Design of a CFRP-to-steel joint for a bus engine mount and experimental testing of joint relaxation

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Abstract
The objective of this study is to investigate aspects of hybrid joint design for a shear connection. Part 1 is design considerations and a case study based on a engine mount. In Part 2, an experimental study is conducted regarding how relaxation and creep affects properties of a hybrid joint between carbon fiber vinylester and steel. Failure modes, loss of clamped force and ultimate strength of the joint and how these are affected by elevated temperature and different clamped forces are presented. Test specimens were 184 × 45 × 5 mm CFRP, in double lap joint between two steel plates of same dimensions, with either a single or double M10 hexagon flange screw. Creep/relaxation testing was preformed over a 28 day period. Results shows that the amount of initial clamped force does not effect amount of loss of clamped force, but increase in temperature significantly does. Average loss was between 8.3 an 8.8% for single joint specimens, initial torque of 28, 37 and 46 Nm, and 10.8% for double joint specimens, 37 Nm. But losses for specimens subjected to heat, 80 °C, was 35.5%. Ultimate bearing strength of the lamina was not effected by any circumstances. Slip load was lower for specimens clamped with lower clamped force, the loss over the 28 day period was not enough to produced any clear results regarding the amount of loss. But tendencies point towards a lowered slip load of the same magnitude as the loss of clamped force i.e around 8-9%. Slip load for specimens subjected to 80 °C did not decreased, even though the clamped force had been lowered. All specimens fail in bearing, single bolted joints had a far more progressive failure than the double joints, which failed catastrophically.
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Part I
Joint design

This part is divided into 3 smaller parts, where the first gives background, some terminology, definitions and presents the basics of viscoelasticity in composites. Second part deals with a simplified engine mount case study, resolving the mathematical approach to screw joint design under static loading. Lastly part 3 covers other considerations of interest when designing mechanically fastened joints.

1 Background

In 2011 a bachelor thesis was conducted by 10 students from Lightweight Structures at KTH in collaboration with Scania Buses. The project was to design, manufacture and evaluate a carbon fiber vinylester C-beam which in theory where to be used as part of the load carrying structure. The objective was to design a beam which at least were to meet the requirements of the beam in current use, while lowering it’s weight. The topic of how to integrated the beam with other structural parts and/or other components was only briefly touched upon. However the requirement of an ability to remove components, decreases the options to the use of either bolted or screw joints.

1.1 Basics considerations of mechanical fastening

- Clamping force: The axial force produced by the fastener, compressing the joined materials.
- Clamping length: The distance between the bottom of the screw head and the first threads engaged.
- Assembly torque: The controlled torque in order to generate a specified amount of clamped force.
- Friction: Critical parameter of fastener design, divided into several groups:
  - Assembly friction plays a significant part of the tightening torque during installation. According to SFN\textsuperscript{1} up to 90\% of the inflicted torque is consumed by overcoming the assembly friction meaning that only 10\% is used to create the clamped force.
  - Glide- or static friction is defined as a constant when to materials aren’t moving with respect to each other. It’s also the amount of clamped force which is transmitted in shear force i.e it contributes to the shear strength of the joint.
  - Dynamic friction is defined as a constant when to materials are moving with respect to each other. It’s generally dependable of the velocity of the moving parts.
- Relaxation: Materials succumbing to high stress on a microscopic level. The uneven surface areas of materials creates contact points were stress levels area elevated. These points will eventually start to yield resulting in a lowered stress level.

\textsuperscript{1}Swedish Fastener Network. Numbers represent a screw joint design

3
• Settling: Important issue to address when designing a bolted/screw joint and it is the result of relaxation in the material. The affect is also a lowered clamped force. Basically settling has two parts, first one being static settling which occurs when the bolt or screw is installed and the second one being dynamic settling which occurs ones the joint has been subjected to load. Typically settling decreases with time, however if one of the involved components should start experiencing creep the loss of clamped force will continue to progress. Settling generally occurs at places being subjected to larger stresses.

• Creep: A continuous settling of the material. A result of a viscoelastic behavior of the suppressed material.

1.2 Friction grip joints vs composite joints

To start of, the concept of friction joint design vs the common way of designing composite-to-metal shear joints, needs to be cleared out. High Strength Friction Grip or HSFG joints prevents materials from sliding with respect to each other. External forces will never exceed the friction resistance during service. This requires a large clamping force. For composites, a larger force transverse to the fiber direction could potentially lead to delamination, i.e. a broken bond between two or more layers in the lamina, or to relaxation and/or creep. Creep leads in turn to a continuous loss of clamped force. As a result, low clamping force is used which also results in low shear. This method demands that the stresses created at the side of the hole are well known.

1.2.1 The friction joint

A friction joint is designed with the purpose of generating enough clamped force on to two or more materials and thus preventing them from sliding in between one another. The clamped force, \( F_{\text{clamp}} \) and the coefficient of friction (cof), \( \mu \) governs the maximum shear load the joint can withstand, \( N \) according to

\[
N = \frac{F_{\text{clamp}}}{\mu}
\]  

(1)

The principle is that the screw is extended, much the same as a spring being stretched, the further it’s pulled, the harder it gets. This type of joint demands that the clamped materials do not yield due to the induced stresses, it’s popular to use when joining metal-to-metal.

1.2.2 The composite joint

Composite material have sufficient stiffness, strength and fatigue properties but as earlier stated, the weak-spot is the through-thickness strength and stiffness[1] but the non-isotropic properties creates constraints which do not apply for metal joints. Composite material has a set of common failure modes for tensile failure, seen in fig (1). These are: net-section (a), bearing (b), shear-out (c) and bolt failure (d). The most preferable failure is bearing failure, it progresses more slowly than others. To achieve it, generally a small aspect ration between \( D/w \), fig (2) and a larger between clamped force and carried load, is to prefer. The opposite to those ratios i.e \( D/w \) is large and \( F_{\text{clamp}}/N \) is small results in net-section failure. Shear-out happens if the lamina contains to many 0\(^\circ\) layers and if the stresses at the edge of hole are too large. This usually happens if width of the laminate or the space in between
holes $e$ is to small compared to the hole diameter. Bolt failure occur if the bolts diameter is to small compared with the laminate thickness.

