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Determination of preload in High Strength Friction Grip bolts in Existing Structures

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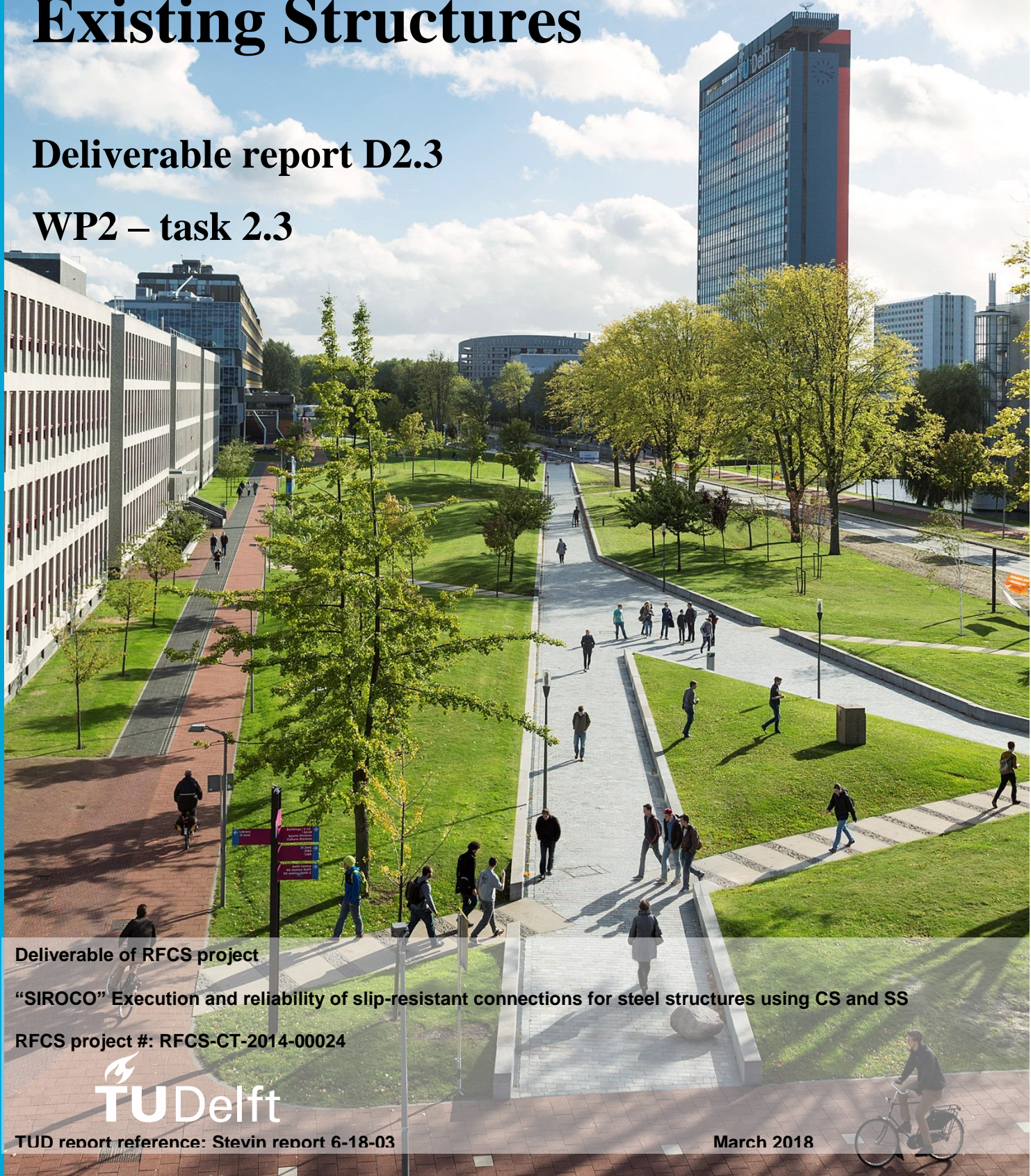
“SIROCO” Execution and reliability of slip-resistant connections for steel structures using CS and SS

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Abstract

A literature study was conducted to review the available methods to determine the actual preload in preloaded bolts. Based on the literature study and innovations in the field of installation of strain gauges suitable for bolts, experiments were carried out in the laboratory and in-situ to determine the actual preload level using the Strain Gauge Method. The experimental results were used to determine the reliability of the Strain Gauge Method as a predictor of preload level in bolts in existing structures.

Preface

The aim of the research in wp2.3 is to provide a guideline on how to determine the preload of HSFG bolts in connections in existing steel structures.

This report provides a State of the Art overview of methods that can be used to assess the preload in bolts. A practical method is developed that can be used to determine the 'actual' pretension in bolts in existing structures. Laboratory and in-situ experiments have been carried out to prove the concept of the method.

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List of Symbols

Symbol	Meaning	Unit
$F_{s,Rd}$	Design slip resistance	[F]
$F_{p,C}$	Nominal preload	[F]
A_s	Tensile area of the bolt	[L ²]
T	Torque	[FL]
P	Pitch of the threads	[L]
D_0	Outer diameter of the nut	[L]
E	Young's modulus	[F/ L ²]
V	Coefficient of variation	[-]
$V_{in/out}$	Input/output voltage	[V]
L_{eff}	Effective length	[L]
GF	Gage factor	[-]
R	Resistance	[Ω]
CF	Calibration factor	[F]
A_{net}	Net cross section area	[L ²]
Y	Electrical admittance	[1/Ω]
C	Acoustoelastic constant	[-]
k_s	Factor considering shape and size of the hole	[-]
f_{ub}	Ultimate tensile strength	[F/ L ²]
r_t	Effective contact radius of the threads	[L]
r_n	Effective radius of contact between nut and plate	[L]
d_h	Diameter of the bolt hole	[L]
d	Shank diameter	[L]
l_c	Clamping length	[L]
v	Ultrasound velocity	[L/T]
v_0	Ultrasound velocity (zero stress)	[L/T]
f_n	Resonance frequency	[1/T]
d_{nom}	Nominal diameter	[L]
d_{min}	Minimum nominal diameter	[L]
d_{max}	Maximum nominal diameter	[L]
μ	Slip factor	[-]
γ_M	Partial safety factor	[-]
μ_t	Coefficient of friction between nut and bolt threads	[-]
β	Half angle of the threads	[rad]
μ_n	Coefficient of friction between nut and plate	[-]
ε	Strain	[-]
σ	Stress	[F/ L ²]
ΔL	Elongation	[L]

1 Introduction

Preloaded bolts are used in connections in which the relative displacements of members must be prevented, as well as in connections subject to load reversals or vibrations. The behavior of such connections depends on the magnitude of the preload. A preload lower than the minimum required preload will lead to excessive deformations or fatigue failure of the connection, whereas a too high preload may cause bolt fracture. Given that preload decreases in time, e.g. as a result of coating creep, it is necessary knowing the preload of the bolts at any point during the service life of a connection in order to judge if the connection still fulfills its functional requirements.

The purpose of this work is to investigate the feasibility of determining the actual preload in preloaded bolted connections. Firstly, a theoretical overview of the available methods for determination of pretension level is made. Based on the theoretical considerations, experiments are carried out to determine the practical feasibility of the Strain Gauge Method using strain gauges and a new type of adhesive. This is done by initially imitating in-situ conditions in the lab (preliminary tests) and then installing and calibrating the strain gauges in a connection of an existing structure. Based on the experimental results, an indication is given on the reliability of the Strain Gauge Method as a way to determine the preload in existing bolted connections.

2 State of the Art

2.1 Preloading of Bolts

Based on EN 1090-2 [1], High Strength Friction Grip (HSFG) bolts should be preloaded to (at least) 70% of the bolt tensile strength, inducing a preload of magnitude $F_{p,c}$ (Eq. 1) in the bolt

$$F_{p,c} = 0,7f_{ub}A_s \quad \text{Eq. 1}$$

With:

f_{ub}	Ultimate tensile strength of bolt material [F/L ²]
A_s	Tensile stress area [L ²]

The precision with which a selected preloaded level is achieved mainly depends on the tightening method selected. When a bolt is tightened, the nut is rotated with respect to the bolt, stretching the bolt and resulting in preload. There are several ways to control the rotation of the nut:

- Measuring the torque which is applied (Torque Method),
- Measuring the angle of rotation of the nut (Turn-of-the-Nut Method)
- Combination of the above (the Combined Method)

In the following sections the most common tightening methods are analyzed.

2.1.1 Torque Method

Since a bolt is designed to be tightened by twisting the nut with respect to the head, the most convenient way to do this is by applying a torque. This method is called torque control method and experience and theoretical analysis say that there is usually a linear relationship (i.e. Eq. 2) between the applied torque the preload developed in a fastener.

$$T = C \cdot F_{p,c} \quad \text{Eq. 2}$$

With:

C	Constant [-]
$F_{p,c}$	Bolt Preload [F]

A number of equations have been derived that attempt to define the constant C . The expression for T presented in Eq. 3 has been proposed by Motosh (1976) [2].

$$T = \left(\frac{P}{2 \cdot \pi} + \frac{\mu_t r_t}{\cos(\beta)} + \mu_r r_n \right) \cdot F_{p,c} \quad \text{Eq. 3}$$

With:

T	Torque [FL]
P	Thread Pitch [L]
μ_t	Friction coefficient of thread interface [-]
r_t	Effective thread contact radius [L]
β	Half thread angle [°]
μ_n	Friction coefficient of nut-plate interface [-]
r_n	Effective nut-plate contact radius [L]
$F_{p,c}$	Bolt Preload [F]

Eq. 3 shows that the input torque is resisted by three reaction torques. These are as follows, respectively:

- the reaction torque as a result of the inclination of the threads with respect to the bolt axis, only this part leads to preload within the bolt;
- the reaction torque as a result of friction between nut and bolt threads;
- the reaction torque as a result of friction between the face of the nut and the washer or plate.

These force reactions affect the amount of initial preload we get when we tighten a fastener. The most influencing factors are the friction components, e.g. under the assumption of an M20 bolt with coefficients of friction $\mu_t = 0,15 [-]$, $\mu_n = 0,15 [-]$ and a preload force $F_{p,c} = 1000 \text{ N}$, it is concluded that each 1000 N of tension produces:

- 0,40 Nm of bolt stretch reaction torque,
- 1,84 Nm of reaction torque from thread friction and
- 1,99 Nm of reaction torque from friction under the nut.

The total torque is 4,23 Nm. Therefore, as shown in Figure 1, only 9% of the input torque results in bolt tension, whereas 44% of the input torque is lost due to friction between the threads and 47% is lost due to friction between the nut and the plate or the washer.

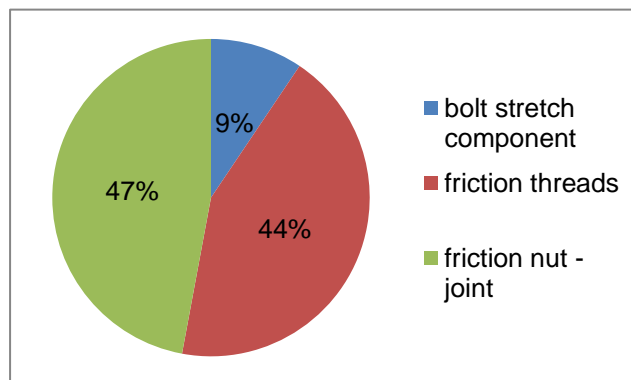


Figure 1 - Percentage of torque leading to bolt stretch and frictional force components

If it is assumed that the coefficient of friction between the nut and the joint is 10% greater than what was initially assumed $\mu_n = 0,165 [-]$, then the required input torque to overcome nut joint friction increases from 1,99 Nm to 2,19 Nm, 52% of the total input torque. So, in order to produce the same tension of 1000 N, the input torque has to increase by 5%. However, this is not possible because the operator has no way of telling that this set of parts is absorbing more torque between nut and joint. Furthermore, that extra 5% will not come from the thread friction component, unless the coefficient of friction at that point decreases somehow to offset the other increase. Therefore, the only "source" of that extra torque is a reduction of the bolt stretch component. The corresponding resisting torque will decrease from 9% to 4%, which results in loss of tension equal to 55%. Therefore, a variation of 10% in friction coefficient can cause a 55% loss in the resulting preload. Likewise, a 10% increase in friction between threads of bolt and nut will lead to a loss of pretension equal to 53%.

Generally, a variation of 10% in friction coefficient is very common. The coefficient of friction is very difficult to control and impossible to predict. There are almost 40 variables that affect the friction seen in a threaded fastener, e.g. the hardness of all parts, surface finishes, type of materials, speed with which the nut is tightened, fit between threads, presence or absence of washers, the temperature of the lubricants involved, etc. (Bickford, 1995) [3]. But, apart from friction coefficient, there are also some other variables that affect the torque - preload relationship. These factors are geometry, as shown in Eq. 3, the operator and tool accuracy and strain energy losses. A part of the input work may end up as strain energy losses due to bolt twist, a bent shank or nut deformation. In one extreme case, for example, if the threads gall and seize, the input torque produces just torsional strain and no preload at all. In that case the input energy does not result in heat due to friction losses but strain energy. Therefore, strain energy losses is another factor that affect the torque – preload relationship.

The influence of the abovementioned factors on the preload force is frequently grouped in a nut factor *k*. The nut factor *k* is an experimental constant which summarizes anything that affects the relationship between torque and preload, including friction, torsion, bending, plastic deformation of threads etc.. It is measured by experimentally applying a torque and measuring the achieved preload and it is defined in Eq. 4.

$$k = \frac{T}{F_{p,c} \cdot d} \tag{Eq. 4}$$

With:

<i>T</i>	Torque [FL]
<i>F_{p,c}</i>	Bolt Preload [F]
<i>d</i>	Bolt diameter [L]

Bickford determined the value of the nut factor for bolts in their as received state, without thread coating [3]. He did that based on the nut factors determined for a large number of such bolts with different dimensions and from different joints. The results are shown in Figure 2, from which it can be derived that the scatter in the *k*-factor is 25%. This scatter is caused by things like different friction conditions between the nuts and the joint surfaces, different hole clearances, operator’s errors, tool accuracy etc.

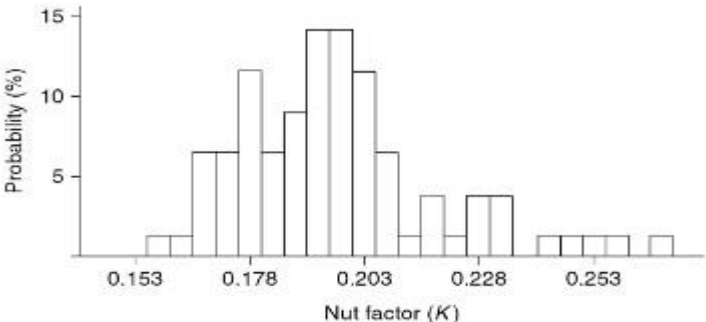


Figure 2 - Histogram of *k* values reported for as – received bolts from a large number of sources [2]

The uncertainty of frictional influences can cause a high scatter in the resulting preload. This means that in order to ensure that the minimum required preload will be achieved, a sufficiently high target preload should be chosen, considering the variation in k-factor. According to the report “Evaluation Tightening Preloaded Bolt Assemblies according to EN 1090-2” (Berenbak, 2012) [4] any target preload for a variation in the k-factor of 10%, results in a very high spread in the preload, leading to a high probability of overtightening or not achieving the minimum required preload. For a variation in the k-factor 6% and a target preload $0,8f_{ub}A_s$, the reliability of achieving the minimum required preload of $F_{p,C} = 0,7f_{ub}A_s$ is 94,8% and the reliability of surpassing a top value of $0,9f_{ub}A_s$ and overtightening the bolt is 5,2%, whereas the prescribed values according to EN 1090-2 [1] are 95% and 5%, respectively. In this case, the resulting spread with a 95% reliability is 25%, as shown in Fig. 2.4, leading to a low risk of overtightening a bolt.

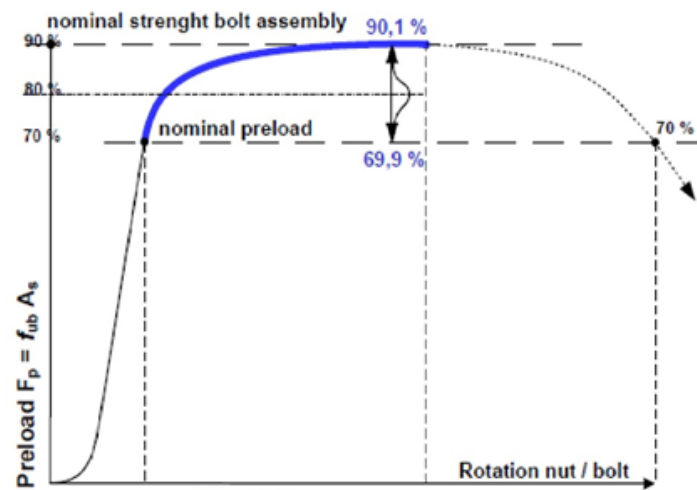


Figure 3 - Distribution of preloads levels using the torque method with a 6% k-factor variation [3]

2.1.2 Turn-of-the-nut Method

The second method is the turn of the nut method. This method is based on a predetermined rotation of the nut to achieve the desired preload force. Normally, a predetermined rotation results in a certain elongation of the bolt, which leads to the desired preload. However, there are some factors that may affect the turn – preload relationship.

The turn of the nut method starts by applying the first few turns of the nut which produce no preload at all, because the nut has not yet been run down against joint members and they are therefore not yet involved. This situation is shown in Figure 4a. When the nut starts to pull joint members together, some tension is produced in the bolt and this process is called snugging, as exemplified through Figure 4b. After the joint has been snugged, all bolts and joint members start to deform simultaneously. Preload now starts to build more rapidly in the bolt, following a straight line, as shown in Figure 4c.

In practice, the Turn-of-the-Nut Method is not as simple as it seems. The problem arises during the snugging process, when the tension in the bolt starts increasing. A high portion of that tension is absorbed by the joint and the bolt sees only a small increase in preload. However, the amount of that tension may vary from bolt to bolt and from joint to joint. This depends on several factors like bent washers or not perfectly flat joint members. If the plates

are not flat and parallel and the worker has not paid attention closing the gaps between the plates, then there is a high risk of not reaching the required preload level. Furthermore, the uncertainty in the rotation of the nut or any difference in the actual dimensions of the bolts may lead to a scatter in the resulting preload for a certain degree of rotation. The effects of plate straightness on the resulting achieved preload are shown in Figure 5.

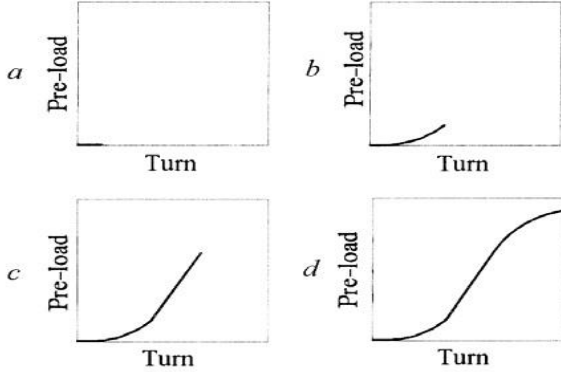


Figure 4 - Step-by-step build-up of preload using the Turn-of-the-Nut Method (Bickford, 1995) [2]

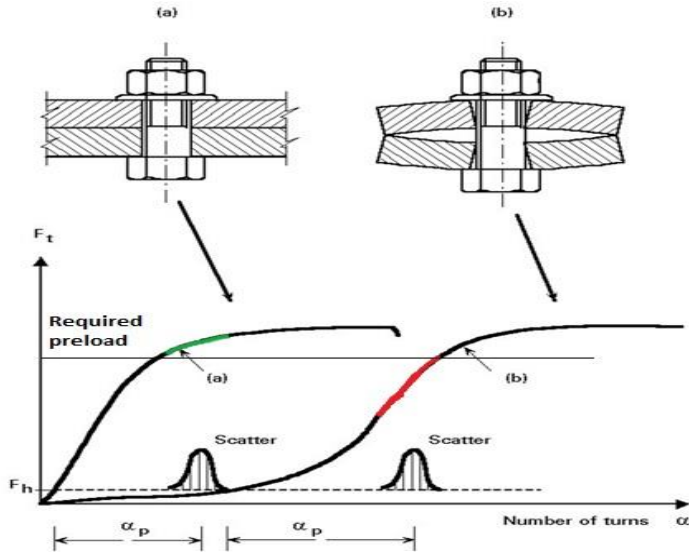


Figure 5 - Effect of parallel (left) and non-parallel (right) plates on the resulting preload (ESDEP) [4]

2.1.3 Combined Method

The Combined Method combines the Torque and Turn-of-the-Nut Methods. First, the plates are snugged by applying a torque of 75% needed to achieve the full preload. Secondly, the nut is then turned further by a predetermined angle which stretches the bolt past its yield point.

The Combined Method is the most advantageous compared to the other two. The first reason is that the torque used in the first step of the combined method is better in compensating for start-up variables, like the closing of the gaps between the plates, even for a high variation in the k factor. According to EN 14399-3 [5] the k-factor can vary between $0.10 \leq k \leq 0.16$. In order to determine the torque required to reach a preload of $75\%F_{p,C} = 0,53f_{ub}A_s$ in the first step, a k-factor equal to 0.13 may be assumed, according to EN 1090-2 [1], resulting in a torque $T = 0,13 \cdot d \cdot 0,53f_{ub}A_s$. In this case, if the actual k factor is 0.16, this torque would lead to a bottom preload level of $0,43f_{ub}A_s$ and if the actual k factor is 0.10, the resulting top preload level would be $0,68f_{ub}A_s$. Therefore, the resulting preload range is high enough to snug fit the plates and low enough in order to avoid overtightening of the bolt.

