2 \pi d_m^2}{4\pi^2 d_m^2 \cos^2 \alpha - 4P^2 \mu_g^2} =
 \end{aligned}$$

For ISO-bolts neglect all μ^2 -terms. This will give a deviation less than 5 % for normal friction coefficients (Broberg & Gustafsson, 1983).

$$\begin{aligned}
 &= F_{p,c} \frac{P d_m^2 2\pi \cos^2 \alpha + 2P^2 d_m \mu_g \cos \alpha + \mu_g 2\pi^2 d_m^3 \cos \alpha}{4\pi^2 d_m^2 \cos^2 \alpha} = \\
 &= F_{p,c} \left(\frac{P d_m^2 2\pi \cos^2 \alpha}{4\pi^2 d_m^2 \cos^2 \alpha} + \frac{2P^2 d_m \mu_g \cos \alpha}{4\pi^2 d_m^2 \cos^2 \alpha} + \frac{\mu_g 2\pi^2 d_m^3 \cos \alpha}{4\pi^2 d_m^2 \cos^2 \alpha} \right) = \\
 &= F_{p,c} \left(\frac{P}{2\pi} + \frac{P^2 \mu_g}{2\pi^2 d_m \cos \alpha} + \frac{\mu_g d_m}{2 \cos \alpha} \right) = F_{p,c} \left(\frac{P}{2\pi} + \frac{P \mu_g \tan \varphi}{2\pi \cos \alpha} + \frac{\mu_g d_m}{2 \cos \alpha} \right) =
 \end{aligned}$$

Approximate $\tan \varphi \rightarrow 0$ for small angles of φ . For ISO-bolts the pitch angle is $\varphi=2.48^\circ$.

$$= F_{p,c} \left(\frac{P}{2\pi} + \frac{\mu_g d_m}{2 \cos \alpha} \right)$$

Appendix B- Deviation from the approximation that $\tan \rho \tan \varphi$ is less than 1.

A visualization of the deviations from the approximation that $(\tan \rho \tan \varphi)$ can be neglected because $\tan \rho \tan \varphi \ll 1$.

Necessary bolt data from (Colly Company AB, 1995). All bolts are ISO-bolts MC6S with the thread angle $\alpha=30^\circ$.

Bolt Size	d_2 [mm]	P [mm]
M12	10.86	1.75
M14	12.70	2.00
M16	14.70	2.00
M18	16.38	2.50
M20	18.38	2.50
M22	20.38	2.50
M24	22.05	3.00
M27	25.05	3.00
M30	27.73	3.50
M33	30.73	3.50
M36	33.40	4.00

$$\tan \rho = \frac{\mu_g}{\cos \alpha}$$

$$\tan \varphi = \frac{P}{\pi d_2}$$

To calculate how much the deviation is for respective bolt and friction coefficient the difference between the answers in the denominator is calculated. The answer in the denominator is either 1 or <1 .

$$M_g = F_{p,c} \frac{d_m}{2} \frac{\frac{\mu_g}{\cos \alpha} + \frac{P}{\pi d_m}}{1 - \tan \rho \tan \varphi}$$

$$\frac{\text{approximated value} - \text{correct value}}{\text{correct value}} = \frac{1 - (1 - \tan \rho \tan \varphi)}{(1 - \tan \rho \tan \varphi)} = \frac{\tan \rho \tan \varphi}{1 - \tan \rho \tan \varphi} =$$

$$= \frac{\frac{\mu_g}{\cos \alpha} \frac{P}{\pi d_m}}{1 - \frac{\mu_g}{\cos \alpha} \frac{P}{\pi d_m}} = \frac{\frac{2P\mu_g}{\sqrt{3}\pi d_m}}{1 - \frac{2P\mu_g}{\sqrt{3}\pi d_m}}$$

The deviation in % for different bolt sizes and friction coefficients are tabulated below.

μ_g	M12	M14	M16	M18	M20	M22	M24	M27	M30	M33	M36
0.1	0.6	0.6	0.5	0.6	0.5	0.5	0.5	0.4	0.5	0.4	0.4
0.2	1.2	1.2	1.0	1.1	1.0	0.9	1.0	0.9	0.9	0.8	0.9
0.3	1.8	1.8	1.5	1.7	1.5	1.4	1.5	1.3	1.4	1.3	1.3
0.4	2.4	2.4	2.0	2.3	2.0	1.8	2.0	1.8	1.9	1.7	1.8
0.5	3.1	3.0	2.6	2.9	2.6	2.3	2.6	2.3	2.4	2.1	2.3
0.6	3.7	3.6	3.1	3.5	3.1	2.8	3.1	2.7	2.9	2.6	2.7
0.7	4.3	4.2	3.6	4.1	3.6	3.3	3.6	3.2	3.4	3.0	3.2
0.8	5.0	4.9	4.2	4.7	4.2	3.7	4.2	3.6	3.9	3.5	3.6
0.9	5.6	5.5	4.7	5.3	4.7	4.2	4.7	4.1	4.4	3.9	4.1
1.0	6.3	6.1	5.3	5.9	5.3	4.7	5.3	4.6	4.9	4.4	4.6

Appendix C- Derivation of the p-term.