![Figure 1: 4 most common failure modes in composite joints](image)

It’s been established that the key aspects are low clamped force and bearing failure. Maximizing bearing failure load means lowering bearing stresses. Low bearing stresses are also important since the highest tension stress alongside a loaded bolt i.e the net-section, is approximately equal to the average bearing stress given that the behavior is linear elastic [1]. See figure (3). However this is really only true for single/ or single-row joints or when the pitch, $e$ is considerably large. In multi-row joints the individual bearing stresses are lower, resulting in the strongest possible failure mode, per unit laminate width, being net-section followed by bearing failure [2].
Bearing- and net-section stresses are considered to be independent for ductile metals (a commonly used approximation is that the net-section tension strength through the holes is equal to bearing strength of the bolted hole) but this cannot be said regarding composites. For composites, the concentrated net-section stress around the hole does not become negligible before the largest possible load is achieved, due to yield.

![Stress distribution at loaded bolt hole, elastic isotropic plate](image)

Figure 3: Stress distribution at loaded bolt hole, elastic isotropic plate

As for tapered head vs. counter sunk fasteners. Hart-Smith debates that the counter sunk fastener reduces the effective bearing area. “Any contribution from the countersunk head should be totally ignored. For typical single-shear installations, the fastener head cannot make contact to transmit a bearing load until after the shank area has failed in bearing”. Hart-Smith refers to the bearing-bypass curve, fig (4) which shown that higher bearing stresses must lead to reduced strain levels in the laminate. Caccese with others [3] tested protruded vs tapered head bolts affect on the clamp-up load on E-glass/ vinylester laminates and found no advantages for using tapered instead of protruded heads.
Park [4] preformed a experimental study (using an acoustic emission technique) on the effects of stacking sequence and clamping force on the bearing strength of bolted joints in carbon fiber epoxy composites. He compared lamina with outer layers in 0° and 90° directions and rather predictably found a small difference in ultimate bearing strength, but delamination bearing strength was twice as high for lamina with outer layers in the 90° direction than for those in 0° direction. Noteworthy is that the bearing strength in turn is about twice as large as the delamination strength.

As for clamped force, the bearing and delamination strength both increases with an increased clamped force up to a saturation point. Increased clamped force actually postponed delamination rather than causing it to occur more rapidly, but of course only up to a certain point. The saturation point is around 5 MPa regardless of stacking sequence and even at 58.6 MPa, bearing strength is only marginally increased.

M. Johnson and F.L Mattehews [5] tested single hole bolted joints in E-glass/ epoxy and polyester laminates and found that cracks were beginning to propagate around the hole once the hole had elongated 0.4 % of it’s original shape.

Caccese [3] with others investigated the loss in clamp-up load after 2000 [h] of relaxation response for single lap, bolted joints in E-glass/ vinylester with a fiber volume fraction of 0.51. Tests were done both with composite-to-aluminum and also compared to results by Weerth and Orlof [6] for composite-to-composite. The losses where up to around 40 % for composite-to-aluminum and 55 % for composite-to-composite. Caccese and co. also preformed block compression tests in order to isolated the creep response from the composite. They found an average loss of 13% after 2000 [h] and explained the lower creep rate with the uniformed load being responsible for the reduction. Stälberg [7] makes another point regarding this when testing viscous effects in coating layers and contributes the significance of the boundary conditions i.e the fact that the material has less physical space to go to.

As for re-tightening. Caccese [3] found only one re-tightening could have a clear positive effect on the joints loss of clamped force.

A list of rules of thumb for composite joint design provided by Swerea SICOMP is available through the SFN Handbook. It regards ratios between the fastener diameter and the span of the laminate, stacking sequences, bolt sizes and more.

Figure 4: Bearing-bypass design chart for bolted composite joints
1.3 Viscoelasticity

Viscoelasticity means the material possesses both viscous and elastic properties. Viscosity defines as a substance inner friction, or the resistance it has to withstand shear or tensile stresses. It has a linear relation between stress and strain over time. Fluids are often considered viscous. Elastic properties consider materials which after deformation regains its original shape. Metals are often considered elastic. Combing these and the property of time dependent strain, is achieved.

Two phenomenons associated with viscoelasticity are: with constant stress, the strain will increase with time and with constant strain, stress will decrease with time, i.e. creep and relaxation. This behavior is show graphically in figure (5).

![Figure 5: Creep/ Relaxation and recovery for viscoelastic material](image)

There is also hysteresis which is the dissipated energy due to plasticity. Popularly this is shown with a stress-strain curve, where the hysteresis is the integrated area in between load- and unloading curve.

A few models are used to described viscoelastic behavior in order to predict their response during loading. These all consist of a combination of springs and dampers, but these are arrange differently, giving each model individually pros and cons. First of is the Maxwell model, which consists of a viscous damper and a viscous spring, connected in series. The model is quite accurate predicting relaxation for most polymers, but not so much for creep. The Kelvin-Voight model on the other hand, predicts creep well but leaves out relaxation. By combining these two, the Standard linear solid model is received. The model is more accurate then the other two but is rather difficult to calculate. Schematics for these models is seen in figure (6).
1.4 Viscoelastic model and loss of clamped force

For analytical approaches Scharperry [8] developed a constitutive model to described nonlinear viscoelastic material (2).

\[ \varepsilon(t) = g_0 D_0 \sigma^t + g_1 \int_0^t \Delta D(\psi^{\tau} - \psi^t) \frac{d(g_2 \sigma^{\tau})}{d\tau} d\tau \]

(2)

where \( D_0 \) is the instantaneous uni-axial elastic compliance, \( \Delta D \) is the uni-axial transient compliance, \( g_0, g_1, g_2 \) are viscoelastic parameters. \( \psi \) is the reduced-time and is given by

\[ \psi(t) = \int_0^t \frac{d\xi}{a_{\sigma}^{\xi} a_T} \]

(3)

where \( a_{\sigma} \) acts as a time scaling factor and \( a_T \) is a temperature dependent that is used to define a time-scale shift factor for thermorheologically simple materials [9]. These rather convoluted expressions has been developed further by others and are the basis of all non-linear viscoelastic models used. Based on this Shivakumar and Crews [10] developed a commonly used expression (4) for loss of clamped up load with respect to time.

\[ \frac{P_t}{P_0} = \frac{1}{1 + F_1 a_{\alpha_{th}} t^n} \]