The second reason that the Combined Method is most advantageous, is that by rotating the nut by 90° (or more/less, depending on the clamping length) in the second step of the combined method, independently of the preload value achieved in the first step, the bolt is preloaded to a value somewhere on its plastic region. The reason is that the spread in the preload in the first step creates a small difference in the rotation on the elastic part of the preload – rotation curve. Therefore, a further rotation by 90° will take the bolt on its plastic region, as shown in Fig. 2.7.

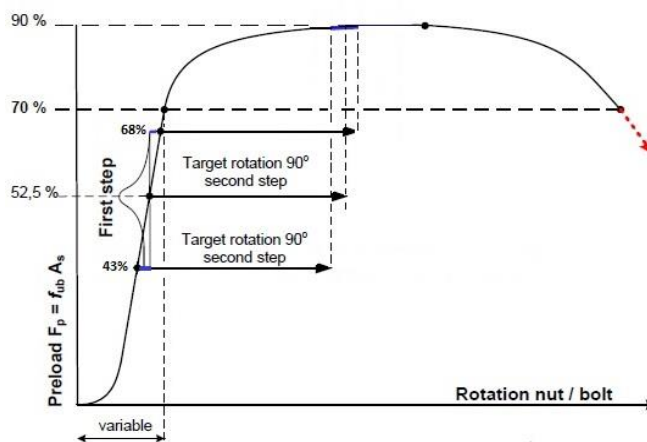


Figure 6 - Steps taken in the Combined Method to achieve (exceed) the required preload level [3]

2.2 Determination of Preload in Bolts

2.2.1 Ultrasound Method

In the Ultrasound Method, the elongation of a preloaded bolt is determined by measuring its length before and after the application of the preload. The instrument measures the Time Of Flight (TOF) of an ultrasound wave to travel from the head to the bottom of a bolt and back to the head, after its reflection on the bottom. Then, the ultrasonic length is obtained by multiplying the TOF by half the material velocity. The ultrasound wave is emitted by an ultrasound transducer which is placed on the head of the bolt, coupled with the bolt by a thin film of a viscous liquid, which permits transmission of ultrasound across the interface. The difference between the final ultrasonic length under load and the initial ultrasonic length under no-load condition, is referred to as the ultrasonic stretch ΔL , which is a measure for the bolt preload via Eq. 5.

$$F = \sigma \cdot A = E \cdot \varepsilon \cdot A = \frac{E \cdot A}{L_{eff}} \cdot \Delta L \tag{Eq. 5}$$

With:

E	Young's Modulus [F/L ²]
A	Bolt cross-sectional area [L ²]
L_{eff}	Bolt effective length [L]
ΔL	Ultrasonic Stretch [L]

Measuring the ultrasonic length or stretch of a bolt is not easy, as there are many factors that affect the measurement. These factors are analyzed in the following sections.

Stress

A bolt's ultrasonic length increases when the bolt is loaded under a tensile load. However, this is caused not only because of the elongation of the bolt, but also because the stress in the bolt causes a reduction of the velocity of the ultrasound wave. This is known as the acoustoelastic effect. Furthermore, the stress distribution along the axis of a bolt is not uniform, as shown in Figure 7, which means that areas of the bolt with higher stress, result in a higher reduction of the ultrasound wave velocity.

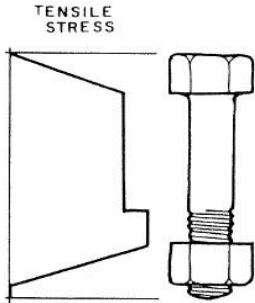


Figure 7 - Tensile stress along the axis of a bolt (Thanh et al., 2015) [5]

Generally, firms which provide the ultrasound equipment used for this method, offer the possibility of compensation for these stress related effects, by giving a procedure for the determination of a correction stress factor.

Residual stress

The stress caused in a bolt because of the external load is not the only stress that affects the ultrasound velocity. Residual stress also affects the ultrasound velocity. If the ultrasound wave experiences the same profile of residual stresses in the stressed and unstressed state of a bolt, their influence on the ultrasonic stretch is cancelled, since they are present in both the stressed and unstressed state of a bolt. However, if the transducer is placed on a different position on the head of the bolt, when (for example) the measurement in the stressed state is taken, the ultrasound wave will experience a different residual stress profile, resulting in an error in the force. However, it is not possible to know the residual stress profile that the ultrasound waves will experience in the bolt. Compressive or tensile residual stress, the direction of the residual stress with respect to the propagation of the wave in the bolt are some factors, which influence the estimation of the force. Egle & Bray (1976) [6] estimated the acoustoelastic coefficients for longitudinal waves propagating parallel or perpendicular to compressive or tensile stresses. In a bolt, the average residual stress level is low and significant residual stress concentrations exist at the threads, head to body fillet etc. The highest influence of residual stress on the velocity of the waves is when the stress (compressive or tensile) is parallel to the propagation of the longitudinal waves. It should be noted that the influence of residual stresses on the ultrasonic length is strongly characterized by their direction.

Temperature

Temperature has a pronounced effect on the ultrasonic stretch. The length of a bolt increases linearly with increasing temperature within certain limits and the velocity of an ultrasound wave decreases linearly with increasing temperature. If the temperature is the same for the measurement in the stressed and unstressed state, then the ultrasonic stretch is not influenced. However, if there is a temperature change between the two measurements, the ultrasonic stretch is affected. It has been estimated that for carbon steel, the velocity of a longitudinal ultrasound wave decreases linearly with increasing temperature with a rate $0,80 \frac{m/s}{^{\circ}C}$ (Heistermann, 2014) [7]. Based on a thermal expansion coefficient for carbon steel equal to $1,08 \cdot 10^{-5} /^{\circ}C$, the error in the ultrasonic stretch at a load level of 250 kN, for a temperature increase of 0.5 °C is 3%. Thus, even for a small temperature change the error in the ultrasonic stretch is already significant, which means that in case of in situ measurements, a continuous and accurate temperature measurement is required in order to compensate for temperature related effects. As mentioned for the stress influence, the ultrasound equipment offers the possibility of compensating for the temperature related effects.

Couplant

Another source of error in the determination of the ultrasonic stretch is the couplant which is added between the head of the bolt and the transducer. The couplant's thickness may vary, depending on the pressure on the transducer and the amount of couplant, resulting in error in the ultrasonic measurements. For example, an HV 10.9 M24 with $l_c = 80 \text{ mm}$ which is preloaded to its minimum required preload, is characterized by a stretch equal to 0,2746 mm. A variation in couplant's thickness equal to 12 μm can result in an error in the ultrasonic

stretch equal to 5%. Therefore, the couplant may have a quite high influence on the measurements.

Other factors that affect the measurements

So far, the error in the determination of the ultrasonic stretch is discussed. However, the purpose of the ultrasound tests is to determine the force and not the stretch. In order to do that, the model described by Eq. 5 is used. However, this requires an accurate determination of the effective length of the bolt, Young's modulus and the ultrasound wave velocity. And even if the uncertainty in Young's modulus value (5%) is considered acceptable, it is not possible to reach the same conclusion about the effective length. There are several models that describe the effective length of a bolt, but these models do not apply on all the bolt sizes, since the actual effective length of a bolt depends on factors like the clamping length and the length of the bolt. In practice, the uncertainty in the effective length of the bolt is high, therefore the resulting error in the force could be very high. All these uncertainties are dealt with by calibration.

Another important factor that may affect the measurements is the surface of the bolt. In order to emit and receive an ultrasound wave, the bolt must have a flat surface for the transducer to contact. The opposite end of the bolt should also have a parallel surface to reflect the ultrasound back to the transducer. Furthermore, a flat and smooth surface is very important to proper coupling of the transducer, otherwise the transducer may not achieve proper contact.

Finally, another point of interest is the accuracy of the measurement of the ultrasonic length. Since the change in the length between the stressed and unstressed state of the bolt is very low, it is required to measure the length to the nearest 0,001 mm (precision of 1 μm). In terms of TOF, this means that it is necessary to resolve $2 \cdot 0,001 \text{ mm} / 5890000 \text{ mm/s} = 3,4 \cdot 10^{-10} \text{ s}$. To achieve such precision, a high sampling rate of $1 \text{ sample} / (3,4 \cdot 10^{-10} \text{ s}) = 3 \text{ GHz}$ is required, which increases the cost of the measuring equipment.

Discussion

Based on the abovementioned, the conventional ultrasound method is sensitive to external influences such as temperature variations between the measurements or environmental noise. Furthermore, attention must be paid to the contact of the transducer with the head of the bolt, which requires flat and parallel surfaces. Moreover, the pressure applied on the transducer may cause variation in couplant's thickness and result in a different ultrasonic stretch measurement. The bolt stretch is sensitive to the location of the transducer on the head, when the measurements in the stressed and unstressed states are taken.

Compensation for temperature or stress related effects requires determination of a correction factor and in order to overcome the uncertainty in the effective length of the bolt or the ultrasound wave velocity, calibration is required. All these factors make the in-situ application of this method relatively difficult and time-consuming.

2.2.2 Velocity Ratio Method

As it is mentioned in the previous section, the conventional Ultrasound Method is based on measurements in the stressed and unstressed state of the bolt, which means that it is required to loosen the nut in order to obtain a measurement under zero force. In order to avoid loosening of the bolt, a new method was developed in the past, which relates the ratio of times of flight of a transverse and a longitudinal wave with the applied force (i.e. the bolt preload) via Eq. 6.

$$\frac{t_T^\sigma}{t_L^\sigma} = \frac{v_L^0}{v_T^0} \left[1 - \frac{L_{eff}}{L} (C_T - C_L) \cdot \frac{F}{A_{eff}} \right] \quad \text{Eq. 6}$$

With:

C_T	Acoustoelastic constant for transverse waves [-]
C_L	Acoustoelastic constant for longitudinal waves [-]
L_{eff}	Bolt effective length [L]
A_{eff}	Effective bolt cross-sectional area [L ²]
v_T^0	Transverse wave velocity (unstressed state) [L/T]
v_L^0	Longitudinal wave velocity (unstressed state) [L/T]
L	Bolt length [L]

In the following sections, the factors which affect the measurement of the ratio of TOFs will be analyzed.

Residual stress

The Velocity Ratio Method relates the ratio of TOFs with the difference in the acoustoelastic coefficients to determine the force on the bolt. Any stress, apart from the stress caused by the preload, results in an error in the ratio of TOFs. Therefore, in this section the influence of the residual stress will be examined.

In contrast to the conventional ultrasound method, in which the influence of the residual stresses is cancelled if the ultrasound wave experiences the same residual stress profile in the stressed and unstressed state of the bolt, the velocity ratio method has no way of cancelling the effect of the residual stress. Although it is expected that the influence of the residual stress in the ratio of TOFs will be limited, since both the nominator and the denominator are affected in the same way, the resulting error in the ratio of TOFs and the estimated force will depend on the different influence of the acoustoelastic effect on the longitudinal and transverse waves. The highest influence on the velocity of the waves is when the stress (compressive or tensile) is parallel to the propagation of the longitudinal and transverse waves.

Temperature

At the reference temperature of 21 °C the ultrasound velocity in steel is equal to $v_L^0 = 5890$ m/s and $v_T^0 = 3218$ m/s at a zero stress state. A change in temperature increases or decreases the ultrasound velocity. An error in the measured bolt force F occurs if calibration is performed at a certain temperature and the measurements of TOFs are done at a different temperature. The error is induced as a result of:

- A change in bolt length as a result of thermal expansion
- A change in ultrasound velocity as a result of temperature difference

In order to estimate the influence of temperature variations, the thermal expansion coefficient of carbon steel equal to $1,08 \cdot 10^{-5} 1/^{\circ}\text{C}$ is considered. According to literature [19], the velocity of longitudinal and transverse ultrasound waves decreases linearly with increasing temperature, with a rate of $0,80 \frac{\text{m/s}}{^{\circ}\text{C}}$ and $0,44 \frac{\text{m/s}}{^{\circ}\text{C}}$ respectively. The average error for different preload levels (between 100 – 300 kN) is 0.1% for a 10 °C temperature increase. The influence of temperature on the force, estimated by the Velocity Ratio Method is much lower compared to the conventional Ultrasound Method. This is reasonable, because the ratio of TOFs remains more or less constant, since both the nominator and denominator increase. The influence of temperature on the error is much smaller than the influence of the residual stresses, because the influence of temperature on longitudinal wave velocity is 1,8 times higher than that on the transverse wave velocity, whereas the influence of residual stresses on longitudinal wave velocity is almost 9,5 times higher than that on transverse wave velocity [6].

Another point of interest is when temperature variations take place between the longitudinal and transverse wave measurements. This can happen if measurements are not taken at the same time. Such a temperature change can have a quite big influence on the measurements. For example, an increase of just 0.5 °C when the longitudinal wave measurements are taken, can lead to an average error in the preload force equal to 1.8%. It is therefore important to carefully check and administer the temperature before any measurement is taken. Generally, it is possible to obtain measurements for simultaneous generation of longitudinal and transverse waves. Several researchers used transverse waves generated by electromagnetic acoustic transducers (EMAT). The transverse wave is obliquely incident to the bolt head and then it is transmitted in the bolt as a pair of longitudinal and transverse waves due to mode conversion (Ding et al., 2014) [8]. In that case, both waves experience the same temperature.

Influence of couplant

The couplant can also affect the ultrasonic measurements. Variations in couplant's thickness between the measurements or between calibration and periodic monitoring of a bolt, introduce a change in ultrasonic measurements.

Yasui & Kawashima (2000) [9] estimated the influence of couplant's thickness, using a transducer which excites and receives simultaneously longitudinal and shear waves, using a viscous couplant at the transducer – bolt head interface. He performed calibration tests on two different types of bolt. A short bolt with a length equal to 28 mm and a longer bolt with a length equal to 55 mm. At each load step he attached and removed the transducer 20 times. He found that the relative change of the ratio at a specific load step was 2.9% for a long bolt and 28.4% for a shorter one, which is reasonable because couplant thickness variations will have a bigger influence in case of a shorter time of flight. However, the error caused by the couplant's thickness can be diminished by using an EMAT transducer which needs no couplant when attached on bolt head (Ding et al., 2014) [8].

Other factors influencing the relationship between time ratios and preload

Determination of the force based on the model described by Eq. 6 requires not only an accurate measurement of the ratio of TOFs, but also an accurate estimation of constants like the acoustoelastic coefficients or the zero stress velocities. Furthermore, an accurate determination of the effective length is needed. Given that there is a high uncertainty in the

way that a bolt is stressed, the only way to deal with these uncertainties is to perform calibration.

Additionally, the requirement for parallel and smooth surfaces, which allow the reflection of the ultrasound waves, applies here too.

Discussion

As discussed, the main advantage of this method is that it does not require untightening of the bolt. However, in order to do that, it is necessary to know accurately many constants, which define the relationship between the ratio of TOFs and the force. Furthermore, this method has no way of cancelling the effect of residual stresses on the measurement of the ratio of TOFs, which means that even if all the constants are determined accurately, the resulting force is characterized by a certain degree of error. Therefore, calibration is necessary, if maximum accuracy is required, which means that untightening of the bolt cannot be avoided. But even in this case, there are several factors which may affect the measurements. Any change in the position of the transducer on the head of the bolt between calibration and periodic monitoring of a bolt, may lead to an error, because the waves will experience a different residual stress profile. The same applies for the influence of the couplant, as any variations in the thickness affect the resulting force. The influence of temperature variations is almost negligible, if there is no change in the temperature between the measurement of the TOF of the longitudinal and ultrasound wave. Finally, the requirement of a very accurate measurement of TOF makes the application of this method in situ challenging, as the environment interference can cause noise and distort the measurements.

2.2.3 Mechanical Resonance Frequency Shift Method

The Mechanical Resonance Frequency Shift Method is a method which measures the applied bolt force F based on the acoustoelastic effect. In the previous methods Time of Flights (TOFs) were measured, whereas in the current method the measuring device is sensitive to frequencies. A transducer is placed on the head of the bolt and generates a band of ultrasonic frequencies. Assuming that complete reflection occurs at the flat and parallel ends of the bolt, the transducer receives the reflected waves looking for that particular frequency which has resonated within the bolt. This will be that frequency which sees the length of the bolt as an exact multiple of its wavelength. The acoustic resonant frequencies are given by Eq. 7.

$$f_n = \frac{n \cdot v}{L} \quad \text{Eq. 7}$$

With:

n	Harmonic integer [-]
v	Acoustic wave velocity [L/s]
$L/2$	Bolt length [L]

When a stress is applied to the bolt, both L and v change, resulting in a change in the resonant frequency. Therefore, based on Eq. 7, the relationship between change in resonant frequency and load is expressed through Eq. 8

$$\Delta f_n = -f_n \cdot \left(C_L + \frac{1}{E} \right) \cdot \frac{F}{A} \quad \text{Eq. 8}$$

With:

f_n	Resonant Frequency [Hz]
C_L	Acoustoelastic constant for longitudinal waves [-]
E	Young's Modulus [F/L ²]
F	Bolt force [F]
A	Bolt area [L ²]

The theoretical resonant frequencies of an M24 bolt with $l_c = 80$ mm are in the order of 25 kHz. Since the order of the magnitude of the resonant frequencies is known, the transducer generates a range of frequencies of several kHz. The resonant frequency is recognized in the frequency response spectrum, because it is characterized by high amplitude (voltage) vibrations. Therefore, in contrast to the conventional ultrasound method and the velocity ratio method, which require a very accurate measurement of TOFs, in the Mechanical Resonance Frequency Shift method, it is sufficient that the sampling rate is such to reconstruct the input signal in a way that provides information about the amplitude of the vibrations. Based on the Nyquist frequency, this frequency can be over 100 kHz, on the order of some kHz or even MHz (the higher the sampling frequency, the better the resolution), which is lower than that of the other two acoustoelastic effect methods, but still quite high. In the following sections, the factors that affect the measurement of the frequency are presented.

Temperature

Temperature increase can cause thermal stresses, which result in an elongation of the bolt. Furthermore, a rise of the temperature results in a reduction of the ultrasound velocity. Both result in a reduction in the resonant frequency. If the temperature of the bolt is constant between stressed and unstressed readings, then no error is introduced in the preload force. However, if there is a change, the error depends on the temperature increase. As shown in

Fig. A4 small variations in temperature, like these that take place in the short time when a bolt is tightened, do not influence much the force. For example, a temperature increase of 0,5 °C in the stressed state will result in an error in the force equal to 1,3%. However, if maximum accuracy is required, it is important checking and administering the bolt temperature before a measurement is taken.

Joshi & Pathare (1984) [10] developed a method based on phase detection to track the frequency shift of the mechanical resonance of the bolt. A Voltage Controlled Oscillator (VCO) is used to emit signals of various frequencies and a phase detector compares the phase of the transmitted signal to that of the received signal. When their phase difference is zero resonance occurs and the VCO locks onto the resonant frequency. A temperature increase however causes a change in the phase of the signal. In that case a variable phase shifter is used which introduces a phase shift to the input signal in such a manner as to compensate for effects of temperature variations, provided that the relation between temperature variation and phase shift is known. It is therefore possible to keep the frequency constant even if the mechanical resonant frequency varies with the ambient temperature.

Residual stress

Residual stresses can affect the frequency and result in an error in the preload force. However, since the difference between the resonance frequencies in the stressed and unstressed state is measured, the influence of residual stresses will be the same for both measurements, provided that the transducer is placed on the same location on the head of the bolt on the stressed and unstressed state. Therefore, in that case it is possible to neglect their influence. However, if this is not the case, the position of the transducer on the head of the bolt plays an important role, because it can introduce an error caused by the difference in residual stresses experienced by the wave in the unstressed and stressed state.

Influence of couplant

The couplant used to bond the transducer on the head of the bolt can affect the measurements. Changes in couplant's thickness between the unstressed and stressed readings can introduce an error in the preload force. For example, a 12µm variation in couplant's thickness leads to an error in the force equal to 1,6%, which is lower than that in the conventional Ultrasound Method.

Other factors influencing the estimated force

As shown in Eq. 8, even if we measure accurately the resonance frequencies, it is required to determine the acoustoelastic constant and Young's Modulus. The uncertainties in these parameters can introduce a high error in the preload force. Therefore, in order to reduce these uncertainties calibration should be performed. Furthermore, it is required that the surfaces of the bolt are flat and parallel so that reflection of the ultrasound wave can occur.

Discussion

The Mechanical Resonance Shift Method is a method based on the acoustoelastic effect, which relates the difference in the resonance frequencies in the unstressed and stressed state with the applied stress on a bolt. Compared to the conventional Ultrasound Method (which is the other method which includes measurements in the stressed and unstressed state of the bolt), it is less affected by temperature variations between the measurements. Provided that the transducer is placed on the same position for the stressed and unstressed measurements, the Mechanical Resonance Shift Method allows measurement of the

frequencies by ignoring the influence of residual stresses. If not, the Mechanical Resonance Shift Method is less affected by residual stresses compared to the conventional ultrasound method. Flat and parallel surfaces are requirements for all the methods, whereas the influence of the couplant is lower for the Mechanical Resonance Shift Method. The required sampling rate is high, but lower compared to the other two methods, because it is easier working in the frequency domain than in the time domain. Finally, the instrument required for the application of this method, is not easy to be used in situ as a result of environmental influences.

2.2.4 Piezoelectric Active Sensing Method

All surfaces are rough to different degrees and as a result, the contact between surfaces is restricted to discrete areas at the tips of the surface asperities, as shown in Figure 8. In this respect, all bolted joints also develop partial contact at their imperfect interfaces. The applied torque on the bolt may change interfacial characteristics like true contact area. Once the interfacial characteristics changes are obtained, the tightness and thus preload of the bolted connections can be determined.