The equation for the tightening moment.

$$M_{pi} = M_g + M_u = F_{p,c} \left(\frac{P}{2\pi} + \frac{\mu_g d_m}{2 \cos \alpha} + \frac{\mu_u D_k}{2} \right) =$$

$$= F_{p,c} (0.159P + 0.577d_m\mu_g + 0.5D_k\mu_u)$$

Dimensions for different bolts.

Bolt Size	d _m [mm]	P [mm]	D _k [mm]
M12	10.86	1.75	15.75
M14	12.70	2.00	18.25
M16	14.70	2.00	20.75
M18	16.38	2.50	23.50
M20	18.38	2.50	26.00
M22	20.38	2.50	28.50
M24	22.05	3.00	31.00
M27	25.05	3.00	35.00
M30	27.73	3.50	39.00
M33	30.73	3.50	43.00
M36	33.40	4.00	46.50

Friction coefficients between different surfaces (Arvid Nilsson AB, 2007).

Ytbehandling		Smörjningstill- stånd	μ _{tot} μ _g μ _u
Skruv	Mutter		
Obehandlad	Obehandlad	Torr	0,14
		Olja	0,125
		MoS2	0,1
		Vax	0,06
Fosfaterad	Fosfaterad eller obehandlad	Torr	0,125
		Olja	0,1
		MoS2	0,08
		Vax	0,06
Elförzinkad eller meka- niskt förzinkad	Elförzinkad eller meka- niskt förzinkad	Torr	0,14
		Olja/emulsion	0,1
		Vax	0,06
	Lättmetall	Olja/emulsion	0,125
Varm- förzinkad	Varm- förzinkad eller obehandlad	Torr	0,2
		Olja/emulsion	0,14
		Vax	0,06
	Lättmetall	Olja/emulsion	0,16
Rostfritt stål	Rostfritt stål eller lättmetall	Vax	0,14
		Olja/emulsion	0,2

By testing different combinations of bolt sizes and friction coefficients the proportions between the three terms can be derived by simply divide one term with the sum of all terms. In the tables below the value and the proportion between the terms is displayed with bolt sizes between M12 and M36 and friction coefficients between 0.06 and 0.20. Even friction coefficient 0.30 is displayed to show how small the first term can be.

$\mu_g = \mu_u$ 0.06				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	0.376	0.473	25	33	42
M14	0.318	0.440	0.548	24	34	42
M16	0.318	0.509	0.623	22	35	43
M18	0.398	0.567	0.705	24	34	42
M20	0.398	0.637	0.780	22	35	43
M22	0.398	0.706	0.855	20	36	44
M24	0.477	0.764	0.930	22	35	43
M27	0.477	0.868	1.050	20	36	44
M30	0.557	0.961	1.170	21	36	44
M33	0.557	1.065	1.290	19	37	44
M36	0.637	1.157	1.395	20	36	44

$\mu_g = \mu_u$ 0.08				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	0.502	0.630	20	36	45
M14	0.318	0.587	0.730	19	36	45
M16	0.318	0.679	0.830	17	37	45
M18	0.398	0.757	0.940	19	36	45
M20	0.398	0.849	1.040	17	37	45
M22	0.398	0.941	1.140	16	38	46
M24	0.477	1.018	1.240	17	37	45
M27	0.477	1.157	1.400	16	38	46
M30	0.557	1.281	1.560	16	38	46
M33	0.557	1.419	1.720	15	38	47
M36	0.637	1.543	1.860	16	38	46

$\mu_g = \mu_u$ 0.10				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	0.627	0.788	16	37	47
M14	0.318	0.733	0.913	16	37	46
M16	0.318	0.849	1.038	14	38	47
M18	0.398	0.946	1.175	16	38	47
M20	0.398	1.061	1.300	14	38	47
M22	0.398	1.177	1.425	13	39	48
M24	0.477	1.273	1.550	14	39	47
M27	0.477	1.446	1.750	13	39	48
M30	0.557	1.601	1.950	14	39	47
M33	0.557	1.774	2.150	12	40	48
M36	0.637	1.928	2.325	13	39	48