(4)

where \( P_t \) is the clamped force at time \( t \), \( P_0 \) is initial clamped force, \( F_1 \) is a material dependent parameter, \( \eta \) the viscoelastic power law constant and \( \alpha_{th} \) a hygrothermal shift factor to account for temperature and/or moisture. This model is popularly used in FE-modeling. Roberts [11] used this model when Swerea Siconp investigated stress relaxation in composite joints as well as Thoppul with others [12]. The expression comes from a series of assumptions regarding viscoelastic behavior. Expression (4) is then rewritten as

\[ \log \left[ 1 - \frac{P_t}{P_0} \right] = n \cdot \log(t) + \log \left( \frac{F_1}{\alpha_{th}^{n}} \right) \]

(5)

Linear regression analysis is used to find the unknown parameters.
2 Design the joint

Popularly, the Ishikawa diagram seen in fig (7) is used to describe the large number of design considerations dealt with when designing a friction joint for use for production purposes and also highlights the complex relation between applied torque and resulting clamped force. The method presented in this work considers the manufacturing and production approach currently being used at Scania. According to Scania standard for clamped force and friction, STD3950, the allowable region for the friction is between 0.10-0.16.

![Ishikawa Diagram](image)

Figure 7: The Ishikawa diagram

2.1 Load case

A static load case of 2.4 kN distributed equally on 4 different engine mounts, see fig 8, was concluded from the bachelor thesis report. To simplify, i.e exclude the need of FE-modeling, each mount consists of 4 joints symmetrically placed to each carrying an equal amount of the load as seen in fig (9), thus resolving in each joint supporting 600 [N].
4. Utredning av lastfall med teknisk balkteori

4.1. Metod för utredning av lastfall med teknisk balkteori

4.1.1. Lastfall för teoretisk analys

Arbetet inleddes med att förenkla det givna lastfallet erhållet från Scania. Komplett information om givet lastfall ges i bilaga A. Belastningen från motorn tas delvis upp av två tvärgående balkar, se figur 4.1. Efter remiss hos Scania, på förslag av Peter C-Eriksson bestod förenklingen i att "vrida" ut de två tvärgående balkarna och ersätta dem med en momentfjäder med korrekt fjäderkonstant, se figur 4.2. Egenskaperna för denna momentfjäder bestämdes genom analys av nedböjningen för befintlig konstruktion. Den befintliga konstruktionen har enligt uppgift från Scania en statisk nedböjning på 20 mm. Teknisk balkteori, där elementarfall (5)1 och 2 har kombinerats, användes för bestämning av fjäderkonstanten hos denna rotationsfjäder. Formlerna för beräkning av denna illustreras i bilaga C.

4.1.2. Designlösningar

Utöver problemformuleringen enligt bilaga A fanns ytterligare geometriska begränsningar, dessa grundar sig på information från Scania. Ett krav som inte finns med i bilaga A och som styr mycket av hur ramen ser ut är kraven på instegslösningar, trappstegshöjder, trappstegsdjup, golvlutningar med flere. Då dörr och trapp monteras bakom bakaxeln vill man ha så mycket vertikalt utrymme som möjligt, detta beror bland annat på lagkrav i vissa länder där stadsbussarna säljs. Det medför att balkarna bör vara 100 mm lägre än ovankant tvärbalk precis bakom bakaxeln, se figur 4.3. Det är den lösning som representeras i bilaga A. Vid beräkning av styvhet med hjälp av teknisk balkteori grundar sig modelleringen i en rak balk utan vertikal höjdvariation, d.v.s. utan "krök".

Figure 8: Forces acting on the engine mount

Figure 9: Load case

2.2 Determining torque

Definitions of screw and bolt measurements are found in appendix A. Following expression (1), the clamped force needed \( F_{\text{req}} \) to support a set load is defined in equation (6)

\[
F_{\text{req}} = \frac{F_Q}{\mu_{\text{slip}} \cdot q_{\text{tr}} \cdot \eta}
\]  

(6)

where \( F_Q \) is the static load, \( \mu_{\text{slip}} \) is the coefficient of friction (cof) between the clamped materials, \( q_{\text{tr}} \) is the number of load transferring surfaces and \( \eta \) is the safety factor. A general friction joint settles about 20-50\% after assembly.

Torsion stresses are generated due the applied torque and lowers the tension stresses (7) resulting in a reduced tension stress \( \sigma_V \).

\[
\sigma_V = \frac{\eta_{\text{fail}} R_{p0.2}}{\sqrt{1 + \frac{4}{1 + d_3/d_2} \left( \frac{P}{\pi d_2} + \frac{\mu_G}{\cos 30^\circ} \right)^2}}
\]  

(7)

where \( \eta_{\text{fail}} \) is a safety factor, here set as 1 since safety already is concerned with in equation (6). \( R_{p0.2} \) is the yield strength, \( d_3 \) and \( d_2 \) are defined in Appendix A, \( P \) is the pitch and \( \mu_G \) thread friction.
This reduces the available clamped force $F_{ava}$ according to (8), ISO 898-1.

$$F_{ava} = R_{p0.2} \cdot A_s$$  \hspace{1cm} (8)

where $A_s$ is the nominal stress area in $[\text{mm}]^2$ (table value). To generate the available clamped force the responding moment is calculated using (9).

$$M_A = F_{ava} \left( \frac{P}{2\pi} + \frac{\mu_G d_2}{2\cos30^\circ} + \frac{\mu K D_h}{2} \right)$$  \hspace{1cm} (9)

where $D_b = \frac{d_w + d_h}{2}$, $d_w$ is the outer diameter of the contact area between bolt (or washer) and material and $d_h$ is the nominal hole diameter. Recalculating (9) using the required clamped force of course results in the required torque. The available torque obviously must be larger than the required, otherwise screw size and/or screw strength needs to be increased.

Due to torque spread during assembly, the resulting torque may vary up to $\pm 17\%$. Adding to the previous allowable region for the cof of 0.10-0.16, the actual applied torque differs a lot, see figure (10). Thus the lowest applied torque with spread, must be equal to the required torque.