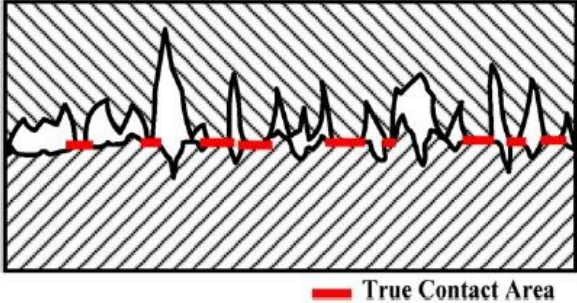


Figure 8 - Microscopic view of imperfect interfaces (Wang et al., 2013) [11]

The behavior of fastening the bolted joints can be regarded as imposing higher contact pressure at the imperfect interfaces. At these imperfect interfaces, a fraction of the nominal contact area, defined as a true contact area, is generated as shown in Figure 8.. The true contact area is known to be smaller than the nominal contact area. This true contact area varies with the contact pressure.

According to the sinusoidal surface model and the classic Hertz contact theory, the true contact area A_t and the contact pressure P have the relationship presented in Eq. 9 (Wang et al., 2013) [11].

$$A_t \propto C \cdot \sqrt{P} \tag{Eq. 9}$$

- With:
- A_t True contact area [L^2]
 - C Constant
 - P Contact pressure [F/L^2]

By increasing the fastening torque, the true contact area will increase and the wave will propagate across the interface with less energy loss. Two pieces of PZT (Lead Zirconate Titanate) are bonded to the different sides of the connection interface as an actuator and a sensor, as shown in Figure 9. PZT1, as an actuator, generates an ultrasonic wave that propagates across the interface, and then the signal is captured by sensor PZT2. Propagating waves are the means of power transmission. As illustrated in Figure 9, incoming waves are split into transmitted and lost waves at the microcontact interface. The transmitted wave energy is proportional to the true contact area, which is affected by the surface pressure generated by the bolt.

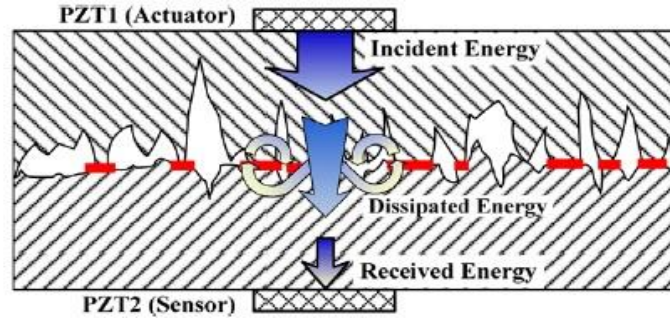


Figure 9 - Energy transmission at the micro-contact interface [9]

The active sensing method is based on the emission of a signal and measures the transient response of the structure with the aid of the PZT ceramics that enable actuating and sensing. For the received digital signal, after appropriate simplification, the signal energy E_s can be expressed in the discrete time domain $[t_s, t_f]$ through Eq. 10.

$$E_s = \frac{1}{f_s} \sum_{t_s}^{t_f} V^2(t) \quad \text{Eq. 10}$$

With:
 f_s Sampling Frequency $[1/T]$
 $V(t)$ Discrete sensor signal

Therefore, by measuring the signal voltage of the received wave, the tightening of the connection can be evaluated.

Factors that affect the measurements

The relationship between the applied force or pressure between the plates and the true contact area between the plates, which is proportional to the transmitted energy requires calibration to determine the relationship presented in Eq. 9. Hence, it is important to know what factors may change the measured energy.

The transmitted energy increases with the increase of clamping force, because of the increase of the true contact area. Therefore, anything that happens in the contact area between the plates affects the amount of the transmitted energy. When the bolts in a connection are tightened, some of the asperities that define contact between the plates are plastically deformed and the true contact area reaches a maximum, even if the preload force on the bolts is increasing. In that case, the transmitted energy remains constant. This was confirmed by Liu et al (2014) [12], who performed tests on four single bolted specimen with different dimensions and roughness by increasing the torque stepwise and measuring the transmitted energy. Experimental results for different fastening force levels are presented in Figure 10.

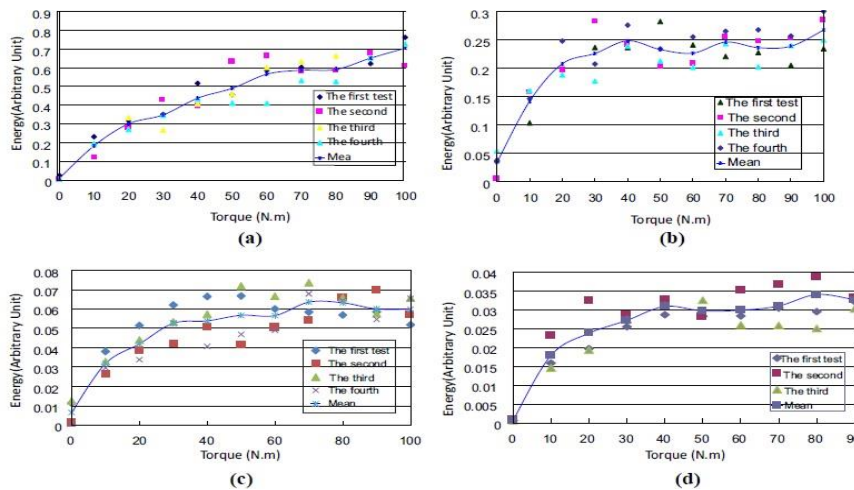


Figure 10 - Transmitted energy as a function of fastener torque (Liu et al., 2014) [10]

It is clear that with increase of the torque level, the received signal energy also increases. However, the energy level becomes constant at a certain torque level. This means that the maximum pressure below which the true contact area is proportional to the applied pressure is surpassed resulting in a maximum true contact area between the asperities and embedment of the plates during the tightening process. When this energy level is reached, the bolt is preloaded to a certain level resulting in a certain pressure between the plates. If saturation of the energy occurs at a preload level lower than the minimum required, then it is not possible to know if the bolt is preloaded over the minimum required preload. If saturation occurs at a preload level higher than the minimum required preload, then it is possible to know, based on calibration, that the bolt is preloaded at least to the force corresponding to the transmitted energy during saturation, which higher than the required preload level $F_{p,C}$. Therefore, it is possible to evaluate the preload condition of a bolt during the tightening procedure, but it has to be taken into account anything that happens in the contact area between the plates affects the measurements. An example of an influencing factor is the change in surface roughness, which could change the amount of transmitted energy [12]. Indeed, experimental results (Figure 10a and Figure 10b) whose only difference is surface roughness, show that there is a big difference in the transmitted energy values. The specimen belonging to the results presented in Figure 10b is characterized by a lower roughness than the specimen belonging to Figure 10a, and hence the transmitted energy is lower and saturation of the energy occurs at a lower torque level. Therefore, for plates with a specific roughness, it is possible to know how much transmitted energy leads to a certain preload. If in situ measurements are performed with this method and the influence of neighboring already tightened bolts on the measurements is taken into account, then it is possible to estimate the preload applied on a bolt during the tightening procedure.

However, this method is suitable for estimation of the preload only during the tightening process. In case that the purpose of the test is to evaluate the preload during the service life of the connection, given that the preload – transmitted energy (pressure) relationship is known, the embedment and creep which takes place at the plates as a result of tightening the bolt, will result in misleading conclusions, because the plastic (permanent) deformation of the asperities will not change the transmitted energy much, even if the pressure between the plates drops.

Discussion

The Piezoelectric Active Sensing Method is a method that evaluates the tightening of bolts by measuring the transmitted energy, which is proportional to pressure through the true contact area of the plates. The roughness of the plates, the distance from the neighboring bolts are some factors that affect the measurements. However, this method is suitable only for real time monitoring during the tightening process, as the embedment of the plates will affect the measurements obtained in case of periodic monitoring.

2.2.5 Strain Gauge Method

The most common method to measure the strain of a bolt is with a strain gauge. The strain gauge is glued in the shank of a bolt, as shown in Figure 12 and it consists of a very fine wire arranged in a grid pattern, as shown in Figure 11. The grid pattern maximizes the amount of metallic wire subject to strain in the parallel direction. The cross-sectional area of the grid is minimized to reduce the effect of shear strain and Poisson strain. The grid is bonded to a thin backing, called the carrier, which is attached directly in the shank of the bolt. Therefore, the strain experienced by the bolt is transferred directly to the strain gauge, which translates a linear change in strain with a linear change in electrical resistance. Strain gauges are available commercially with nominal resistance values from 30 to 3,000 Ω, with 120, 350, and 1,000 Ω being the most common values [13].

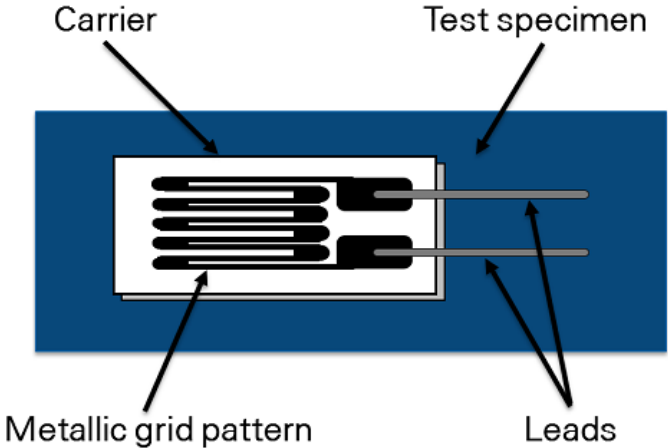


Figure 11 - BTM strain gauge [13]



Figure 12 - Strain gauge within bolt shank

If the relationship between the change in strain and the change in electrical resistance is known, it is possible to determine the strain applied on a bolt and hence also the bolt force. A fundamental parameter of the strain gauge is its sensitivity to strain, expressed quantitatively as the gage factor (GF). Gage factor is defined as the ratio of fractional change in electrical resistance to the fractional change in length (strain) [13], as expressed through Eq. 11.

$$GF = \frac{\frac{\Delta R}{R}}{\frac{\Delta L}{L}} = \frac{\Delta R}{R \cdot \epsilon} \tag{Eq. 11}$$

- With:
- ΔR Change in electrical resistance [Ω]
 - R Electrical resistance [Ω]
 - ϵ Strain in longitudinal direction [-]

In practice, strain measurements rarely involve quantities larger than a few millistrain ($\epsilon \cdot 10^{-3}$). Therefore, to measure the strain requires accurate measurement of very small changes in resistance. These very small changes in electrical resistance are measured by

using a Wheatstone bridge. As shown in Figure 13, a supply voltage is supplied across the bridge, which contains four resistors and the output voltage V_o is measured across the legs in the middle of the bridge. The output voltage is given by Eq. 12.

$$V_o = V_s \cdot \frac{R_3 R_1 - R_4 R_2}{(R_2 + R_3)(R_1 + R_4)} \tag{Eq. 12}$$

With:
 V_s Supply voltage [V]
 R_1, \dots, R_n Electrical resistance of the nth resistor [Ω]

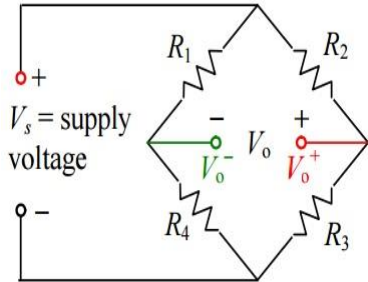


Figure 13 - Wheatstone bridge

When the strain is obtained using the strain gauge, the bolt force can be determined based on Eq. 13.

$$F_b = E \cdot A_{net} \cdot \epsilon \tag{Eq. 13}$$

With:
 E Young's Modulus of bolt [F/L^2]
 A_{net} Net cross sectional area of bolt [L^2]
 ϵ Measured strain [-]

In the following section, the factors that affect the measurements are discussed.

Factors that affect the measurements

Temperature variations are the most serious error source in the practice of strain measurement with a strain gauge. Provided that the strain gauge is installed successfully in the shank of the bolt, in addition to the desired measurement signal indicating strain, the strain gauge also produces a temperature – dependent measurement signal, which may result in an apparent strain or bolt force that does not exist. There are four effects that may cause this error:

- Thermal expansion of the bolt
- Temperature – dependent change in the strain gauge resistance
- Thermal expansion of the strain gauge measuring grid foil
- Temperature response of the connection wires

For a typical strain gauge attached to steel, the apparent strain as a function of the temperature is shown in Figure 14.

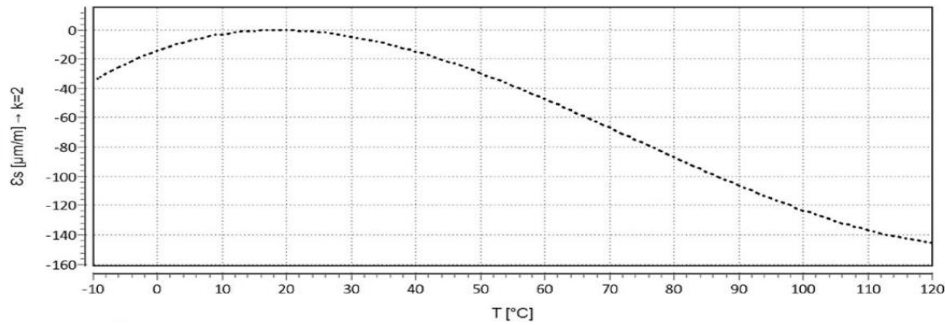


Figure 14 - Apparent strain in a strain gauge as a function of temperature (HBM) [14]

As shown in Figure 14, for temperature variations close to room temperature, the apparent strain is very low. For lower and higher temperature, the apparent strain increases, which results in an increase of the error in the strain.

However, there is a way to compensate for temperature related effects. As discussed before, the change in the electrical resistance is measured by a Wheatstone bridge. Based on Figure 13, when the electrical resistance of the four resistors is the same, the bridge is balanced and $V_o = 0\text{ V}$. However, when at least one of the electrical resistances changes, it results in $V_o \neq 0\text{ V}$ and the responsible resistor is called an active resistor. Depending on the type of the Wheatstone bridge configuration which is used, there are three types of strain gauge configurations: quarter bridge, half bridge and full bridge. Quarter bridge includes one active resistor/strain gauge, half bridge includes two active resistors and full bridge includes four active resistors. The temperature variations affect the electrical resistance of the resistors, which means that if a quarter bridge is used the electrical resistance of the one and only active resistor will change because of temperature variations, resulting in an error. However, when four active resistors are used in a full Wheatstone bridge configuration, all the resistors change their resistance in the same proportion, thus cancelling the effects of temperature change.

Bolt strain gauges series BTMC

Measuring the preload force of a bolt by using strain gauges is an old technique. However, the installation of a strain gauge was not always an easy process. In order to glue the strain gauge in a bolt, a specific type of adhesive was used. The hardening of this adhesive required the bolt to be subjected under a specific vacuum treatment, to ensure that no air was left inside the adhesive and afterwards, the bolt had to be given a heat treatment to allow for curing of the adhesive. Recently, in order to facilitate the installation of a strain gauge, a new type of adhesive was created, called CN adhesive, which can harden by taking up water from the environment. The amount of water required to cure the CN adhesive is included in the upper part of the strain gauge, making the installation procedure easy and fast.

Discussion

The strain gauge method offers the possibility of determination of the force in the bolt by measuring the strain in the shank. It requires a time consuming installation process, however the development of a new type of adhesive reduces the required process time. The strain gauge measurements are affected by temperature variations, but the use of a full

Wheatstone bridge is a simple way to diminish their influence. The feasibility of performing in situ measurements with the strain gauge method is to be further investigated in this document.

2.2.6 Measuring Bolt Preload in the Future: Permanent Mounted Transducer Systems (PMTS)

In order to determine the bolt preload as a function of time for newly built structures or for renovated HSFG connections, so called Permanent Mounted Transducer Systems (PMTS) can be used. The system consists of two parts:

- Permanently Mounted Transducers on the bolt head (Figure 15, left)
- Associated ultrasonic measuring system

PMTS also uses the Time-of-Flight (TOF) principle to determine the bolt preload, as illustrated through Figure 15 (right). Since the transducer is permanently mounted to the bolt head, both loaded and unloaded measurements are available for each bolt. This means that in fact all bolts are individually calibrated, minimizing the effects of for example residual stresses on the measured bolt force. An ultrasonic measurement in unloaded state is saved within the transducer, which ensures that all data belonging together is kept together. The readings have a relatively small error of $\pm 3\%$.

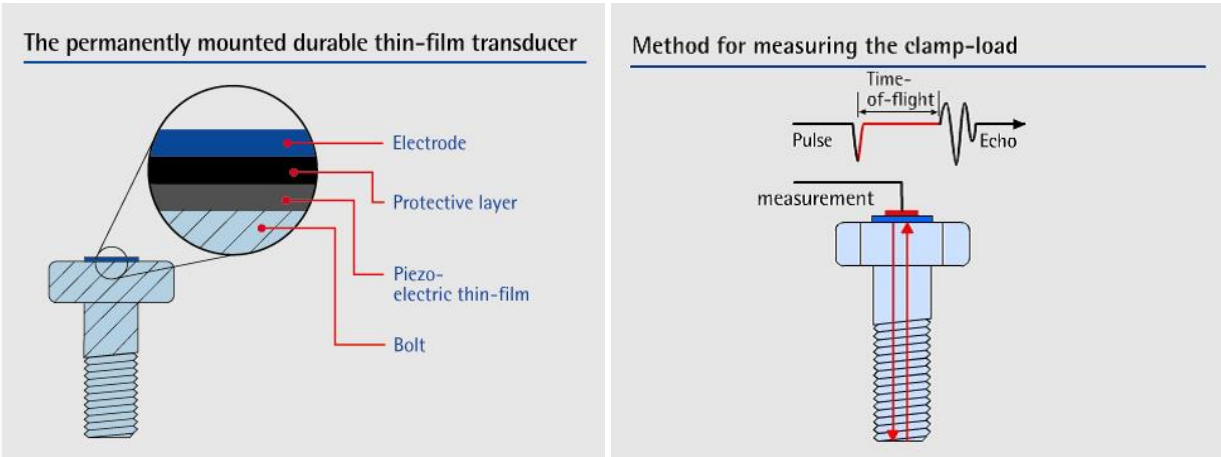


Figure 15 - Schematic view of the Permanently Mounted Transducer (left) and underlying principle to determine bolt preload (right) (Intellifast) [15]

The durability of the transducer is optimal, given that the materials used are immune to all environmental conditions.

The bolt preload and the wrench torque and angle are read in parallel, meaning that the target preload can be achieved precisely due to applying more or less torque, depending on the amount of friction. Since the PMTS requires the use of special instrumented bolts, the system is only relevant for newly built structures or renovated HSFG connections. Current applications include wind turbine towers and mining machinery.

2.2.7 Comparison of Available Methods to Determine Preload

In this chapter, five methods and their potential on performing in situ measurements for the determination of the preload of High Strength Friction Grip Connections were discussed. In the following table, the influence of several factors on the measurements, the duration of the installation procedure and the required material constants which should be determined if calibration is not performed, are described. The Piezoelectric Active Sensing Method is not included in the comparison, since this method is not suitable to determine the actual preload of the bolts, because of the influence of the embedment of the plates on the measurements. Similarly, the Permanent Mounted Transducer System is also not included since it cannot be used to determine the preload in bolts in existing structures/connections.

Table 1 - Comparison between available methods to determine preload

	Installation Process	Temperature Compensation	Residual Stress Influence	Couplant Influence	Flat surfaces required	Material constants needed	Sensitivity to external influences
Conventional Ultrasound Method	Short	Requires tests/calibration	No (if transducer in same position)	Yes	Yes	E C_L L_{eff} d	High
Velocity Ratio Method	Short	Requires tests/calibration	Yes	Yes	Yes	E C_L C_T L_{eff} d	High
Mechanical Resonance Frequency Shift Method	Short	Requires tests/calibration	No (if transducer in same position)	Yes	Yes	E C_L L_{eff} d	Low
Strain Gauge Method	Time-consuming	Using full Wheatstone Bridge	No	No	No	E d	Not at all

As shown in Table 1, all the acoustoelastic effect methods, which are sensitive to temperature variations, require tests to compensate for the temperature related effects. All acoustoelastic methods are influenced by variations in couplant's thickness and residual stresses, require smooth and flat surfaces and they are heavily affected by the environmental noise. If calibration is not performed, then the determination of numerous material constants and the effective length of the bolt is required, which is characterized by a high uncertainty. However, the installation process is short and simple.

The strain gauge method requires a relatively time consuming installation, but its measurements are affected only by temperature variations. However, in contrast to the acoustoelastic methods which require tests to determine the influence of temperature on the measurements, the temperature influence on the strain gauge measurements can be easily compensated for by using a full Wheatstone bridge. Determination of the bolt force based on the model is much easier with the strain gauge method, because it requires measurement only of the shank's diameter and an assumption on Young's modulus value, instead of measurement of numerous complicated material constants. Finally, this method is not affected by noise or residual stresses.

In conclusion, the acoustoelastic effect methods are mainly affected by factors which are governing the in situ measurements (environmental conditions) or factors difficult to control (e.g. couplant's thickness variations), which make any estimation on the accuracy of these

methods difficult. Compensation for temperature or stress effects requires time consuming tests. The strain gauge method offers a quite simple compensation for temperature variations and the new type of adhesive simplifies the installation process. Therefore, given that a strain gauge is properly installed in the shank of the bolt, it is possible to determine accurately the force. The feasibility of performing in situ measurements with this method is further assessed within this document.