$\mu_g = \mu_u$ 0.12				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	0.752	0.945	14	38	48
M14	0.318	0.880	1.095	14	38	48
M16	0.318	1.018	1.245	12	39	48
M18	0.398	1.135	1.410	14	39	48
M20	0.398	1.273	1.560	12	39	48
M22	0.398	1.412	1.710	11	40	49
M24	0.477	1.528	1.860	12	40	48
M27	0.477	1.736	2.100	11	40	49
M30	0.557	1.921	2.340	12	40	49
M33	0.557	2.129	2.580	11	40	49
M36	0.637	2.314	2.790	11	40	49

$\mu_g = \mu_u$ 0.14				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	0.878	1.103	12	39	49
M14	0.318	1.027	1.278	12	39	49
M16	0.318	1.188	1.453	11	40	49
M18	0.398	1.324	1.645	12	39	49
M20	0.398	1.486	1.820	11	40	49
M22	0.398	1.647	1.995	10	41	49
M24	0.477	1.782	2.170	11	40	49
M27	0.477	2.025	2.450	10	41	49
M30	0.557	2.241	2.730	10	41	49
M33	0.557	2.484	3.010	9	41	50
M36	0.637	2.700	3.255	10	41	49

$\mu_g = \mu_u$ 0.16				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	1.003	1.260	11	39	50
M14	0.318	1.173	1.460	11	40	49
M16	0.318	1.358	1.660	10	41	50
M18	0.398	1.513	1.880	10	40	50
M20	0.398	1.698	2.080	10	41	50
M22	0.398	1.883	2.280	9	41	50
M24	0.477	2.037	2.480	10	41	50
M27	0.477	2.314	2.800	9	41	50
M30	0.557	2.562	3.120	9	41	50
M33	0.557	2.839	3.440	8	42	50
M36	0.637	3.085	3.720	9	41	50

$\mu_g = \mu_u$ 0.18				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	1.129	1.418	10	40	50
M14	0.318	1.320	1.643	10	40	50
M16	0.318	1.528	1.868	9	41	50
M18	0.398	1.702	2.115	9	40	50
M20	0.398	1.910	2.340	9	41	50
M22	0.398	2.118	2.565	8	42	50
M24	0.477	2.292	2.790	9	41	50
M27	0.477	2.603	3.150	8	42	51
M30	0.557	2.882	3.510	8	41	51
M33	0.557	3.194	3.870	7	42	51
M36	0.637	3.471	4.185	8	42	50

$\mu_g = \mu_u$ 0.20				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	1.254	1.575	9	40	51
M14	0.318	1.466	1.825	9	41	51
M16	0.318	1.697	2.075	8	41	51
M18	0.398	1.891	2.350	9	41	51
M20	0.398	2.122	2.600	8	41	51
M22	0.398	2.353	2.850	7	42	51
M24	0.477	2.546	3.100	8	42	51
M27	0.477	2.893	3.500	7	42	51
M30	0.557	3.202	3.900	7	42	51
M33	0.557	3.548	4.300	7	42	51
M36	0.637	3.857	4.650	7	42	51

$\mu_g = \mu_u$ 0.30				%proportions		
Bolt Size	P-term	d _m -term	D _k -term	P-term	d _m -term	D _k -term
M12	0.279	1.881	2.363	6	42	52
M14	0.318	2.200	2.738	6	42	52
M16	0.318	2.546	3.113	5	43	52
M18	0.398	2.837	3.525	6	42	52
M20	0.398	3.184	3.900	5	43	52
M22	0.398	3.530	4.275	5	43	52
M24	0.477	3.819	4.650	5	43	52
M27	0.477	4.339	5.250	5	43	52
M30	0.557	4.803	5.850	5	43	52
M33	0.557	5.323	6.450	5	43	52
M36	0.637	5.785	6.975	5	43	52

Appendix D- Drawings of the test specimens.

Appendix E- Positions for the indenters in the tests.

Appendix F- Indentation from all tests.

Appendix G– Indentation diameter for respective indenter in test No. 1 for the second test set.

The first digit indicates the row and the second the order of the indenter counted from the bolt centre.

1.1

1.2

1.3

1.4

1.5

1.6

2.1

2.2

2.3

2.4

2.5

2.6

3.1

3.2

3.3

3.4

3.5

3.6

4.1

4.2

4.3

4.4

4.5

4.6