### 2.3 External loads

The joints ability to endure the affects of external loads is mainly depending on the clamping length and the stiffness ration between screw and clamped material. This is usually described using a Force-Deformation chart, as seen in figure (11). The angle of the orange line resembles the stiffness i.e the spring constant of the screw and the shorter green line, that of the clamped material. As the screw is tightened, it elongates and the clamped material is pressed together, thus the different deformation directions. At the intersection is the maximum clamped force.
As settling occurs, the clamped material is further compressed, meaning that the intersection is move further in towards origo, leading to a decreased clamped force. For the joint not to separate, the external force must remain less than the clamped force.

To avoid settling either:

- Make the screw less stiff either by increasing clamped length or decreasing Young’s modulus of the screw. A less stiff screw enables it to have a larger deformation and it is thus not as sensible to settling. This is presentable in figure (11) as a lower sloped green (screw) line.
- Increase the contact area between screw and material, i.e washer thus lowering the induced stresses onto the material.

2.4 Spring constants

In order to predict the safety towards settling, as well as calculating the clamped force it’s important to know the stiffness of both screw and clamped material. This is done by considering them to be springs where there stiffness is resembled by a spring constant. Using Hooke’s law and the constitutive equation for stress and strain, the spring constant $k$ is defined as

$$k = \frac{EA_i}{L_i}$$  \hspace{1cm} (10)

where $E$ is the Young’s modulus, $A_i$ is the area and $L_i$ is the length.

2.4.1 Screw

Mythology for determining the spring constant for the screw is as follows. Figure (12) shows the different geometries of the screw.
Spring constant for the screw stem, $k_S$

$$k_S = \frac{E_b A_d}{L_S}$$  \(11\)

where $E_b$ the Young’s Modulus of the bolt $L_S$ is the length of the stem and $A_d = \frac{\pi d^2}{4}$.

For the threads which is engaged, $k_{en}$

$$k_{en} = \frac{E_b A_{d_3}}{L_{en}}$$  \(12\)

where $L_{en}$ is set to $0.5d$, $d$ being the screw diameter and $A_{d_3} = \frac{\pi d^2}{4}$.

The other parts follows the same pattern:

$$k_{core} = \frac{E_b A_d}{L_{core}}, k_K = \frac{E_b A_d}{L_K}, k_G = \frac{E_b A_{d_3}}{L_G}$$

$k_{core}$ is the part of the core engaged, $k_K$ is screw head and $k_G$ is the threaded part of the screw within the clamping length.

The resulting spring constant for the entire bolt is summation of the different parts according to

$$\frac{1}{k_b} = \frac{1}{k_s} + \frac{1}{k_G} + \frac{1}{k_{en}} + \frac{1}{k_{core}} + \frac{1}{k_K}$$  \(13\)

Elongation of the is the acquired with (14)

$$\delta = \frac{N}{k_b}$$  \(14\)
2.4.2 Clamped material

For the clamped material the diameter of the stress inflicted area $D_A$ according to figure (13) has to be determined. Generally and in this projects alignment, case c) is the most likely case. $k$ is calculated for each of the materials according to (10), where $A_{ers}$ is the area and is calculated by

$$A_{ers} = \pi \left( \frac{d_w^2}{4} - \frac{d_h^2}{4} \right) + \frac{\pi}{8} L_k d_w \left[ (x+1)^2 - 1 \right]$$

(15)

where $x = \sqrt[3]{\frac{L_k d_w}{(L_k + d_w)^2}}$

Calculating the constant for the individual materials is done in the same way as for the screw i.e according to (10) and summation according to (13) and lastly elongation according to (14).

![Figure 13: Cases for clamped material stiffness calculations](image)

The transverse Young’s modulus of composite material is lower than most metals. Miyagawa with others [13] investigated the transverse modulus of uni-directional carbon fiber epoxy prepegs of different thicknesses and the transverse modulus of carbon fiber by itself. DMA experimental testing of fibers and epoxy, produced a value of 7.02 GPa, fiber volume fraction was 0.5. Results for carbon fiber transverse modulus were between 5-14 GPa. The authors points out the significant difference compared the longitudinal stiffness of around 230 GPa. In Foundations of fiber composites [14] values for carbon fiber epoxy laminas are given as table values and are around 10 GPa. Resulting spring constant is gained by adding constants for both materials.

3 Other design interests

In the following section deals with other design aspects of mechanical fastening: COF, CTE, Washer size and galvanic corrosion.

3.1 Coefficient of friction

The coefficient of friction, cof, is the single most important parameter for these types of joints. There’s friction between the clamped materials, which determines the amount of clamped force needed. Then
there's friction in the treads and friction between rotating parts of the joint (rotating during tightening) which inflicts the amount of clamped force received when applying a certain torque. In this study the cof of interest is of course mainly the one between the carbon fiber and the steel. No coating had been selected up to the point of this project, resulting in an uncertainty of what the actual friction constant will be. However, the friction constant between not coated carbon fiber vinylester and steel was investigated. The purpose is to compare it to the cof between steel and steel and thus determine whether there's a large enough different to have an significant effect on the joint. Main focus are on the effects of material hardness, fiber direction in the outer layers of the composite and temperature.

Joakim Schön [15] tested a carbon fiber epoxy composite in contact with aluminium and determined an initial cof of approximately 0.23. He also determined that the coefficient, rather predictable, was independent of the normal load applied. Heerington and Sabbaghian [16] tested different washer materials against a graphite fiber-reinforced epoxy and found an initial coefficient with a stainless steel washer of about 0.10. They also conclude that the angle of the outer layer in general had a small effect on the resulting cof. It seemed to be governed more by characteristics of the resin (roughness and hardness) as well as how well the washers had been degreased. Eliezer, Schulz and Barlow [17] investigated epoxy resin sliding against a steel disc with a hardness of 388 [HB] (intermediate hardness) and came to the conclusion that the steady state cof was independent of the temperature up to the glass transition temperature but it was considerably larger than the previous ones at a value of 0.8.

Sung and Suh [18] studied the friction and wear behavior of graphite/epoxy and Kevlar/epoxy against steel. Their values pointed towards an initial cof of about 0.3 for the graphite and 0.4 for the Kevlar.

It seems fair to conclude that the cof mainly depends on the roughness and hardness of the clamped materials. The large variety of the results points towards the possibility of tailoring the cof if a certain value is desired and that grease and other unwanted surface treatments may change the results dramatically. The lowest value reported, 0.1 is the same value as for steel-steel [19]. In short it’s unlikely to go lower, i.e worse than that.