3 Methods

3.1 Implementation of Strain Gauge Method

Based on the literature research carried out in Section 2.2, it was decided to perform tests with the Strain Gauge Method. Within the next subsections, the methods followed to arrive at experimental results are extensively discussed.

3.1.1 Installation of Strain Gauges in Bolt

In order to measure the strain of a bolt, which is subjected to a tensile load, the strain gauge is installed and glued in the shank of the bolt. It is very important that the strain gauge is properly placed into the bolt, so that the strain is accurately transferred from the bolt, through the adhesive and strain gauge backing, to the strain gauge itself. In this section the installation procedure of a strain gauge in the shank of the bolt is described.

Strain gauges of type *BTMC-3-D20-006LE* are used. As stated in the corresponding manual, the gauge has to be embedded into a hole of 2 mm diameter, drilled at the center of the bolt head with a depth of 40 mm, as shown in Fig. 4.6. The dimensions of the strain gauge used are shown Figure 16 [37].

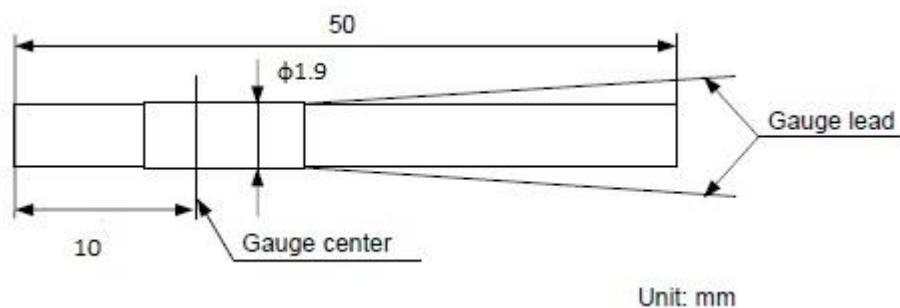


Figure 16 - Dimensions of BTMC strain gauge (TML) [16]

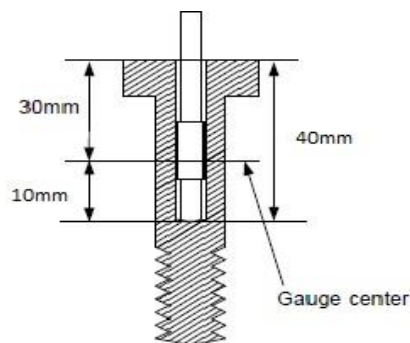


Figure 17 - Installation depth of BTMC strain gauge (not to scale) [16]

In order to ensure that the hole will be parallel to the bolt axis and drilled in the center of the head of the bolt, a hexagonal steel die is used. This hexagonal die has a hole drilled in the center of its upper side, designed in such a way to fit on the head of the bolt, as shown in Figure 18

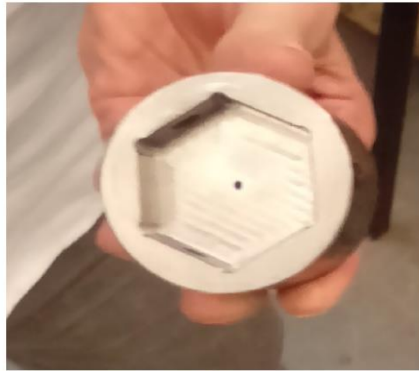


Figure 18 - Hexagonal steel die, designed to be fitted on the head of the bolt to (1) drill in the exact centre of the bolt and (2) ensure drilling parallel to the bolt axis

After positioning of the hexagonal die on the bolt head, the die was clamped on the head of the bolt in order to prevent any movement during the drilling of the hole, as shown in Figure 19.



Figure 19 - Clamping of the hexagonal die on the head of the bolt

After the hexagonal die is clamped on the head of the bolt, the drilling process begins. To drill the hole, a drill is used and two drill bits of different length, as shown in Figure 20. The shorter one is used in the beginning of the drilling process, because it is stiffer and it can withstand higher pressure compared to the longer one. The longer one is used to increase the depth of the hole. In order to make sure that the required depth of 40 mm, according to the technical specifications is not surpassed, a piece of tape is placed on the drill bit at the required length.



Figure 20 - Drill and drilling bits used in drilling process

As the drilling progresses, it is possible to see steel dust pieces coming out of the hole, which is a sign that the depth of the hole grows, as shown in Figure 21. After the whole length of the short drilling bit is in the hole, the drilling process continues with the longer one until the drilling depth of 40 mm is reached.

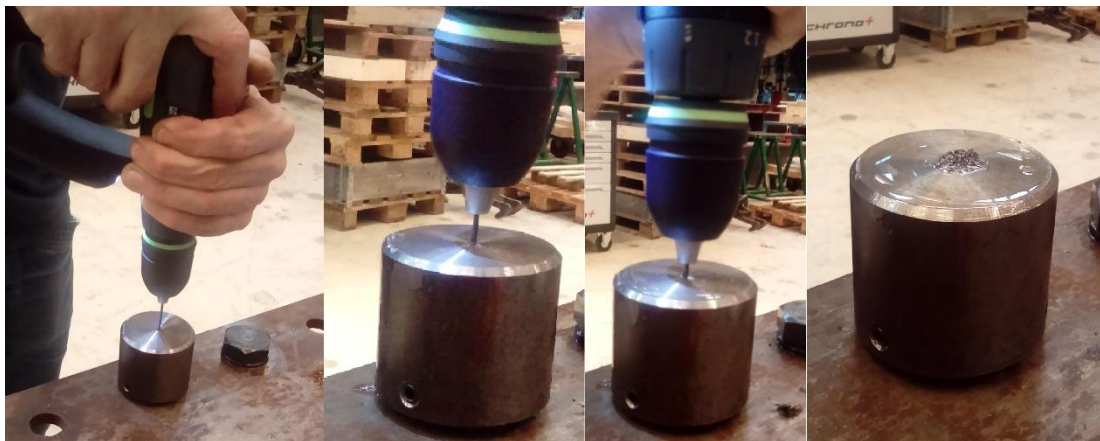


Figure 21 - Drilling process with the short drill bit

After the required depth is reached, the drill and the hexagonal die are removed. The hole on the head of the bolt is shown in Figure 22. After the drilling of the hole (and before the installation of the strain gauge), the hole has to be cleaned. A solvent was poured into the hole using a syringe to wash out dust. To remove remainders in the hole, a solvent-dampened tissue was used which was rolled on a drill bit. Then, the remaining solvent in the inner hole was removed with a clean tissue. The removal of the solvent is of high importance, because if solvent or dust remain in the hole, curing failure of adhesive may occur. After that, the head of the bolt was cleaned.

Now that the hole is cleaned the installation of the strain gauge can commence. In Figure 23 the strain gauge and the gauge wires, which transfer the changes in the electrical resistance of the strain gauge due to strain to the terminal are shown.



Figure 22 - Strain gauge hole drilled from the bolt head into the shank



Figure 23 - BTMC strain gauge

In the next step the adhesive is injected into the hole by a syringe, as shown in Fig. 4.12.



Figure 24 – Injection of CN adhesive using a syringe into pre-drilled hole

Immediately after filling the hole with adhesive, the gauge was inserted until the bottom of the gauge hit the bottom of the hole. The strain gauge is being held from its upper part by a tweezer and installed with attention in the hole in such a way that the wires stay out of the hole in order to be connected later with the terminal, as shown in Figure 25.



Figure 25 - Installation of the strain gauge in the hole filled with adhesive

After installation of the strain gauge, the bolt was kept calm in upright position at room temperature for 24 hours to allow for curing of the adhesive. After the adhesive was cured, the pipe extruded from the hole was cut. After cutting the pipe, a connecting terminal was installed and connected to the gauge lead and an instrumentation lead wire was connected to the terminal.

3.1.2 Calibration of Bolt Force

Since a strain gauge measures strain rather than force, it is necessary to derive the (theoretically linear) relationship between force and strain by experiments. Calibration experiments were carried out for the cases in which a BTMC strain gauge was installed:

- in unloaded vertical bolt;
- in a loaded vertical bolt;
- in an unloaded horizontal bolt;
- in an unloaded vertical bolt using drilling oil.

The different set-ups provide insight in the practical feasibility of the method.

M24 bolts were used of property class 10.9, as well as washers with a thickness of 4 mm. The bolts were lubricated and they were placed in a Skidmore – Wilhelm device with a clamping length of 50 mm. The Skidmore – Wilhelm instrument is shown in Figure 26 and Figure 27. The instrument consists of hydraulic cylinder with a hole through the middle. The bolt runs through the hole and the nut and the washer are added and the bolt is tightened. This raises the pressure in the cylinder and a calibrated pressure gauge interprets the increase in pressure in terms of clamping force.

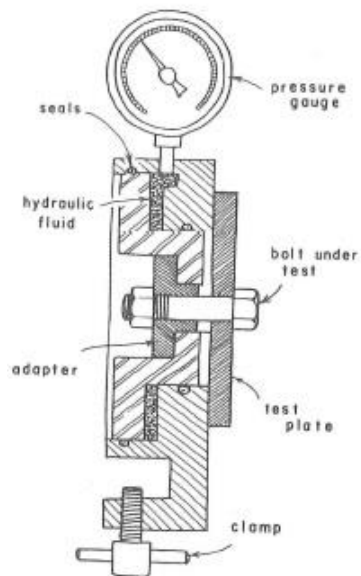


Figure 26 – Schematic of Skidmore – Wilhelm instrument [2]

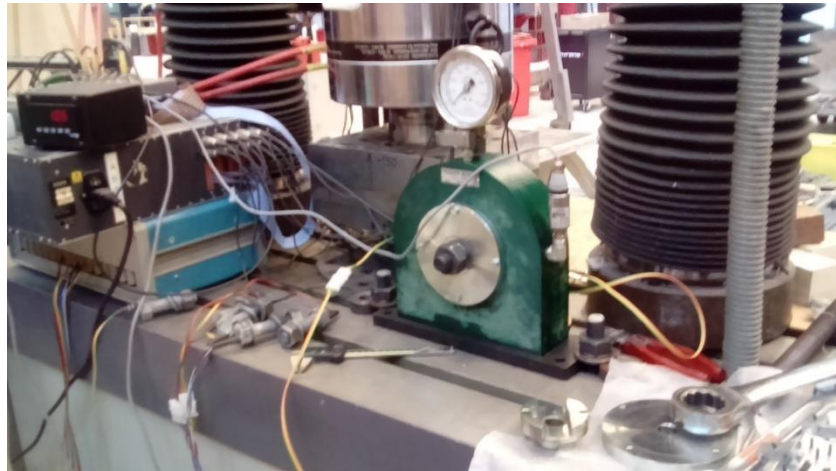


Figure 27 - Skidmore – Wilhelm device as used for calibration

After installation of the bolt, it was preloaded using a pneumatic tightening device to the minimum level prescribed by the Eurocode, in this case $F_{p,C} = 0,7f_{ub}A_s = 247$ kN. The preload applied to the bolt was registered by a computer that was connected to the Skidmore – Wilhelm instrument. The strain gauge measured the strain, and this data was recorded continuously. The strain gauge was connected with a quarter bridge configuration, as the test was performed indoors, where no temperature variation is expected.

The bolts are loaded and unloaded for a total of three times in order to obtain three calibration lines. For each calibration line, the slope was derived, which is called calibration factor (the ratio between the force on the bolt and the strain in the shank of the bolt). The comparison of the three calibration factors of each bolt, give an indication of the uncertainty of the method.

3.2 Determination of Preload in Bolts of an Existing Bridge

In order to test the Strain Gauge Method in true practice, the preload in the bolts of the Middachter bridge was determined after approval of the Dutch Ministry of Infrastructure. The Middachter bridge was built in 1974, and for this reason it was considered important to know the actual preload in one of its connections. The bridge is illustrated through Figure 28, whereas the connection under consideration is presented in Figure 29.

The preload in the bolts of a bottom flange connection of an inverted T beam is determined. Based on the original design plans, the connection consists of two cover plates of 26 mm thickness, a main plate with a thickness equal to 32 mm and two washers with a thickness of 4 mm, resulting in a clamping length equal to 92 mm, as shown in Figure 30. A total of 84 M24 bolts of a grade 10.9 with a nominal length of 120 mm hold the plates together. In Figure 31 the dimensions of the connection are shown.

The in-situ measurements were completed within three days. The strain gauges were installed in the shank of the bolts and the bolts with a strain gauge in their shank were loosened to determine the strain and then they were replaced by new bolts. In this chapter, the procedure which was followed in situ and in the lab, in order to convert the strain gauge measurements to force, is described. Furthermore, the initial preload is estimated based on the tests which were performed in the lab and it is compared with the expected (theoretical) values. Lastly, two ways of achieving the estimated initial preload are discussed.



Figure 28 - Middachterbrug (Middachter bridge) in 1974 (Rijkswaterstaat)



Figure 29 - The connection of the Middachter bridge under consideration

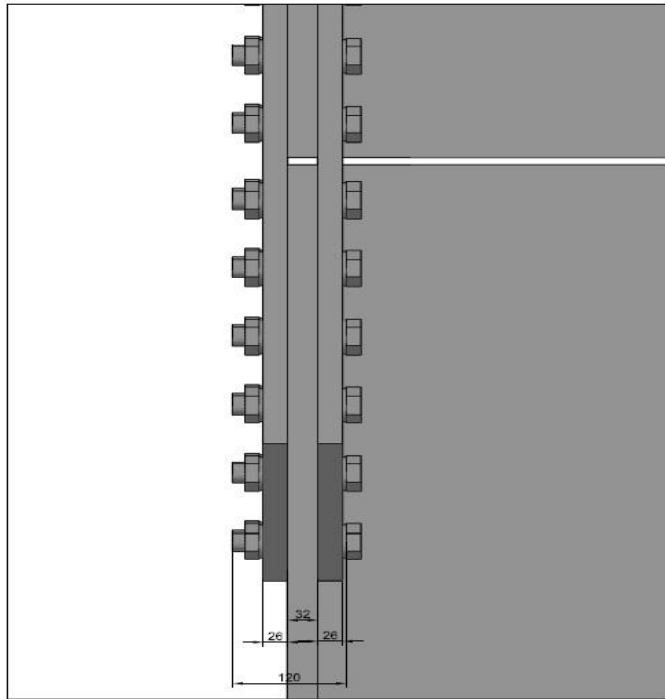


Figure 30 - Side view of the tested connection

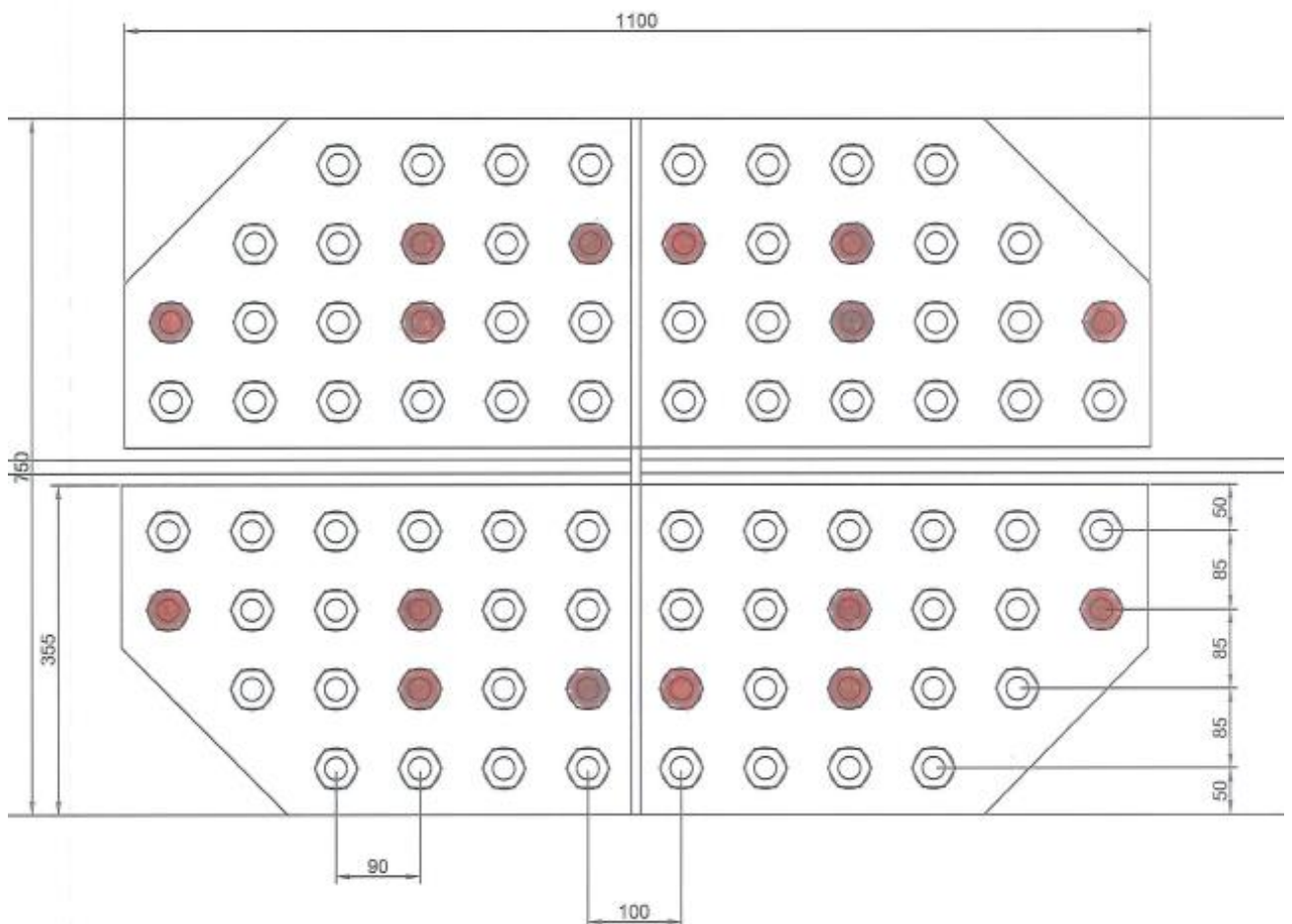


Figure 31 - Plan view of the connection under consideration, indicating the four quadrants of the connection. In orange the bolts of the experimental programme are indicated.

3.2.1 Procedure

Sixteen strain gauges were used for the in situ measurements. Therefore, 16 bolts were chosen in a way that the measurements could provide as much information as possible about the actual preload of the bolts. The strain gauges were installed in the shank of the bolts and then the bolts were loosened to obtain the change of the strain in the shank of the bolts. Then, the tested bolts will be replaced by new bolts, which will be preloaded using the Combined Method.

The only information about the preloading method of the bolts in the connection is that the preload was either achieved by the Torque or the Combined Method, but it is not explicitly known which method was used. However, based on the fact that all the bolts were tightened by the same method, it is to be expected that the bolts of each quadrant will have a similar r preload as that in the counterpart bolts located at the other quadrants. If the preload of each bolt in one quadrant and the preload of their counterparts in the other quadrants is known, it will be possible to determine the scatter in the initial preload applied on these bolts. For this reason, it was decided to test four bolts in one quadrant and their counterparts in the other three quadrants.

The next step is to choose these 4 bolts out of the 16 bolts in each quadrant. There are two questions that need to be answered:

- What is the relationship between the actual preload of bolts of the same vertical row?
- What is the influence of tightening a bolt on the neighboring bolts?

In order to give an answer to the first question, two bolts in the same vertical row were selected to be tested. To answer the second question, also a pair of adjacent bolts was chosen to be part of the experiments. This way, during preloading of the newly replaced bolt, it can be determined if the preloading affects the strain gauge signal of the other bolts. Therefore, in each quadrant a different tightening sequence was chosen, in such a way that in the end of the process the influence of tightening of each new bolt on the other bolts is known. Figure 32 presents the tested bolts and their numbering.

The in-situ experiments started on October 31 2016 and were completed on November 2 2016. On the first and second day, the strain gauges were installed in the bolts and on the third day the bolts were loosened and the measurements were obtained.

In order to drill a hole in the head of the bolts to install the strain gauges, the painting layer had to be removed with a peening tool. Then a hole was drilled in the head of the bolts. Drilling oil was used in order to facilitate the drilling process. Then, the hole was cleaned by injecting a solvent with a syringe to wash out the dust. At the second stage of the cleaning process, an air pistol was used in order to remove any remaining dust or drilling oil from the strain gauge hole.

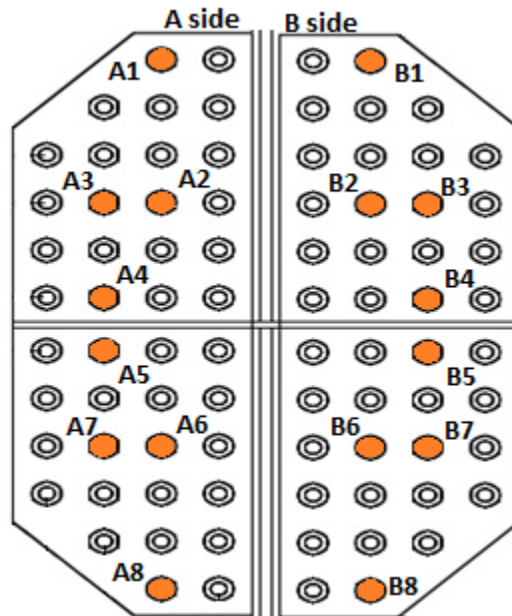


Figure 32 - Tested bolts and their numbering

After having injected the adhesive in the hole, the strain gauges were inserted in the shank. Then, the bolts were left for 48 hours to allow hardening of the glue. Some of the strain gauges were installed in the second day, so the time for hardening of the glue was 24 hours. The gauge leads were connected with the wire connectors and covered with varnish to prevent e.g. moisture from influencing the measurements. In Figure 33 the bolts after the installation of the strain gauges are shown.