3.2 Coefficient of thermal expansion

Thermal expansion has been proven to potentially cause sever problems for these types of joints. Figure (14) shows results form a test at VW in Kassel, where a M8 steel screw in a transmission made of magnesium where subjected to 120°C for 800 hours, causing it to loose 96% of its initial clamped force [19].
The larger cte of the magnesium caused an increase of the force acting on the screw, which in turn caused the magnesium to yield and succumb to stress relaxation. For composites the larger stresses will cause creep rate to increase and thus clamped force is lost more quickly. But the problem is even more complex for composites due to the anit isotropic material properties. Here there’s two ctes of interest, parallel to the fiber directions and transverse the fiber directions and the transverse cte is less investigated and is what governs the loss of clamped up force.

Lengthwise i.e in the fiber direction, aero space graded carbon fiber has a linear cte between $-0.1$ and $0.1 \times 10^{-6} [1/\degree C]$ , even standard carbon fiber has a cte of around $2 \times 10^{-6}[1/\degree C]$ compared with steel which has a linear cte, $\alpha_L$ of $\sim 11 - 13 \cdot 10^{-6} [1/\degree C]$. Composite laminas can, and often are though, tailored to have a very small cte. The linear elongation of a material $\delta_L$ at an given temperature change is given by

$$\delta_L = length \cdot \Delta\text{temp} \cdot CTE$$ (16)

The increased length of the screw and clamped materials in this study for a temperature increase from $23^\circ C$ to $80^\circ C$ is presented in table (1)

<table>
<thead>
<tr>
<th>Part</th>
<th>$\delta_L [\mu m]$</th>
<th>$\delta_{L,tot} [\mu m]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Screw</td>
<td>29.57</td>
<td>29.57</td>
</tr>
<tr>
<td>Steel plates</td>
<td>7.41</td>
<td></td>
</tr>
<tr>
<td>Spacer</td>
<td>18.45</td>
<td>25.86</td>
</tr>
<tr>
<td>CFRP</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

Table 1: Elongation due to temperature increase
Due to the relatively small thickness of the carbon fiber, vinylester the difference is accordingly less than 4 $\mu$m. Caccese with others [3] measured the “through thickness” cte of a E-glass vinylester lamina, fiber volume fraction was 0.51, using DMA (Dynamic mechanical analysis) and found it to be, on average $46 \times 10^{-6}$ [1/°C], significantly larger than steel. The linear cte for vinylester is $16 - 22 \times 10^{-6}$ [20] and for E-glass its $54 \times 10^{-6}$ [21]. Caccese and co. makes a comment on limited experimental data and that further experimental studies are recommended. Sideridis [22] tested unidirection glass-fiber epoxy and results varied from $30 \times 10^{-6}$ to $22 \times 10^{-6}$ for fiber volume fraction 0.6 to 2. There were only small differences comparing E-glass to carbon fiber. The “thin” thickness makes the elongation quite small anyway, around 6 $\mu$m. It seems fair to conclude that the fiber volume fraction significantly affects the cte and that it decreases due to its increase. The figures points towards a cte value of around the same magnitude as alloys like aluminium. Noteworthy is that the constant area expansion $\alpha_A$ is twice as large as $\alpha_L$ and the volumetric expansion coefficient $\alpha_V$ is 3 times $\alpha_L$.

### 3.3 Washer size

The main purpose of the washer is to distribute load over a larger area, thus decreasing the applied pressure and lower the risk that the material underneath the bolt head starts to yield. The fairly simple equation (17) describes the relations between clamped force, washer diameters and the amount of pressure achieved.

$$ P = \frac{4 \cdot F_v}{\pi (D_0^2 - D_h^2)} $$

where $D_0$ is the outer diameter of the pressurised area e.g outer washer diameter and $D_h$ is the inner diameter of the pressurised area, preferably the hole diameter. Maximum pressure values for different materials are found through experimental testing and for most metals their well known. Composite materials faces different challenges, though they don’t yield the pressure creates the risk of delamination. The maximum pressure is therefore a lot less than that for metal, exact figures regarding this comes sparse but unofficial data suggest values around 15-20 N/mm². For the case in this study, with $F_v = 18$ kN and $P \leq 20$ N/mm², the need of a washer is quite clear.

The size of the washer, i.e the amount of pressure, will influence the strength of the joint. Khashaba [24] with others investigated what effect the washer size had on performance of bolted joints in composites. Using constant pressure and varying the outer diameter of the washer, they found there to be an “optimal” outer diameter which gave the joint the largest bearing strength. An increasing washer size decreased the stiffness of the joint as well as the failure load. However the bearing strength reached its maximum at an intermediate washer size. Khashaba and co. concluded that, increasing the pressurised area decreased the strength, but increasing the area also made the lateral constrained area larger, which in turn increased the strength. With these two phenomenons acting against each other there bound to have a meeting point where the load caring ability is at its largest.

Yan and others [25] preformed an experimental study of how clamp-up effected the net-tension failure of composite plates. They found that an outer washer diameter/ hole diameter ratio equal to 2 or less, resulted in a 20% reduction of tensile strength. It seemed though that a ratio larger than 3 did not increase the strength of the joint considerably.
3.4 Galvanic corrosion

Galvanic corrosion occur when two or more materials have electrical contact with each other through an electrolyte. The electrolyte provide means for ions to move from the anode (the material with lower electrode potential) to the cathode (higher potential). The anodic material therefore corrodes more rapidly then it otherwise would have. A difference of 50 mV in electrode potential is needed for the reaction to start [26]. Usually the reaction is speed up when the anodic surface is small and the cathodic is large. The risk of galvanic corrosion has led to significant reduction of usable metallic material for the joint design. The most commonly used are alloys of titanium or corrosion resistant steels. Aluminium is basically out of the question. Tavakkolizadeh and Saadatmanesh [27] investigated the galvanic corrosion of 38 carbon fiber epoxy specimens against steel. They tested two different aggressive environments, and several different levels of epoxy coating, including none at all. Lastly the effects of sizing agents was also investigated. Results showed of galvanic corrosion and a increase of corrosion rate by a factor of 24 and 57 for carbon fiber in direct contact with steel, in deicing salt solution and seawater respectively. The rate of corrosion is also governed by the thickness of the epoxy coating. A 0.1 mm layer decreased corrosion rate by 5 and 7 times, for deicing salt and seawater and a 0.3 mm by 21 and 23 times. The deicing salt solution, on average had a 15 % higher rate of corrosion. Sizing agents also decreased corrosion.

Torres-Acosta with others [28] performed similar experiments, testing CFRP and steel in chloride contaminated concrete, water-to-cement ratio of 0.41, in different humidity. They too confirmed existence of galvanic corrosion with the steel being the anodic. Galvanic current proved to be greater at intermediate humidity levels.

Part II

Relaxation and creep testing

The purpose is to determine how parameters clamped force and high temperature affects the relaxation of a CFRP-to-metal joint. Loss of clamped force, loss in ability to transfer load and failure modes is investigated.

4 Before testing

This section covers pre-testing aspects and considerations.

4.1 Boundary conditions

According to the bachelor thesis the CFRP-beam should be able to resist temperatures in between -40 to 100 °C without lowering any of its structural properties. From this the temperatures of 23° C and 80° C were chosen. Negative temperatures is considered not to increase the relaxation and were therefore abolished.

The load has been formulated with respect to the minimum clamped force calculated, $F_{req}$, with a value of 18 kN. Friction coefficients are chosen conservatively, 0.16 for the threads and 0.1 for the CFRP-Steel. In assembly the applied torque may change up ±50 %, creating the risk that a less than
50% difference in torque, would still result in the same clamped force. This is however not an issue for testing since the joints are being tightened using much more accurate tools.

4.2 Material

Test specimens are made of Toray T700, non-crimp carbon fiber weaves, [±45°] and [0°/90°] directions. Surface weights for both are 450 [g/m²]. The matrix is Reichholds DION 9102 and has among other things a very good high-temperature stability, good resistance to corrosion and is also suitable for use in vacuum infusion [29]. Additional information regarding the fiber and matrix is available in the bachelor thesis as well as the Reichhard bullet-in.

4.3 Specimens

The specimens are ≈ 184 × 45 × 5 mm. Stacking sequence of the specimens are [±45° 0°/90° ±45° 0°/90° ±45°]. The Young’s modulus for the lamina in tension was determined to 32.7 [GPa] [30]. Thickness at the holes where measured using a digital caliper and two measurements where taken for each hole and was then averaged. Thicknesses had a spread from 4.51 to 4.74 mm averaging at 4.64 mm. Hole diameter is, quite accurately, 10.0 mm and centred 33 mm from the edge. For the specimens with two holes, the second one is located 35 mm from the centre of the first hole. Average weight was 52.7 grams for specimens with one hole and 52.2 grams for specimens with two. In total, 23 specimens, 6 with 2 holes and 17 with 1 hole. The specimens were manufactured using vacuum infusion and the fiber volume fraction was determined with the burn-off method to an average of 0.58 [30]. The holes were drilled using orbital drilling.

4.4 Test rig

The test rig consists of the specimen being clamped in between two steel plates of the same size and thickness as the specimen itself, as seen in figure (15). The steel plates are made of S235, common construction steel. The spacer is 24.9 mm long with an outer radius of 22 mm and inner of 12.3 mm and with a hardness of 230 HV, strength classification 8, Scania article number 135718430. Flange nut is an M10, strength classification 10, provided by Nedshroef. The screw is an M10, ultrasonic flange screw, strength class. 10 with zink flake coating giving it a cof of about 0.15 against steel. The screw were pick to as well as possible correspond to the M10 hexagon head screw with flange screw art number 812537, Scania standard STD3353. Dimensions for the test fixtures is described in appendix B.
For the shear test the fixture was set in an Instro, structural testing device (100 kN) and pulled at a rate of 2mm/min. An extensors was used to measure the displacement between steel plates and carbon fiber specimen. Load vs displacement was plotted. Test rig is show in figure (16).

5 Method

This section describes the method used for the experimental part of the study.

5.1 Joint assembly

The joints was tightened at Atlas Copco, Stockholm using the software Powermac’s Gauging®. An ultrasonic measuring equipment and software was used to measure the actual applied torque (even though the equipment is very accurate the exact torque is rarely achieved), clamped force and stretch of the screw. The equipment recorded the echo from the signal and calculated wanted data. The echo for each individual screw was then saved. Noteworthy is that only about 1/3 of the stretch of the screw is actual physical elongation, the rest is an “error” due to a temperature increase, thus a
material resistance change, while the screw is being tightened. Fixture were hold still in an vice and the rotation of the bolt was hindered by a wrench.

5.2 Testing

Testing was divided into two parts. Loss of clamped force and stretch were measured at Atlas Copco using the ultrasound measuring equipment, accuracy of 1µm. The equipment consisted of one probe measuring the echo from the ultrasonic signal and one measuring the temperature of the screw in order to compensate for differences at the time of the initial tightening. Current echos were then compared with initial echos giving the current loss of clamped force. Measurements were done twice a week for 28 days. Specimens were kept in a climate chamber at KTH.

Load transfer i.e. shear testing was tested twice, one as reference and one after 28 days, at KTH. Before each measuring, fixtures were cooled down to room temperature to prevent results from being effected by the thermal expansion. Because of this and the fact that tests were not carried out at the same place as specimens were stored, also introduced a cyclic temperature exposure. Specimens were transported in an isolated suite case with a average temperature of 30° C on the way to measuring and 22° C away from measuring for an average of 35 minutes for each leg. In total the specimens spent about 17 h out of the oven. Average time for measuring was 35 min and average temperature was 24° C.

All specimens were then subjected to shear testing to determine change in load carrying abilities and/or failure modes. The joints were double lap joints, as shown in figure15 to eliminate secondary bending due to non transverse forces acting on the joint.

The complete test matrix is presented in table 2.