Figure 33 –Bolts with their installed strain gauges

On the third day, the strain gauges were connected with the terminal, using a full Wheatstone bridge configuration, which cancels the effects of temperature variation, as it is already explained in Section 2.2.5. For the in situ measurements the full Wheatstone bridge was used with three resistors responding only on changes because of temperature variations. The Wheatstone bridge was attached to the head of the bolt so that the three resistors experience the same temperature variations as the fourth resistor/strain gauge, which was inserted in the shank of the bolt.

The measurements started by loosening the nut of the first bolt with a hydraulic torque wrench by holding the head of the bolt stationary with another wrench, as shown in Figure 34 (left). When the nut was completely loosened the resulting voltage signal was received in a laptop which was connected with the terminal, as shown in in Figure 34 (right).



Figure 34 - Left: loosening of nut, right: removed bolts connected to terminal and laptop

After the bolt with the implanted strain gauge was loosened, a new bolt was tightened with a hydraulic torque wrench by the combined method. Before the new bolt was tightened, the surface of the plate near the hole was ground to remove any layers which would result in a non-perpendicularity between the plate surface and the bolt axis

The abovementioned procedure was repeated for all the bolts of the connection. In Figure 35 one of the sides of the connection with the newly installed bolts is shown, and in Figure 36 the (old) removed bolts with the implanted strain gauges are shown, which were taken to the TU Delft laboratory to perform calibration and hence determine the preload present before bolt removal. The calibration was carried out using an Universal Testing Machine (UTM) in which the bolt was axially loaded, as exemplified through Figure 37. After calibration, the bolts have been loaded to failure.



Figure 35 - One side of the connection after installation of the new bolts

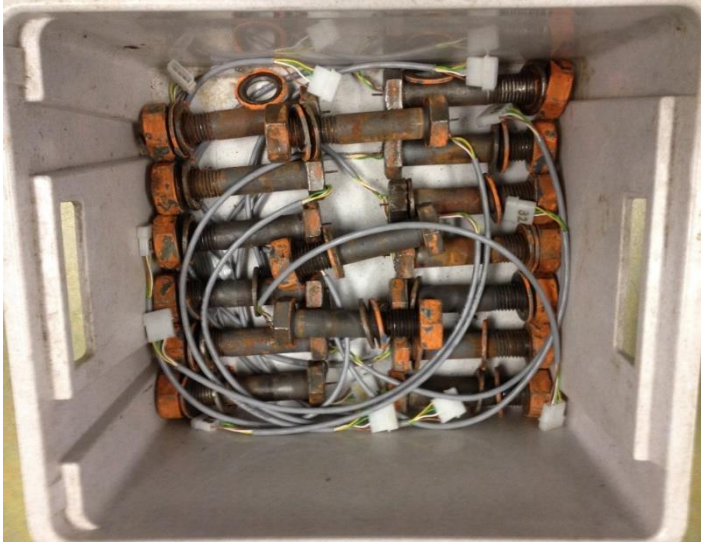


Figure 36 - Bolts removed from the connection were brought to the TU Delft laboratory for calibration

A detailed picture sequence of the method is included in Appendix A.

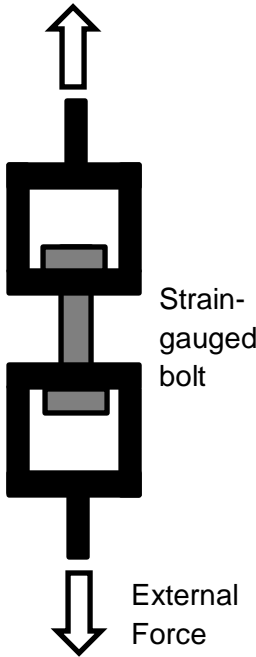
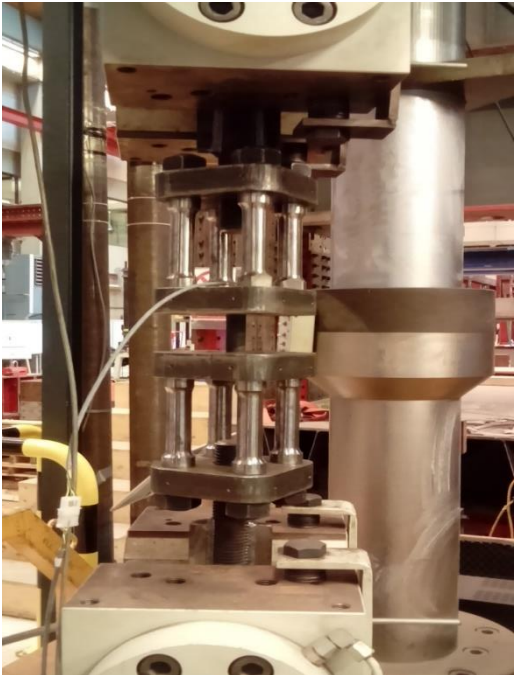


Figure 37 - Universal Testing Machine (UTM) used to perform calibration (left) and schematic overview (right)

4 Results

4.1 Calibration of Bolt Force based on Strain Gauge Method

The calibration of the strain gauges was done three times for all four types of bolts tested. Based on the strain (measured by the strain gauge) and clamping force (measured by the Skidmore – Wilhelm device) the calibration is performed by determining the slope of the strain-force relationship, as exemplified through Figure 38. In Table 2 the calibration factors for the four different tested bolts are presented.

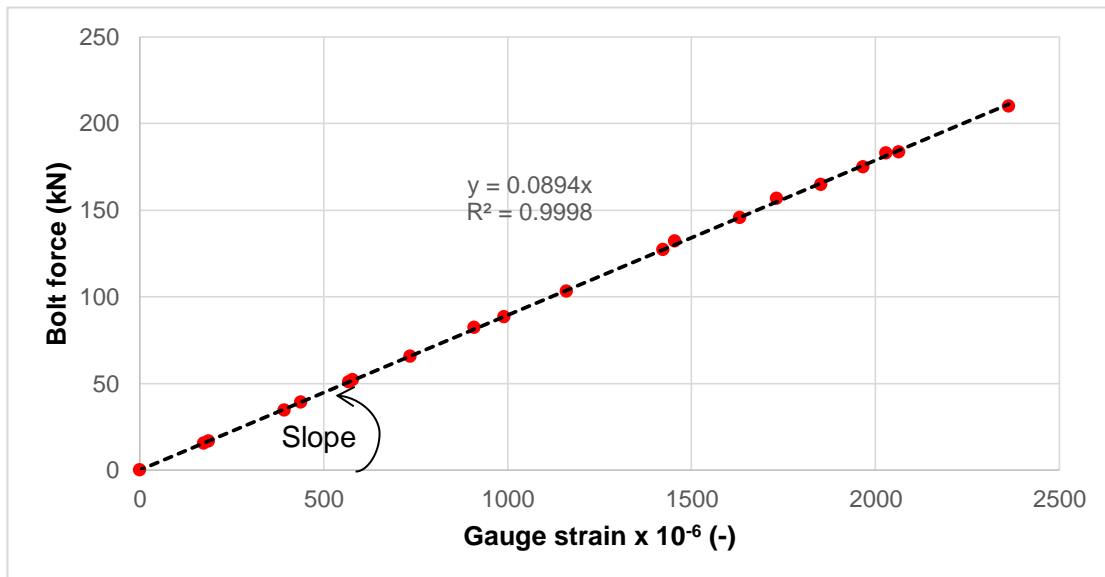


Figure 38 - Example of relationship between bolt force and bolt strain used to determine the calibration factor

Table 2 - Calibration factors for the strain gauges within bolts for laboratory tests

	Calibration factor (CF) [kN/ $\mu\epsilon$]			Spread
	Test 1	Test 2	Test 3	
unloaded vertical bolt	0,0833	0,0836	0,0837	0,3%
unloaded vertical bolt using drilling oil	0,0917	0,0894	0,0892	1,8%
loaded vertical bolt	0,0847	0,0845	0,0848	0,2%
unloaded horizontal bolt	0,0868	0,0869	0,0867	0,1%

The bolt force is related to the strain gauge measurements via $F = CF \cdot \epsilon$

The values of coefficients of determination R^2 were between 0.9998 and 1 for all strain gauges. The high values of coefficient of determination R^2 indicate that there is a high strength of linear association between the applied force F and the measured strain ϵ . The spread in the calibration factors for the same bolts is very limited. However, the difference of calibration factors of different bolts is much higher, but it is reasonable as these four bolts do not come from the same batch and they are also characterized by different actual dimensions.

4.2 Preload in Bolts of an Existing Bridge

As explained in Section 3.2, the bolts A1-A8 and B1-B8 were first instrumented with a strain gauge, after which the preload was removed from the bolts in order to capture the change in bolt strain $\Delta\varepsilon$. Unfortunately, some of the installed strain gauges did not work. The strain gauges of bolts A3 and B5 gave unreliable readings. Bolt B8 was accidentally loosened before connected with the terminal. Therefore, bolts A3, B5 and B8 are excluded in the Results Section.

The difference in strain as measured by the strain gauges upon unloading of the bolt is presented in Table 3 for all bolts. The calibration factors as determined using the Universal Testing Machine (UTM) are presented in Table 4; the calibration lines are presented in Appendix C. Combining Table 3 and Table 4, the preload in the original bolts can be determined, as expressed in Table 5.

Table 3 - Difference in bolt strain compared to original (loaded) and unloaded situation

Bolt ID	$\Delta\varepsilon \cdot 10^{-6}(-)$	Bolt ID	$\Delta\varepsilon \cdot 10^{-6}(-)$
A1	3239	B1	3466
A2	3346	B2	3385
A3	-	B3	3531
A4	3579	B4	3537
A5	3502	B5	-
A6	3625	B6	3676
A7	2104	B7	3074
A8	3754	B8	-

The bolts taken from the connection of the Middachter bridge come from the same batch of bolts, therefore it is expected that their calibration factors will be of similar magnitude. Differences between calibration factors observed in Table 4 are mainly caused by the different actual diameters of these bolts and the difference in the actual position of the strain gauges in the shank of the bolts.

Table 4 - Calibration factors for bolts removed from Middachter bridge

Bolt ID	Calibration Factor [kN/ $\mu\varepsilon$]	R ²	Bolt ID	Calibration factor [kN/ $\mu\varepsilon$]	R ²
A1	0,0889	0,9988	B1	0,0897	0,9999
A2	0,0985	0,9999	B2	0,0899	0,9996
A3	-	-	B3	0,0899	0,9999
A4	0,0917	0,9999	B4	0,0895	0,9997
A5	0,0931	0,9999	B5	-	-
A6	0,0919	0,9999	B6	0,0876	0,9989
A7	0,0908	0,9999	B7	0,0898	0,9999
A8	0,0914	0,9999	B8	-	-

Table 5 - Preload in bolts taken from the Middachter bridge

Bolt ID	Preload (kN)	Bolt ID	Preload (kN)
A1	280	B1	315
A2	335	B2	305
A3	-	B3	315
A4	346	B4	310
A5	330	B5	-
A6	331	B6	323
A7	189	B7	277
A8	344	B8	-

The results presented in Table 5 are visualized through Figure 39 and Figure 40 for the bolts from the A- and B-series, respectively .

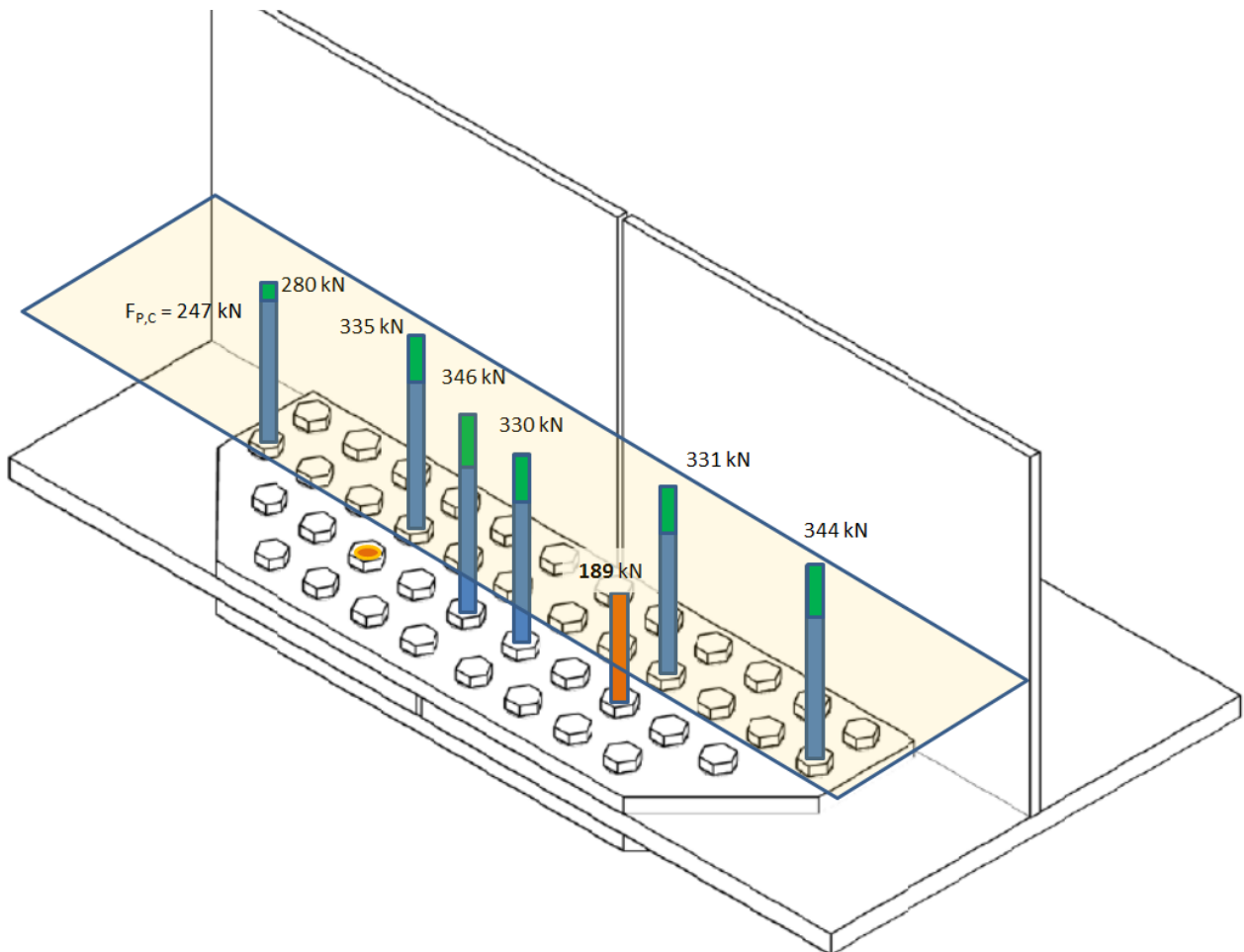


Figure 39 - Preload in bolts from A-series, blue: minimum preload $F_{p,c}$, green: amount of preload exceeding $F_{p,c}$, orange: bolt with preload lower than $F_{p,c}$. No bar means that procedure could not be followed properly.

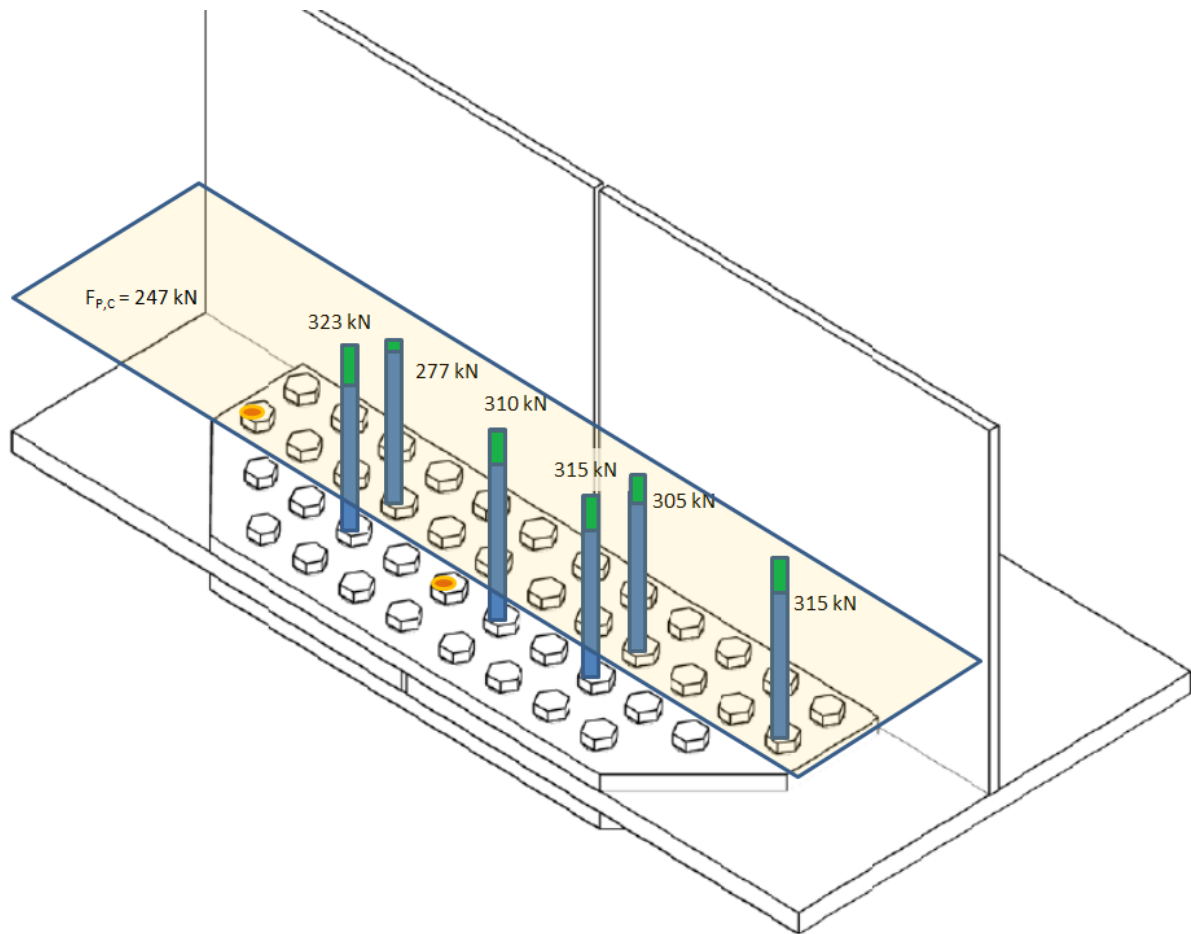


Figure 40 – Preload in bolts from B-series, blue: minimum preload $F_{p,c}$, green: amount of preload exceeding $F_{p,c}$, orange: bolt with preload lower than $F_{p,c}$. No bar means that procedure could not be followed properly.

The ultimate loads the bolts were able to resist in the Universal Testing Machine are presented in Table 6. The individual load-displacement diagrams are presented in Appendix D.

Table 6 – Force leading to bolt failure in Universal Testing Machine

Bolt ID	Force at failure (kN)	Bolt ID	Force at failure (kN)
A1	376	B1	365
A2	388	B2	368
A3	358	B3	373
A4	364	B4	351
A5	385	B5	363
A6	394	B6	387
A7	-	B7	385
A8	395	B8	343

5 Discussion

5.1 Feasibility of the Strain Gauge Method to determine Preload

Both for the laboratory and in-situ experiments, a linear relationship between bolt force and measured strain was found by calibration. The idea of measuring the bolt preload using the Strain Gauge Method is not to calibrate every bolt, since this is rather laborious and time-consuming. Therefore, the feasibility of the Strain Gauge Method to determine the actual preload in a bolt is assessed based on nominal properties, such as the nominal area and Young's Modulus of the bolts. The difference between the calibrated results and nominal results then gives insight into the spread and hence certainty with which the preload can be determined.

Both the laboratory and in-situ experiments are used for the feasibility study, since the combination gives a broad spectrum of data. The goal is to make a prediction for the calibration factor based on nominal material properties, i.e. via the expressions in Eq. 14-Eq. 16.

$$F = (CF)_{act} \cdot \varepsilon = (EA)_{act} \cdot \varepsilon \quad \text{Eq. 14}$$

$$CF_{nom} = (EA)_{nom} = E \cdot \frac{1}{4} \pi d^2 \quad \text{Eq. 15}$$

$$CF_{act} = \beta \cdot (CF)_{nom} \quad \text{Eq. 16}$$

With:

E	Young's Modulus (actual or nominal) [F/L ²]
A	Bolt area (actual or nominal) [L ²]
ε	Bolt strain [-]
β	Correction factor [-]

The actual calibration factor CF_{act} and nominal calibration factor CF_{nom} are plotted in Figure 41 for all conducted experiments.

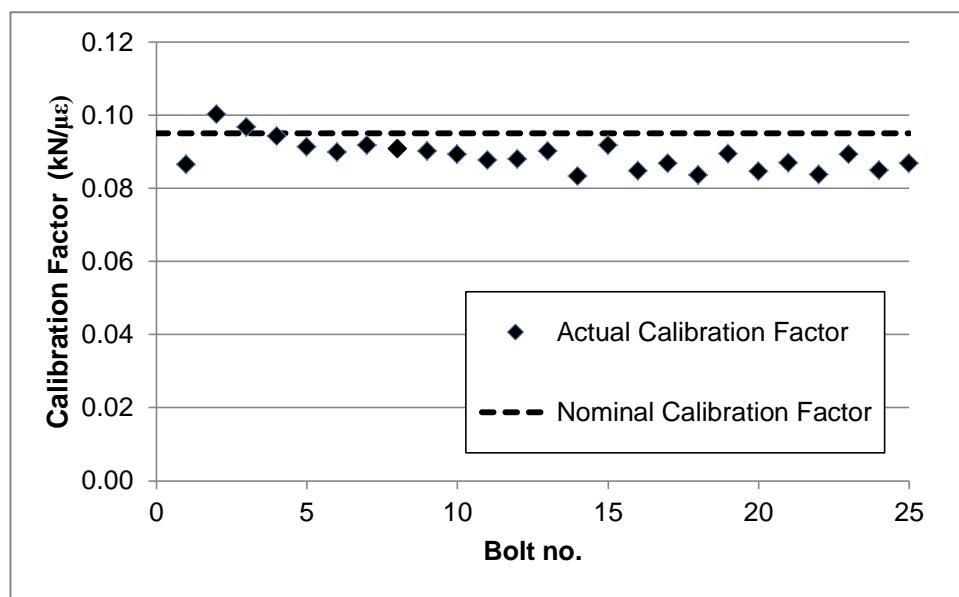


Figure 41 - Actual Calibration Factor for all tested M24 bolts

Based on the data presented in Figure 41 the correction factor β can be determined, which serves as a correction for the average value of the actual and nominal correction factors. It is determined that $\beta = 0,935$.

The spread within the values for the actual calibration factor CF_{act} is now analysed. The mean value of the actual calibration factor is $CF_{act,\mu} = 0,889 \text{ kN}/\mu\epsilon$, whereas the standard deviation is $\sigma = 0,00411 \text{ kN}/\mu\epsilon$. The magnitude of the actual calibration factor with a 95% exceedance probability for $n - 1 = 24$ is determined through Eq. 17 under the assumption that the actual calibration factor is normally distributed.

$$(CF)_{ch} = (CF)_{act,\mu} - 1,711 \cdot \sigma \quad \text{Eq. 17}$$

With:

$(CF)_{ch}$	Characteristic calibration factor (95% exceedance prob.) [F]
$(CF)_{act,\mu}$	Mean observed calibration factor [F]
σ	Standard deviation [F]

By filling out Eq. 17 it appears that $(CF)_{ch} = 0,0818 \text{ kN}/\mu\epsilon$, which means that the characteristic calibration factor is 8% smaller than the average calibration factor. Hence, to have 95% certainty of having a certain preload, it is necessary to prove that the bolt strain in practice is $1/\beta \cdot [(CF)_{act,\mu}/(CF)_{ch}] = 1/0,935 \cdot 1/0,92 = 16,1\%$ higher than required based on nominal bolt dimensions and properties.

One of the aspects that may negatively influence the true preload within the bolt is if the actual bolt diameter is smaller than the nominal diameter, since the diameter has a large influence on the true bolt area A . For an M24 bolt, the ratio between $E_{nom}A_{min}$ and $E_{nom}A_{nom}$ is 0,93, meaning that for a full guarantee the bolt strain that needs to be proven in practice increases to 124% of the nominal bolt strain corresponding to a certain preload. However, given that the nominal bolt diameter is the average bolt diameter, it may not be necessary to include such a correction since the effects will cancel out over the entire connection. It must be noted that when using the Torque Method for a bolt with a smaller diameter than the nominal diameter, the initial bolt preload achieved within the bolt is slightly larger (cf. Eq. 4, p.4) which partially offsets the effect described above.

Based on the limited spread in the results, it is suggested that the Strain Gauge Method offers a good first estimate of the bolt preload. It should be noted that not the bolt force itself is measured, but the bolt strain. Using a correction factor β to account for differences in nominal and actual bolt stiffness, it is concluded that a roughly 10% higher preload needs to be proven than the nominally required preload level in order to reach the conclusion that this nominally required preload level is still present within the fastener. However, it should be noted that the current investigation was only performed with M24 bolts, but did include include bolts from at least two different batches with different age.

5.2 Level of Preload in Existing Bridge

The preload of the bolts in the Middachter bridge is significantly higher than the minimally required pretension force $F_{p,C} = 247$ kN, on average 27% if bolt A7 is not taken into account. Such high preloads can only exist if the bolt has been significantly overtightened during the erection of the structure. Proof of overtightening is found in the form of damaged threads of several bolts. The most heavily damaged threads belong to bolt A7, which is the only bolt with a preload lower than $F_{p,C}$, and was the only bolt that was stuck within its hole and was loaded in bearing. In Table 7 the preload in the bolt, as well as the bolt force at failure and the thread condition per bolt are presented.

Table 7 –Preload, failure force and thread condition per bolt

Bolt ID	Preload (kN)	Force at failure (kN)	Thread Condition	Bolt ID	Preload (kN)	Force at failure (kN)	Thread Condition
A1	280	376	Bad	B1	315	365	Good
A2	335	388	Good	B2	305	368	Good
A3	-	-	-	B3	315	373	Good
A4	346	364	Bad	B4	310	351*	Good
A5	330	385	Good	B5	-	363	Good
A6	331	394	Good	B6	323	387	Good
A7	189**	-	Poor	B7	277	385	Good
A8	344	395	Good	B8	-	343*	Poor

*: Failure force is lower than nominal failure load ($f_{ub}A_s = 353$ kN)

It is not well documented which method was used in the 1970s to tighten the bolts. According to the design plans, the Torque Method should be used using a torque $T = 1100$ Nm and a k-factor $k = 0,177$ [–]. This torque and k factor result in a preload force of 259 kN. The assumed k-factor is relatively high, which means that bolts with a relatively lower thread friction coefficient were significantly overtightened, resulting in the damaged threads. Based on the results, it appears that the complete batch of bolts has a lower k-factor than was assumed during construction. If the bolts were tightened to 110% of the average bolt preload (compensating for any preload losses by 10%) using the Torque Method with $T = 1100$ Nm, then the corresponding k-factor would be $k = 0,133$ [–] which is a courant value for lubricated bolts.

Another possibility is that the Combined Method was used to tighten the bolt, as was prescribed by the governing authority. According to EN 14399, when the Combined Method is used, the k-factor should be $0,10$ [–] $\leq k \leq 0,16$ [–]. For the determination of the torque applied for the first step the k-factor can be considered equal to $k = 0,13$ [–]. Hence, the bolt preload after the first step of the Combined Method is maximally $0,68 \cdot f_{ub}A_s$ and minimally $0,43 \cdot f_{ub}A_s$. During the second step of the Combined Method the bolts must be turned 90°, which leads to the bolt being loaded into its plastic region to a preload of roughly $0,90 \cdot$

$f_{ub}A_s = 317,7$ kN (mostly independent of true k-factor). However, when the preload levels are compared to the expected preload level belonging to the Combined Method (Figure 42), it appears that the preload levels are generally too high to have been caused by tightening using the Combined Method. This is supported by the fact that in the 42 year service lifetime the bolts will have experienced a loss in preload of at least a few percent, meaning that initially the bolt preload was even higher than now.

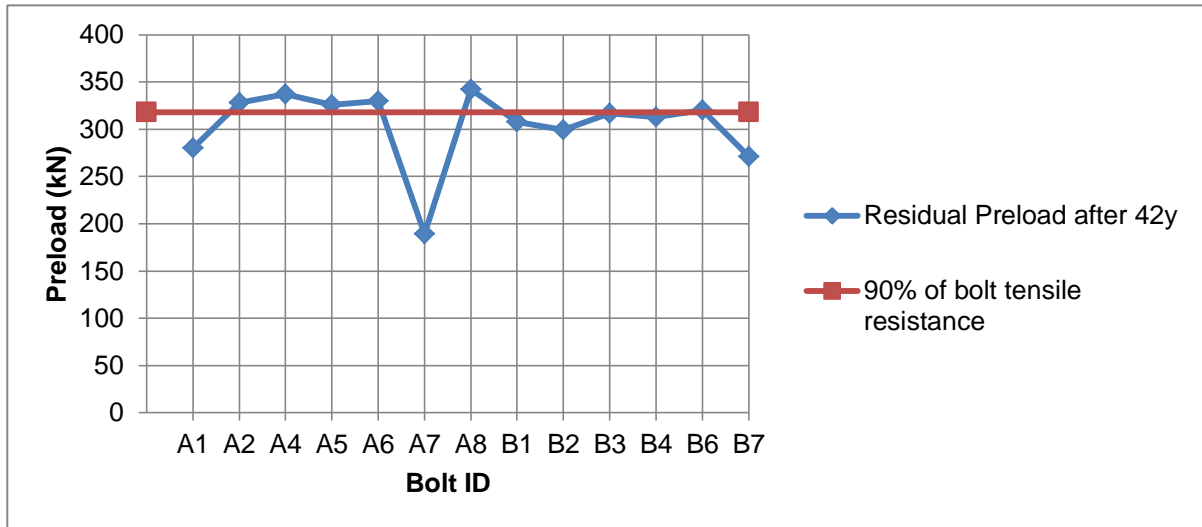


Figure 42 - Residual preload compared to the average bolt force obtained using the Combined Method

The fact that overtightening of bolts took place is supported by the shape of the bolt fracture surface. Since the bolts are loaded to failure by direct tension, a fracture cf. Figure 43(a) is to be expected. However, in 10 out of the total of 16 bolts (63%) fracture occurred cf. Figure 43(b), which is typical for bolts torqued into tension. Since the fracture mechanisms presented in Figure 43 are valid for new bolts, it can be concluded that these bolts are internally damaged, most likely as a result of overtightening of the bolts, causing a change in fracture mechanism.

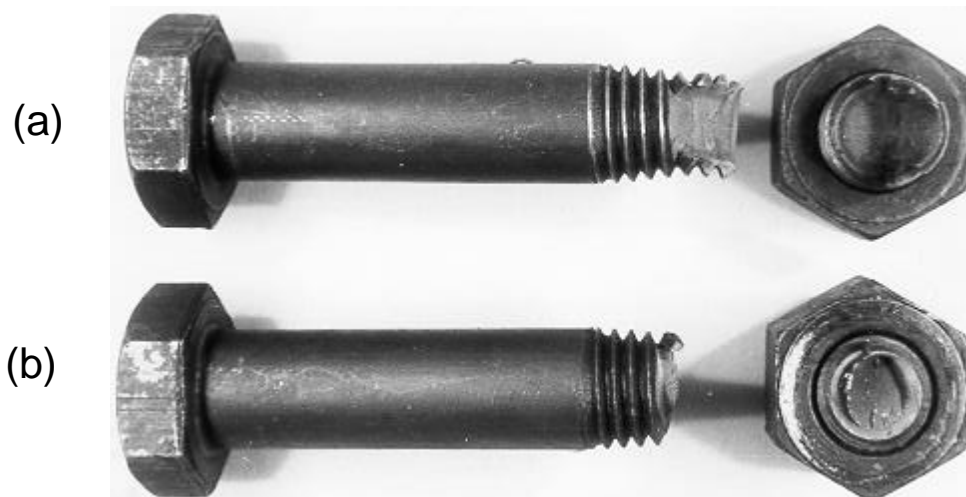


Figure 43 - Bolt fracture for a bolt loaded in (a) direct tension and (b) torqued tension [20]

It is concluded that the procedure used for bolt tightening cannot have been the Combined Method, since the preload levels are higher than what can generally be achieved using this method. The Torque Method, however, is capable of reaching such rather high preload levels (near bolt failure), especially in case that the k-factor is lower than assumed (as for the bolts originating from the Middachter bridge). A possible reason for having bolts with a lower k-factor, is that it was originally planned to use non-lubricated bolts, whereas in practice lubricated bolts were used.

6 Conclusions & Recommendations

6.1 Conclusion

Measuring preload is necessary in order to determine whether or not the bolt preload has dropped below the minimum required preload $F_{p,C}$ and can give an indication on whether or not structural improvement is necessary. An actual preload lower than $F_{p,C}$ for instance decreases fatigue life and may negatively affect the slip behaviour of HSFG connections.

Determining preload using any ultrasonic methods requires detailed information on several material constants, including the longitudinal and transversal acoustoelastic constants (C_L, C_T), the Young's Modulus E , as well as the effective length of the bolt (L_{eff}). Moreover, any ultrasonic method is rather sensitive to (external) influences such as temperature, residual stresses, surface finishing and couplant thickness variation. Only if Permanent Mounted Transducer Systems (PMTS) are used, the bolt preload can be determined precisely ($\pm 3\%$) without requiring material constants, since all bolts are instrumented and calibrated individually by the manufacturer. However, at present the use of PMTS is not possible to determine the preload, since the method requires the use of (new) special instrumented bolts.

In the past, the Strain Gauge Method, as a method to determine the preload, was not suitable to be used in-situ, because of a specific vacuum treatment needed to ensure that no air was entrained within the adhesive. Recently, a different type of adhesive has become available on the market that does not require any vacuum treatment. As a result, the bolt preload can be determined by untightening the bolt and capturing the change in bolt strain, which in combination with the bolt axial stiffness is a predictor for the bolt preload. Key advantages of the Strain Gauge Method over ultrasonic methods are that:

- Temperature differences can be compensated for using a Wheatstone bridge;
- Residual stresses do not influence the measurements, since the strain gauge is in the centre of the bolt;
- There is no influence of smoothness/flatness of surfaces;
- Only well-defined material property/geometry with a relatively low spread (E, d) are required to transfer bolt strain into bolt force.

Based on laboratory and in-situ measurements on the Middachter bridge, it can be concluded that the bolt strain is a good predictor for the bolt preload. In order to have 95% confidence of achieving a certain minimum preload, a roughly 10% higher bolt strain needs to be proven than minimally required (i.e. after taking into account the conversion factor β). Hence, it appears that the Strain Gauge Method is a good method to give an initial prediction on the bolt preload and that only in those cases, in which the margin with respect to $F_{p,C}$ is less than 10%, additional and more exact determination of bolt preload is required.

Finally, it is concluded that it is promising to further investigate the Strain Gauge Method as a predictor of the bolt preload, given the developments in adhesive technology and the low spread in the in-situ results. Although the Strain Gauge Method is more time-consuming than ultrasonic methods, the Strain Gauge Method precedes all other currently applicable methods with respect to precision and insensitivity to external influences. In all currently available methods it is required to loosen a bolt to exactly determine the residual bolt preload, except if a method based on PMTS is used.

6.2 Recommendations

It is recommended to further investigate:

- The relationship between bolt force and bolt strain for differently sized bolts, in order to determine the general applicability of the Strain Gauge Method in practice.
- New methods which can determine bolt preload without untightening of the bolt.
- The (financial) feasibility and necessity (e.g. minimum amount of instrumented bolts needed per connection) of bolts with permanently mounted transducers (PMTS) in civil engineering structures

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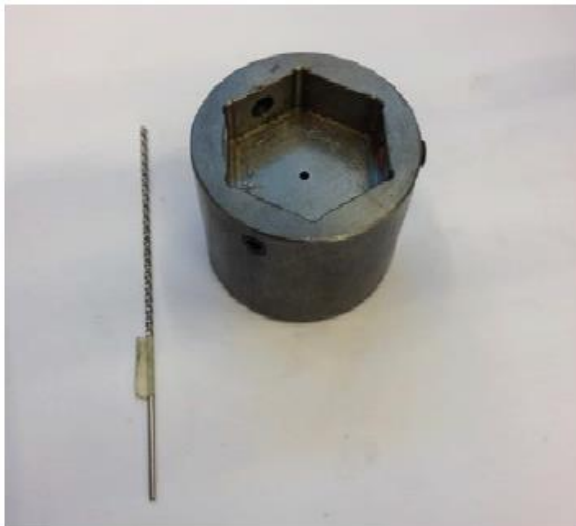
Appendix A: Photo Sequence of Strain Gauge Method



BTMC-3-D20-006LE strain gauges in combination with CN adhesive were used to instrument the bolts



Strain gauge insertion pipe. The strain gauge itself is 3 mm long and is situated 10 mm from the lower end of the pipe.



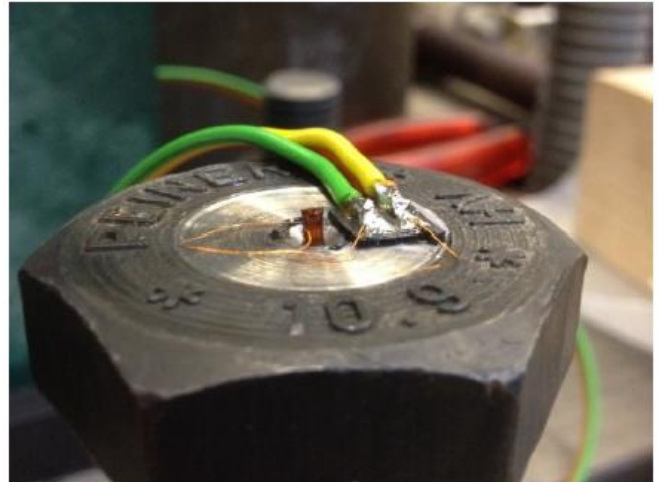
Steel drilling support device that is clamped on the bolt head to ensure controlled/correct drilling of the 2 mm holes in the HV10.9 bolts.



A hand held, battery driven machine was used to drill the holes. Special long VHSS steel drills were necessary to drill in the 10.9 material.

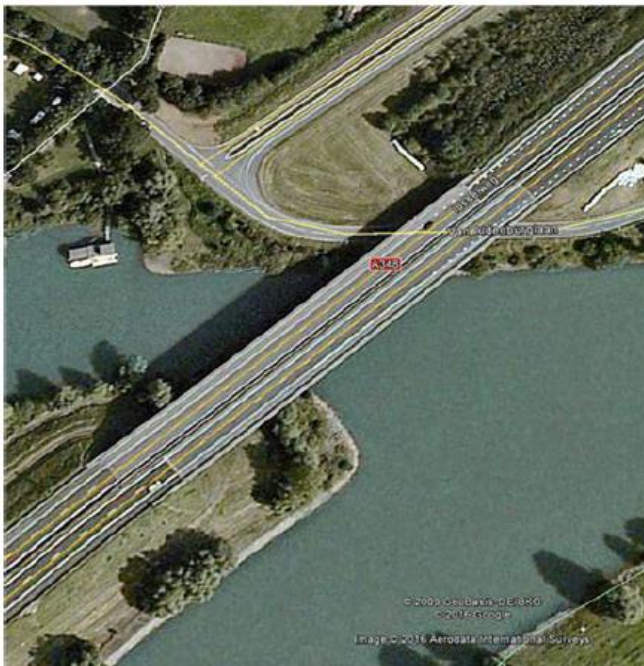


Skills were developed to achieve an efficient procedure to manually drill the 40 mm deep, 2 mm diameter strain gauge holes in both horizontal and vertical positions. Picture shows horizontal drilling of strain gauge hole in preloaded bolt (bolt preload applied / measured in Skidmore-Wilhelm bolt tension measurement device) in one of the initial steps of proof-of-concept sequence carried out in the Stevin lab.



Lab experiments were carried out to verify correct measurements results of strain gauges that were glued in with the bolt axis positioned in both horizontal and vertical direction.

Correct linear behaviour and consistency (repeatability) of the strain gauges was confirmed by calibration tests for all execution alternatives (horizontally / vertically drilled strain gauge holes, strain gauges installed in preloaded/non preloaded bolts).



A348 Highway bridge (1974) near 'De Steeg', The Netherlands that was used to carry out the in situ tests



Main girders of the bridge. The girders are built-up sections with varying height and plate thickness of web and flanges. Steel grade S355. In the time the bridge was built faying surfaces of HSFG connections were coated by aluminium spray metallizing ('AL geschoopeerd').



Overview of HCFG bolted splice joint in lower flange of the girder that was used to test the system. Bolt pattern consists of $4 \times 21 = 84$ bolts. 16 of these were instrumented in situ.



Detail of the joint the method was tested on. Measurements were carried out on M24, HV10.9 bolts with a total clamp length (washers + steel plates: $4+24+32+24+4$) of 88 mm.



Top view of flange prior to the in-situ measurements. All 8 bolts on each side of the web that were instrumented were attached to data acquisition systems simultaneously. This way it was possible to monitor interaction between the bolts during (un)tightening.



Prior to drilling of the hole the coating on/around the bolt head was removed. Picture shows bolt head with protruding stain gauge insertion pipe and glued on wire terminal.



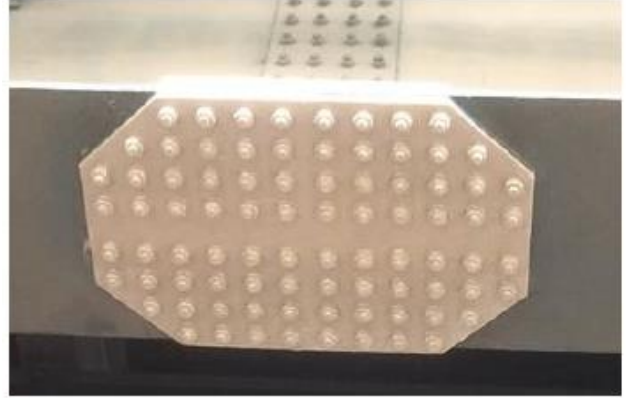
Detail picture of wire terminal on bolt heads. To minimise influence of temperature variation and cabling on the strain gauge measurements, a full Wheatstone bridge was integrated on each wire terminal. High precision micro resistors were used to create the bridge. Stevin report 6-18-03



During the experiment the bridge was open for traffic. To ensure structural safety, directly after untightening a bolt, it was removed and replaced by a new bolt. This way the reduction of the resistance of the joint was restricted to only one bolt.