<table>
<thead>
<tr>
<th>Test series #</th>
<th>Temp [°C]</th>
<th>Time [days]</th>
<th>Moment [Nm]</th>
<th>#Test</th>
<th>#Bolt</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 (Ref)</td>
<td>23</td>
<td>-</td>
<td>37</td>
<td>3</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>23</td>
<td>-</td>
<td>28</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>23</td>
<td>-</td>
<td>46</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>23</td>
<td>28</td>
<td>37</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>80</td>
<td>28</td>
<td>37</td>
<td>3</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>23</td>
<td>28</td>
<td>28</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>23</td>
<td>28</td>
<td>46</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>23</td>
<td>-</td>
<td>37</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>9</td>
<td>23</td>
<td>28</td>
<td>37</td>
<td>3</td>
<td>2</td>
</tr>
</tbody>
</table>

Table 2: Test matrix

6 Results

6.1 Predicted initial results

The required clamped force, required torque, spring constant for screw and clamped material considering the load case described in section 2.1 were all predetermined using the method described by equations in section 2. Calculations were done using Matlab. Results are presented in table (3).
6.2 Torque and initial clamped force

Results for the average applied torque and resulting initial clamped force for the test series is presented in table (4). Temperatures at 24 °C.

<table>
<thead>
<tr>
<th>Series num.</th>
<th>Torque [Nm]</th>
<th>Aimed</th>
<th>Clamped</th>
<th>Aimed clamped</th>
<th>Stretch, δ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>[Nm]</td>
<td>torque</td>
<td>force [kN]</td>
<td>force [kN]</td>
<td>Screw</td>
</tr>
<tr>
<td>1</td>
<td>36.2</td>
<td>37</td>
<td>16.7</td>
<td>18</td>
<td>0.068</td>
</tr>
<tr>
<td>2</td>
<td>27.3</td>
<td>28</td>
<td>12.5</td>
<td>13</td>
<td>0.052</td>
</tr>
<tr>
<td>3</td>
<td>44.9</td>
<td>46</td>
<td>22.1</td>
<td>22</td>
<td>0.091</td>
</tr>
<tr>
<td>4</td>
<td>36.2</td>
<td>37</td>
<td>16.8</td>
<td>18</td>
<td>0.071</td>
</tr>
<tr>
<td>5</td>
<td>36.1</td>
<td></td>
<td>18.0</td>
<td></td>
<td>0.075</td>
</tr>
<tr>
<td>6</td>
<td>27.4</td>
<td>28</td>
<td>12.2</td>
<td>13</td>
<td>0.051</td>
</tr>
<tr>
<td>7</td>
<td>44.8</td>
<td>46</td>
<td>20.4</td>
<td>22</td>
<td>0.084</td>
</tr>
<tr>
<td>8</td>
<td>36.2</td>
<td>37</td>
<td>16.7</td>
<td>18</td>
<td>0.069</td>
</tr>
<tr>
<td>9</td>
<td>36.2</td>
<td></td>
<td>17.1</td>
<td></td>
<td>0.071</td>
</tr>
</tbody>
</table>

Table 4: Initial data

6.3 Results loss of clamped force

Figure (17) and (18) shows loss of clamped force for the test series using a single screw.
Figure 17: Loss of clamped force, single bolt at room temperature and 80° C

Average total loss for 37 Nm, room temperature, 28 days (series 4) is 8.3%. Average total loss for 37 Nm, 80° C, 28 days (series 5) is 35.5%.
Figure 18: Loss of clamped force, single bolt, 28 and 46 Nm

Average total loss for 28 Nm, room temperature, 28 days (series 6) is 8.6%. Average total loss for 46 Nm, room temperature, 28 days (series 7) is 8.8%.

Figure (19) shows loss of clamped force for double jointed test series. Observe that each graph represents one fixture.
6.4 Results load bearing strength

Figures (20) to (23) shows force-deformation curves with ultimate strength, accompanied with graphs more accurately showing the slip load.
Figure 20: Force-deformation graph with test series 1, 2 and 3
Figure 21: Force-deformation graph with test series 4 and 5

Figure 22: Force-deformation graph with test series 6 and 7
Figure 23: Force-deformation graph with test series 8 and 9

Average maximum bearing strength and slip load for reference series 1-3 and 8 is:

- Series 1: Max: 45.4 [kN], Slip: 9.7 [kN]
- Series 2: Max: 44.5 [kN], Slip: 7.3 [kN]
- Series 3: Max: 49.5 [kN], Slip: 12.2 [kN]
- Series 8: Max: 72.3 [kN], Slip: 18.4 [kN]\(^2\)

Average maximum bearing strength and slip load for series 4-7 and 9 is:

- Series 4: Max: 48.8 [kN], Slip: 10.0 [kN]
- Series 5: Max: 49.3 [kN], Slip: 13.1 [kN]
- Series 6: Max: 45.1 [kN], Slip: 7.2 [kN]
- Series 7: Max: 49.1 [kN], Slip: 12.3 [kN]
- Series 9: Max: 75.4 [kN], Slip: 20.0 [kN]

Slip load on average increased 59% due to an increased clamped force of 60%. Maximum bearing load deviated by 4.8 kN for single bolted joints and by 3.1 kN for double joints. Difference between single and double joints on average 26.5 kN.

\(^2\)Specimen 8.3 was abolished due to faults occurring while testing
6.5 Long term evaluation

Data for the loss of clamped force graphs was curve fitted using Matlab to evaluate the losses in long term.

- Specimens with a loss of between 8 and 9% after 28 days had lost all clamped force after approximately 5 years.
- Specimens with a loss of between 36% after 28 days had lost all clamped force after approximately 3.7 years.

7 Conclusion and observations

- The applied torque nearly reached the targeted values and the resulting initial clamped up force showed good accuracy to the expected.
- Slip load increased with increased clamped force, which was expected.
- Slip load was larger than was expected, likely reason is uncertainties with cof, calculations were done conservatively. No real data on cof between these specific materials, also the cleanliness of the specimens can significantly effect the resulting cof.
- Ultimate strength did change for specimens tested after 28 days.
- All specimens fail in bearing.
- For single bolted specimens failure was far more progressive than for double joint, which failed catastrophically.
- Slip load for specimens subjected to 80 °C for 28 days, did not decrease (in fact a small increase was observed) even though clamped force had been reduced.

The progressive failure of the single joint can be seen in figure (24). The screw has nearly moved a distance of it’s own diameter into the carbon fiber/ vinylester and still catastrophic failure has not occur.