The new bolts were immediately pretensioned. The combined method was used to apply the preload



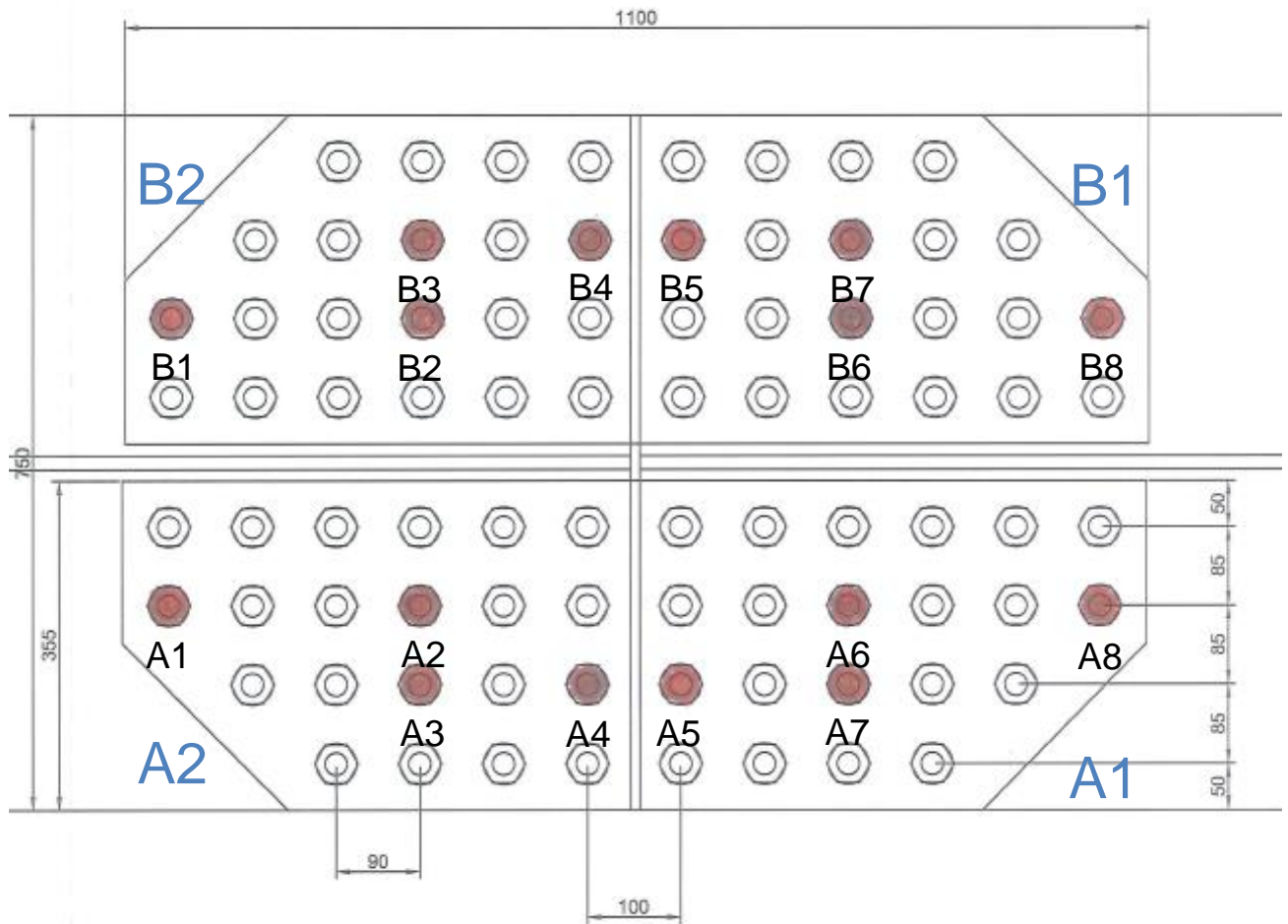
Connection after restoration of the coating system.



The 16 bolts that were removed from the connection were taken to the laboratory to verify linearity of the strain gauge signals and to determine the pretension that was present before untightening



In the laboratory all bolts were loaded to determine the tensile force that corresponded to the strain measured during the in-situ untightening of the bolt. Linearity of all strain gauge signals was checked and confirmed. Finally the test setup was used to determine the tensile strength of the bolts.



The joint consists of 4 equivalent quadrants (A1, A2, B1 and B2). In each quadrant 4 out of 21 bolts were instrumented to perform the in-situ measurements. The bolt IDs are given in black, the quadrants in blue. Dimensions in mm.

Appendix B: Calculation sheet

Bolts	Experimental Bolt Microstrain (-)	Experimental Residual Bolt Force (kN)	Nominal Bolt Stiffness (N)	Theoretical Bolt Microstrain (-)	(CF) _{act}	(CF) _{nom}
A1	3239	280	9,50E+07	2947,3	0,0864	0,0950
A2	3346	335	9,50E+07	3526,3	0,1001	0,0950
A4	3579	346	9,50E+07	3642,0	0,0967	0,0950
A5	3502	330	9,50E+07	3473,6	0,0942	0,0950
A6	3625	331	9,50E+07	3484,1	0,0913	0,0950
A7	2104	189	9,50E+07	1989,4	0,0898	0,0950
A8	3754	344	9,50E+07	3621,0	0,0916	0,0950
B1	3466	315	9,50E+07	3315,7	0,0909	0,0950
B2	3385	305	9,50E+07	3210,5	0,0901	0,0950
B3	3531	315	9,50E+07	3315,7	0,0892	0,0950
B4	3537	310	9,50E+07	3263,1	0,0876	0,0950
B6	3676	323	9,50E+07	3399,9	0,0879	0,0950
B7	3074	277	9,50E+07	2915,7	0,0901	0,0950
L1*			9,50E+07		0,0833	0,0950
L2*			9,50E+07		0,0917	0,0950
L3*			9,50E+07		0,0847	0,0950
L4*			9,50E+07		0,0868	0,0950
L5*			9,50E+07		0,0836	0,0950
L6*			9,50E+07		0,0894	0,0950
L7*			9,50E+07		0,0845	0,0950
L8*			9,50E+07		0,0869	0,0950
L9*			9,50E+07		0,0837	0,0950
L10*			9,50E+07		0,0892	0,0950
L11*			9,50E+07		0,0848	0,0950
L12*			9,50E+07		0,0867	0,0950
Average					0,0889	0,0950
Standard deviation					0,00411	0,0

Mean value correction of $\beta = 0,935$ ←

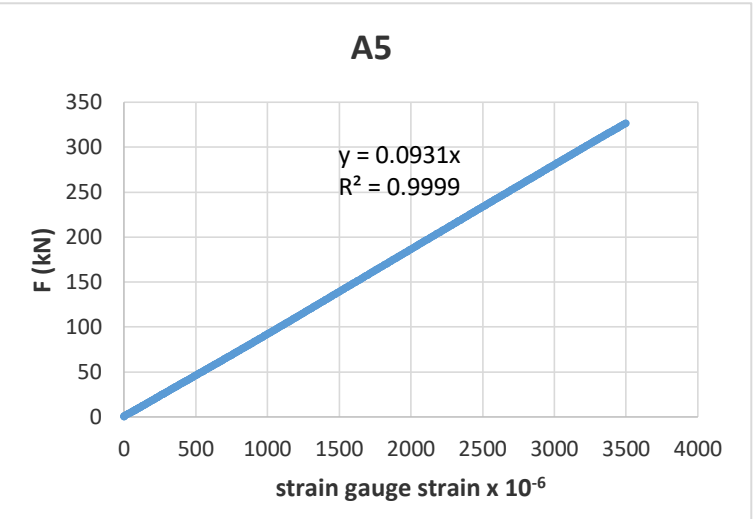
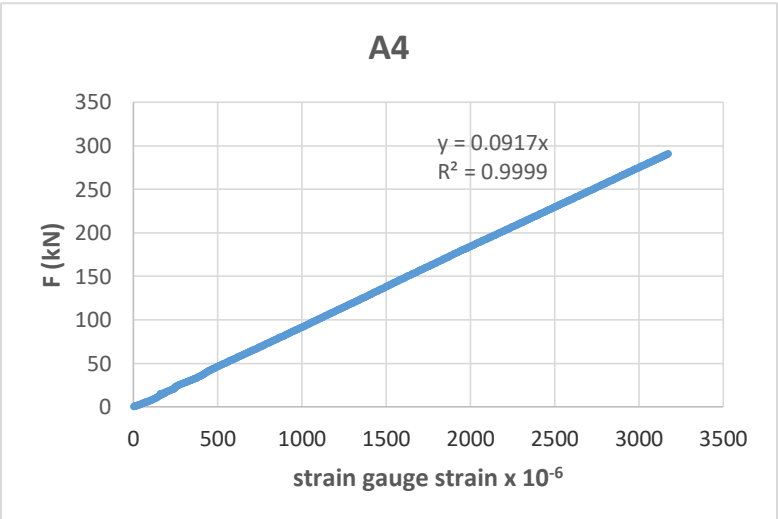
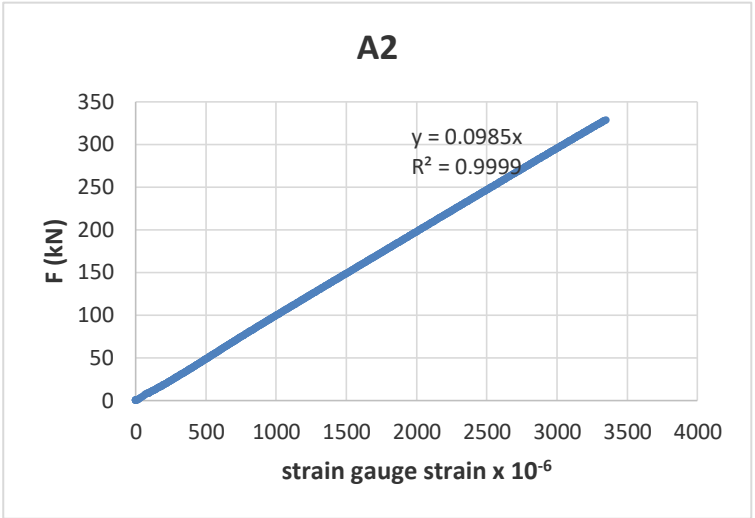
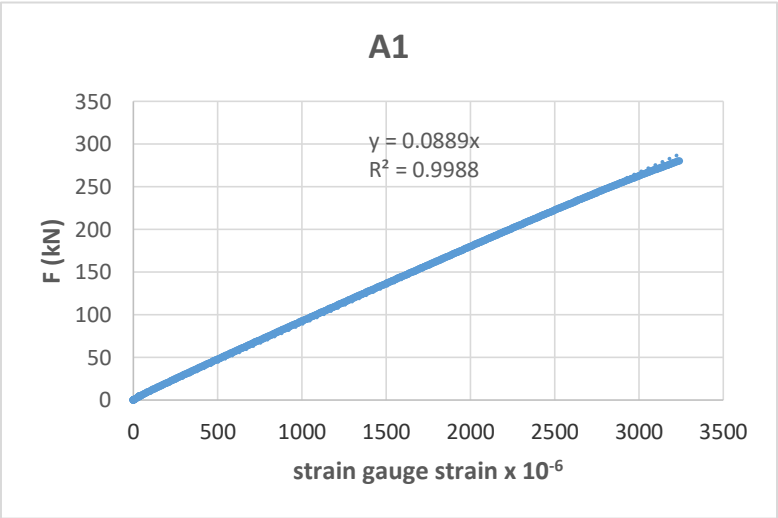
L*: Bolt tested in laboratory only in either of the four conditions during installation of strain gauge

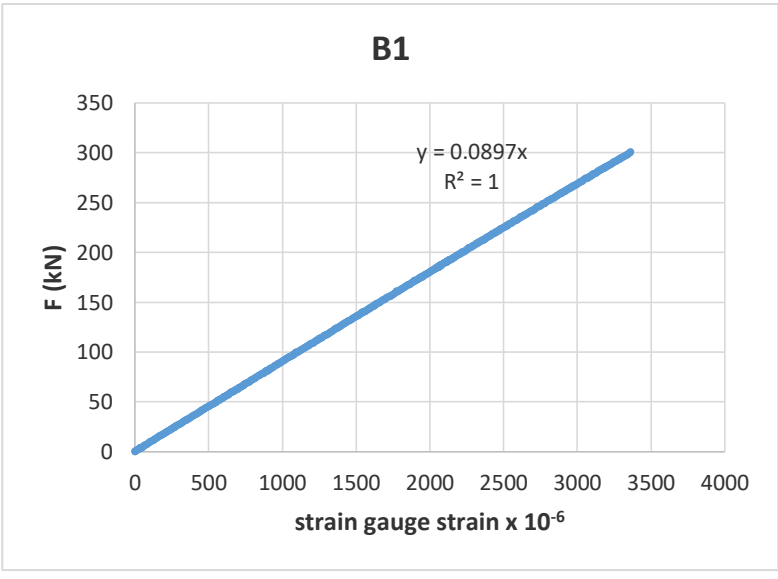
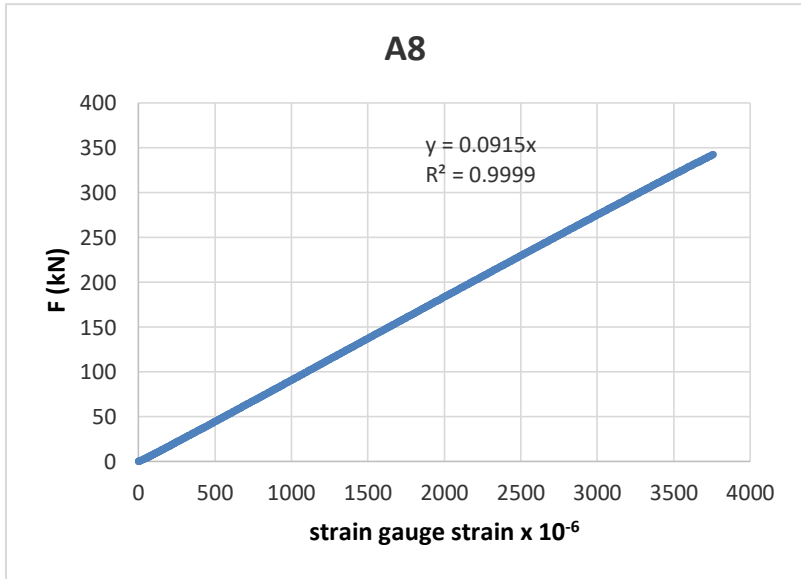
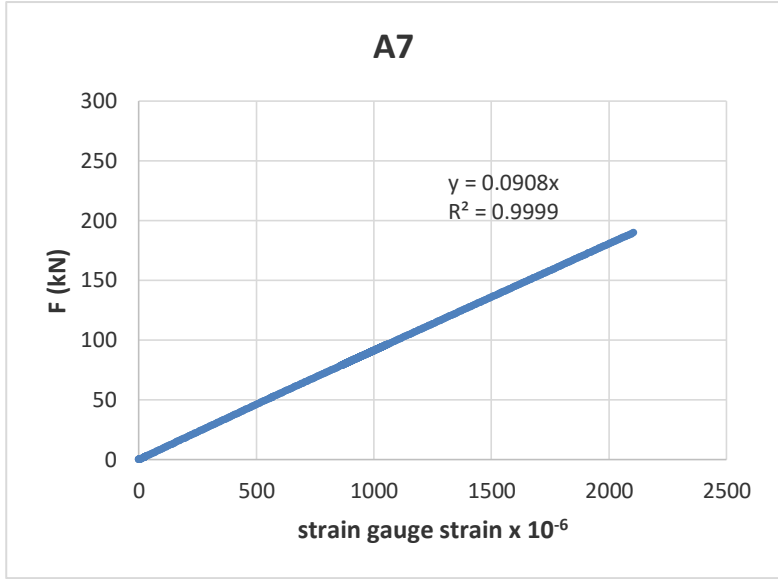
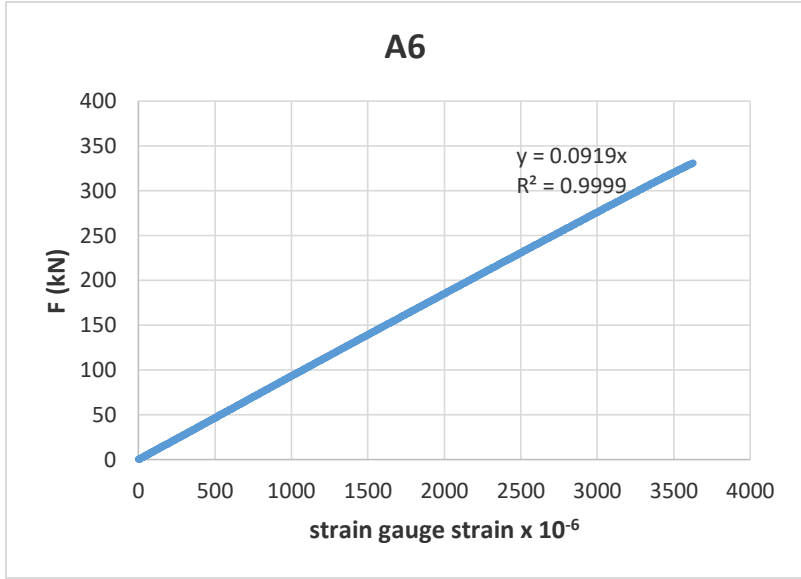
$$(CF)_{nom} = (EA)_{nom} = E \cdot \frac{1}{4} \pi d^2$$

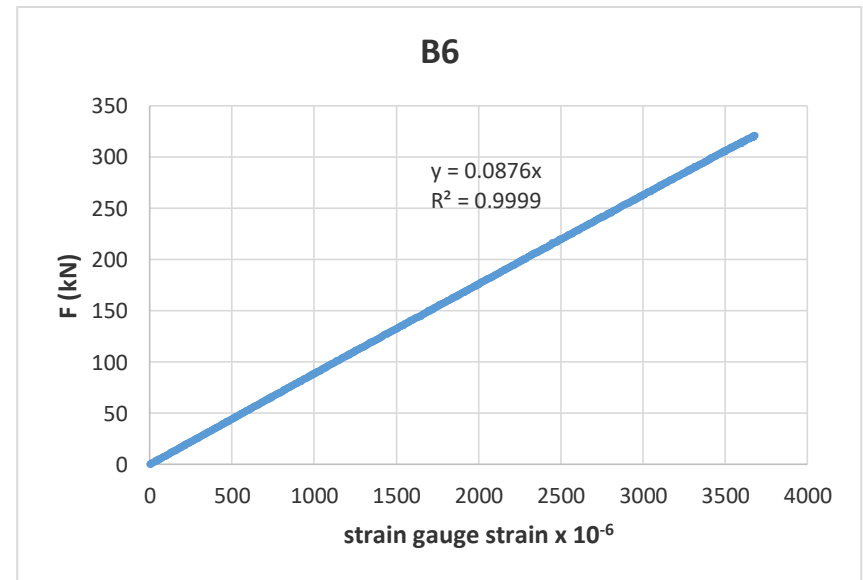
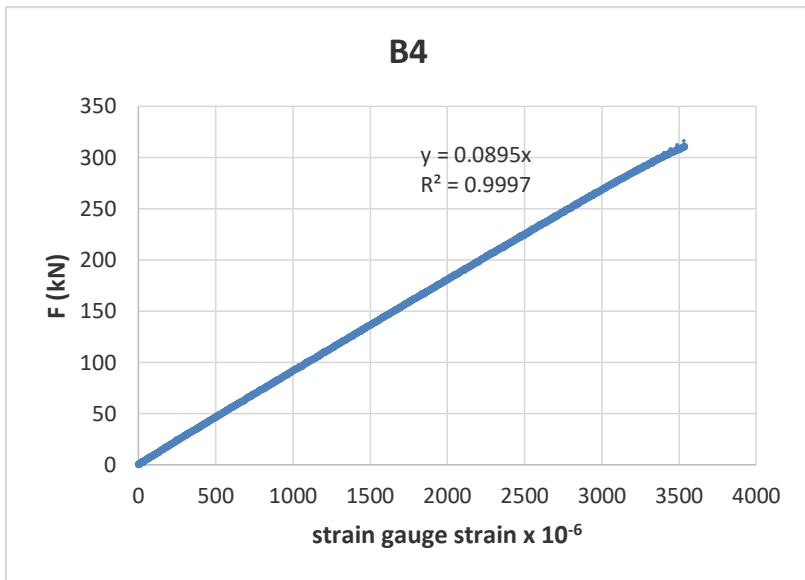
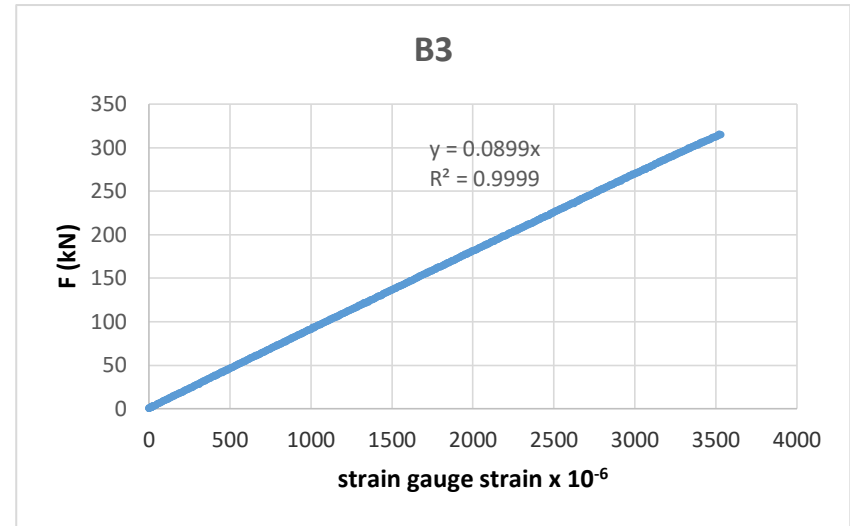
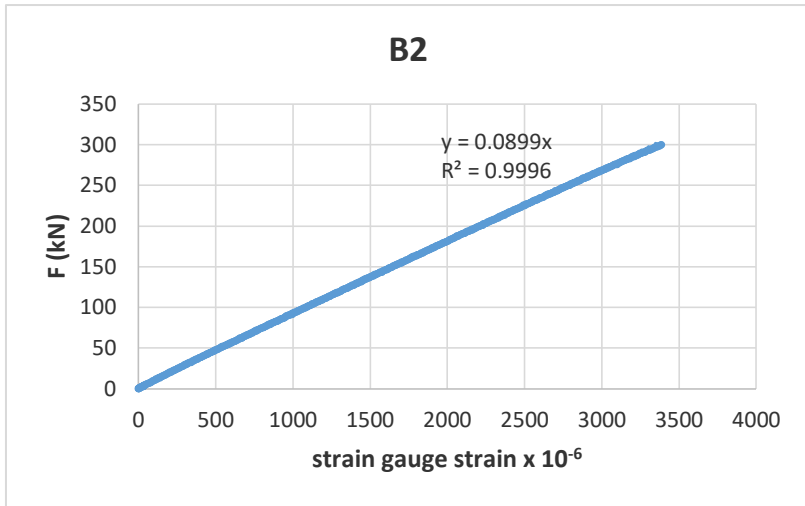
$$(CF)_{act} = \beta \cdot (CF)_{nom}$$

$$(CF)_{ch} = (CF)_{act, \mu} - 1,711 \cdot \sigma \quad (n = 24)$$

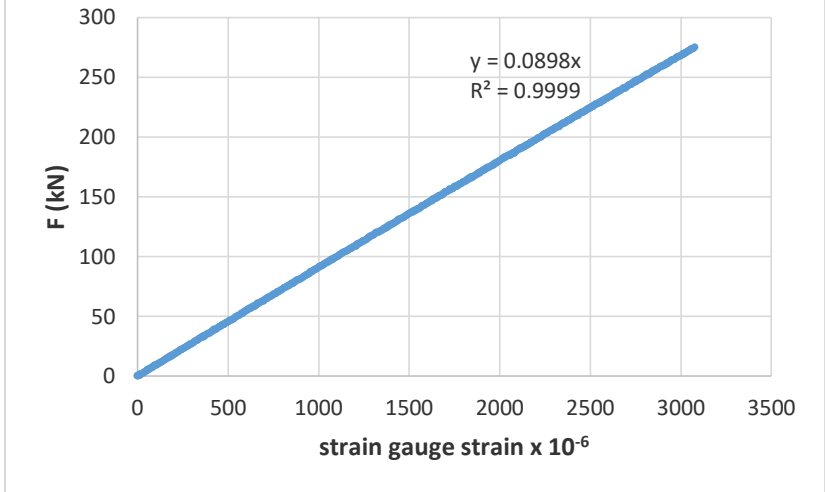
Appendix C: Calibration of Bolts from Middachter Bridge