![Image](image_url)

Figure 24: Progressive bearing failure of the single bolted joint
7.1 Regarding loss of clamped force and shear strength

Perhaps the most interesting results is that the slip load for series 5 (80°C) did not decrease even though the loss of clamped force was about 36%. In comparison the around 60% decrease in clamped force from 46 Nm to 28 Nm initial torque, resulted in a slip load averaging at 12.3 kN and 7.3 kN, also about a 60% decrease. It is however in some cases hard to determine the correct slip load. The amount of tested specimens are to few to make any considerable assumptions but the results are still quite interesting. At least for these specimens it seems that the cof must have increased thus compensating for the loss of clamped force. Contradicting this, as described in section 3.1 Elizer, Schultz and Barlow came to the conclusion that the cof should not be affected by temperature, but at the same time their cof was significantly larger than what other experiments had shown.

Clamped force was measured after 28 days for the three of the test specimens (#1.3, 2.2 and 3.2) which had undergone shear strength testing the first thing. Testing was halted just before the specimens had failed and interestingly enough the clamped force was actually about equal to or even larger than the initial clamped force and of course substantially larger than for the corresponding specimens which had not undergone tensile testing. Looking at one the fixtures, figure (25), may give some clues to why this is.

The steel plates have clearly been bended outwards and the screw has slightly bended as well (hard to see in figure), possibly causing the bending of the plates. Either the plates or the screw itself or a combination of these factors must have contributed to a further elongation of the screw, thus creating a larger clamped force. However, it’s of course wrong to believe that the strength of the joint would actually have increased due to this. As earlier established the increase in clamped force doesn’t increase the bearing load much and failure is just around the corner for these specimens.

Test results show that an increase of about 60% in clamped force (28 to 46 Nm in initially torque) or exposure to elevated temperatures did not result in an effected bearing strength. Similar results was also produced by Park [4], see section 1.2.2. This seems fair enough since after the materials has slipped between one another and it’s only the carbon fiber/ vinyl ester itself which carries the load. Potentially a large clamped force could have weakend the material and thus triggered and failure at an earlier stage, but since these specimens were clamped between “thick” steel plates, no such risk occurred. As for the higher temperature, it also made no difference, which could be expected since

Figure 25: Fixtures after shear strength testing
the temperature was under glass transition temperature and therefore no structural changes in the material will occur. Increasing the clamped force will increase the initial slip load somewhat, but the ultimate failure load is about 5 times that load, leaving one questioning why one would design with a large clamped force, thus risking applying to much pressure and increase settling and chances of creep, at all. But, a low clamped force also equals a short elongation of the screw or bolt and short elongation means low safety insurance against dynamic loading.

The lowered slip load seems to be of the same magnitude as the loss of clamped force, i.e a decrease of clamped force of around 8% did result in a lowered slip load by about 8%, which was expected. Again data is though not enough to make definitive conclusion regarding this.

It’s not likely that viscoelasticity single handed caused the increased loss of clamped force for the specimens subjected to elevated temperatures. As show by the test performed at VW [19] joints which “only” has different materials will also have a significant loss due to the differences in CTE. It seems reasonable to conclude that if the total CTE of the clamped material is larger than the screw, the pressure applied to the clamped material will of course increase and that will increase rate of creep and relaxation. But as said before, for the experimental test in this study, it’s hard to determine exactly how much of the losses is due to creep and how much is settling in other parts of the joint. One can only determine the total losses and conclude that it’s likely that settling in other parts of the joint is relatively small compared to the creep/relaxation in the carbon fiber/vinyl ester.

7.2 Sources of error

Looking at figures (17) to (19) one can see that the values for the clamped force in some cases momentarily increases. This is of course nothing that actually happens and can simply be “blamed” on somewhat dodgy and inconsistent measuring equipment. The fact is that a constant reference values had to be taken in order to compensate for the errors and average them out as well as possible. Two loose screws were used as reference as their stretch and clamped load obviously should be zero. Their echoes were recorded initially and the difference they had for each time of measuring was averaged to a correction factor, which in turn was either added on taken of the results shown for the each of the joints. These reference values were frequently taken, up to 7 times for each time of measurement, as they could even change over the time of the measuring. A lot of effort was put into trying to minimize these “built in” errors and the overall results should not have become compromised to much by this. On a second note, the stretches measured, which in turn determines the clamped force, are considerably small. A difference of only 10µm could change the indicated clamped force by a 100 N, meaning that it’s obviously very easy for it to make that slip.

8 Future work

Concerning an engine mount in a bus, it should be interesting to consider other possible load cases like bending and fatigue. Also the effect of moisture on the loss of clamped and load transferring abilities. The experimental part of this study showed that the specimens subjected to increased temperatures did not suffer from a lowered slip load, potentially meaning that the cof has increased. Data from this testing was to scarce, thus it would be interesting to investigate reasons why this might have happened and see if results can be replicated. Bearing strength was not effected by the clamped load in these experiments, it could be interesting though to know at which pressure it starts to effect the bearing strength of the composite. Otherwise other environmental testing such as corrosion rate are always of interest. Regarding the mathematical approach to determine the long term loss of clamped force. In
this study “only” a rather simple curve fit was used and it would be interesting to use the model which was described in this study, however these model requires some specific material parameters and the authors don’t really give out what they are. There is more model similar to that one but they all lack of important information regarding these material specific parameters. Further inquiry is needed.

Acknowledgement

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Appendix A

Definitions:

Figure 26: Definitions
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal diameter of the thread (out-going)</td>
<td>d</td>
</tr>
<tr>
<td>Nominal diameter of thread (in-going)</td>
<td>D</td>
</tr>
<tr>
<td>Out-going thread’s inner diameter</td>
<td>d₁</td>
</tr>
<tr>
<td>In-going thread’s inner diameter</td>
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</tr>
<tr>
<td>Out-going thread’s average diameter</td>
<td>d₂</td>
</tr>
<tr>
<td>In-going thread’s average diameter</td>
<td>D₂</td>
</tr>
<tr>
<td>Diameter of the screw stem</td>
<td>ds</td>
</tr>
<tr>
<td>Friction diameter</td>
<td>Db</td>
</tr>
<tr>
<td>Pitch</td>
<td>P</td>
</tr>
<tr>
<td>Tolerance level: Defined as ( \frac{\sqrt{3}}{4} )</td>
<td>H</td>
</tr>
<tr>
<td>Not in figure: Defined as ( d₁ - \frac{H}{4} )</td>
<td>d₃</td>
</tr>
</tbody>
</table>