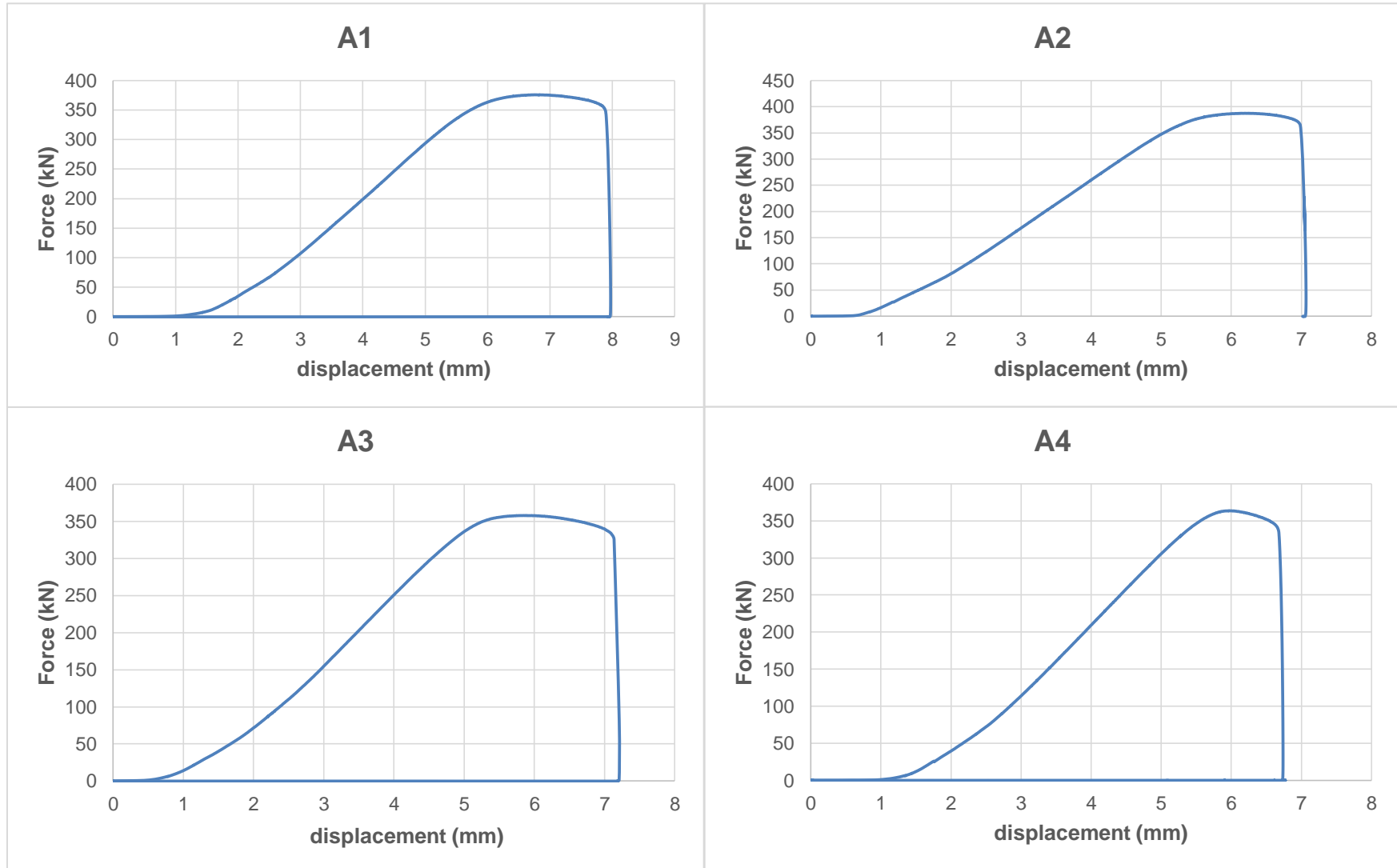


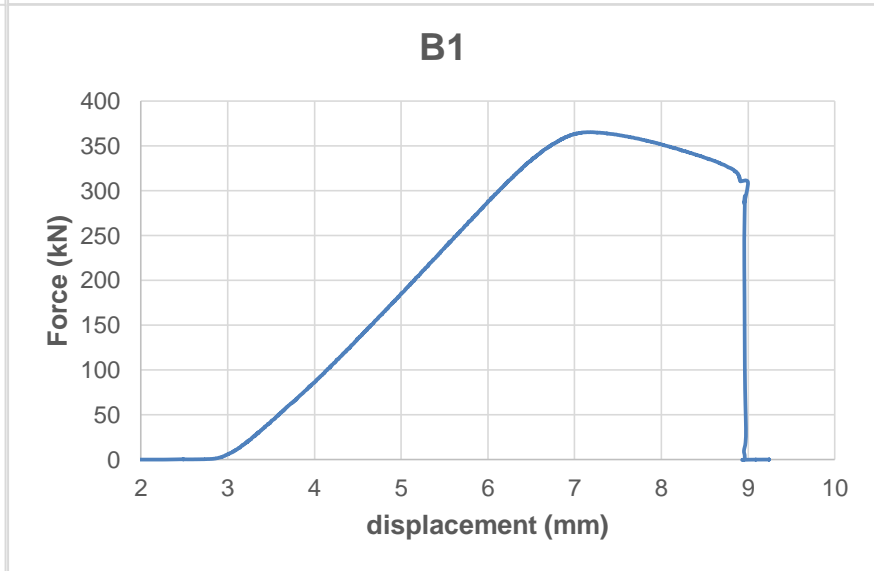
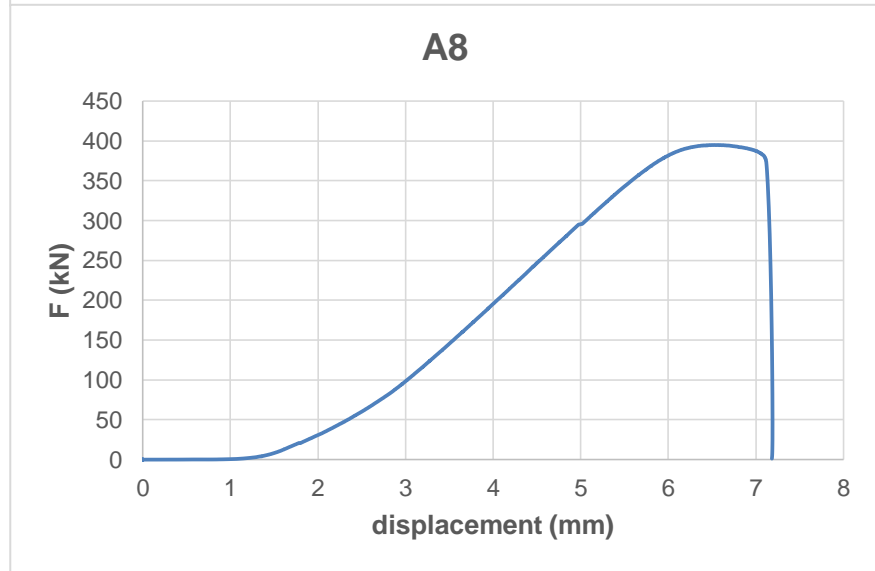
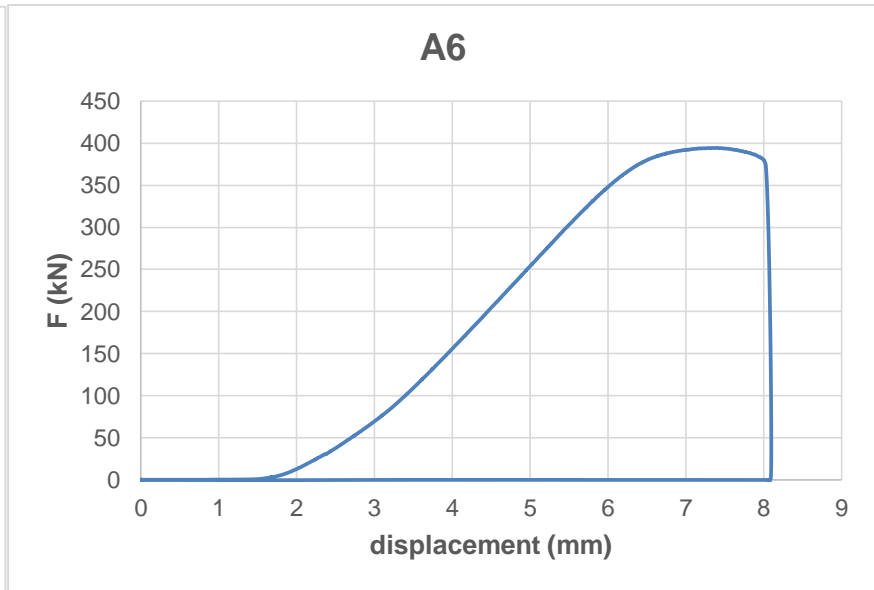
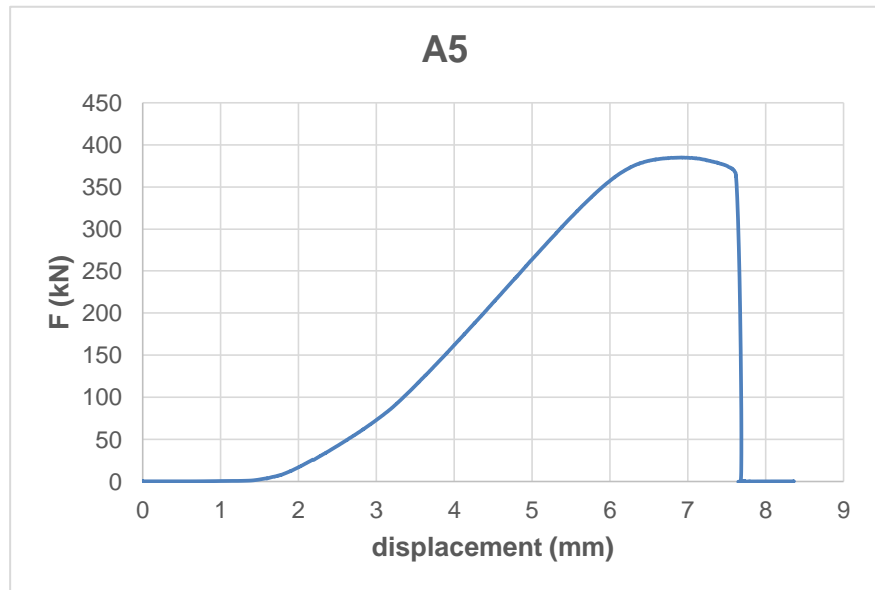
B7

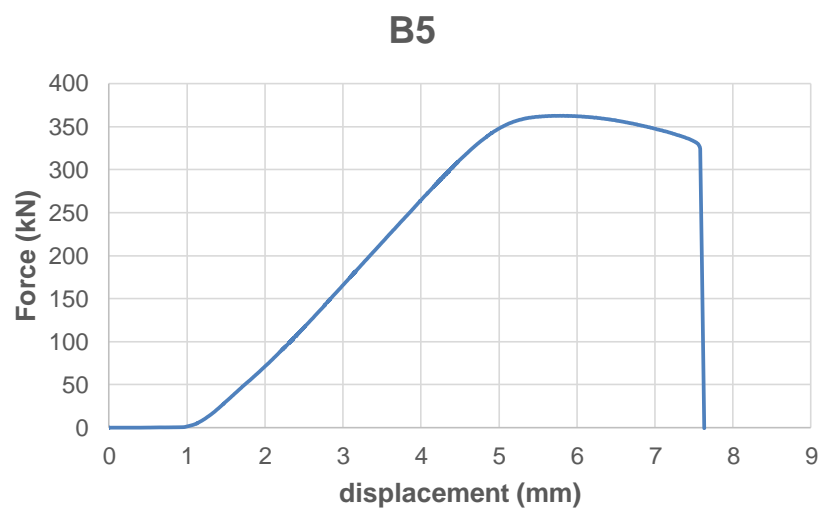
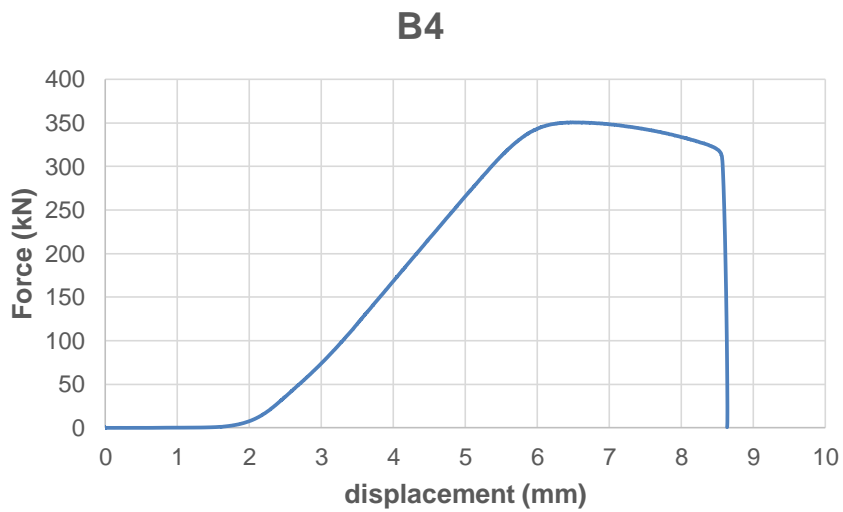
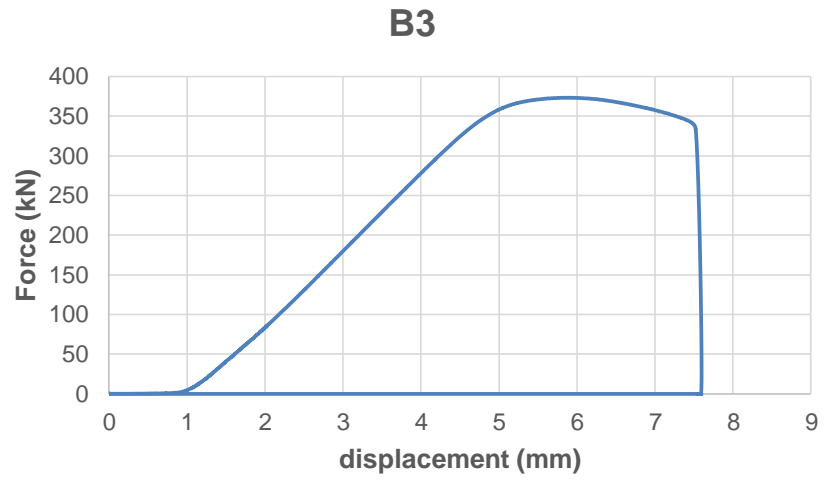
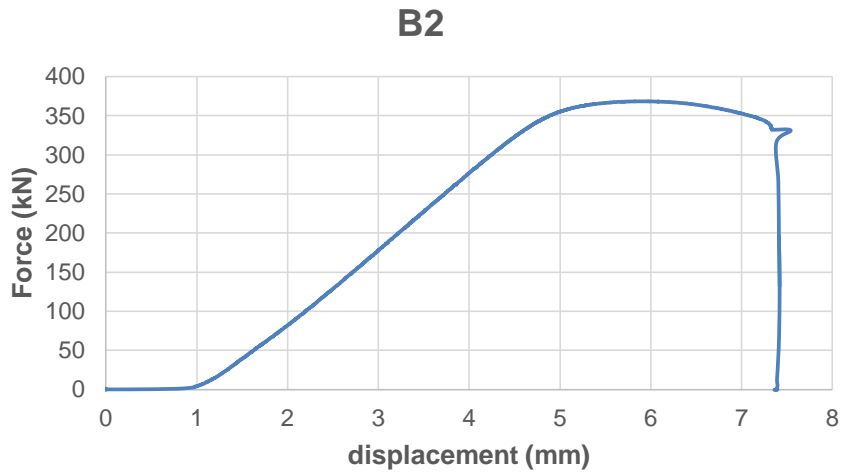


Appendix D: Determination of Resistance of Bolts from Middachter Bridge

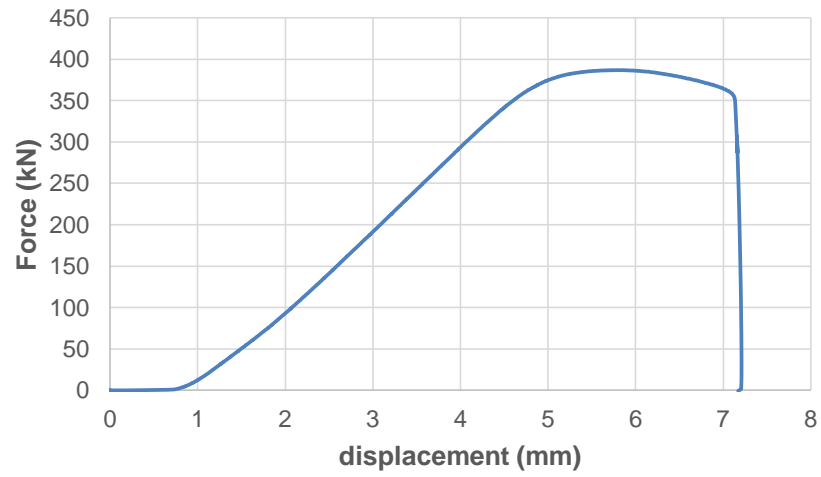
Note: the displacement is that of the actuator.



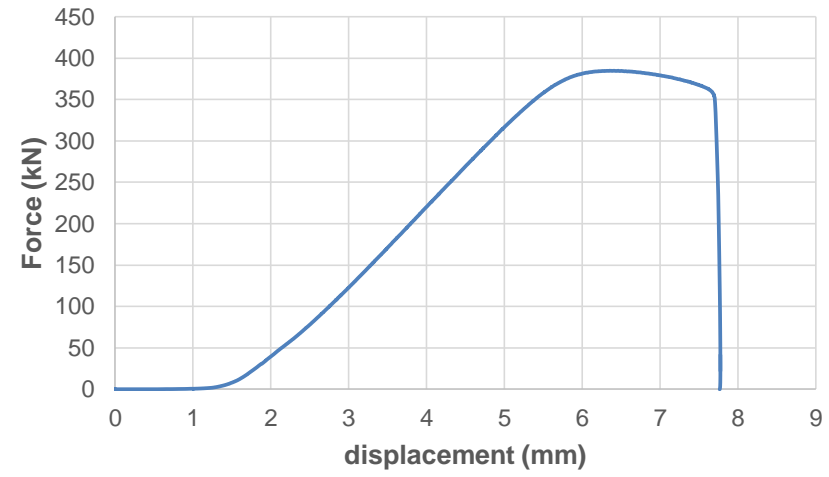




B6



B7



B8

