1. Project objectives and goals

Bolted joints represent one of the most commonly used forms of connections in modern day steel constructions. They are an invaluable asset to both civil engineering as well as mechanical engineering, allowing the fastening of seemingly arbitrary cross sections. Possessing a relatively small cross-section however, they are often a critical element of any structure they are used in. Especially the mated threads are subjected to high concentrations of stresses. Under cyclic loading, this can lead to a premature failure of the connection with catastrophic consequences.

To counteract failure under cyclic loading conditions, bolted joints are preloaded in most cases. By preloading the connection, the resulting stress amplitude under service loads is decreased leading to a significant increase in durability as well as reduced deformations under load. In most cases, the preload is introduced into the connection by over-tightening of the nut. Different preloading procedures are given in EN 1090-2 [1] and the National Annex of EN 1993-1-8 [2]. These preloading procedures differ in the level of preload to be achieved as well as the tightening conditions. The rotation of the nut can either be applied by a torque wrench (torque-controlled) or by observing the rotation angle (angle-controlled).

During these preloading operations, the connection is already subjected to high loads. The resulting thread stresses are near impossible to determine experimentally. The project goal of the present thesis is therefore to perform numerical analyses on single-bolt bolted connections subjected to different preloading procedures with varying tightening conditions. The numerical analyses allow a focused evaluation of thread stresses generated during tightening of the connection. Different procedures are compared and evaluated with respect to preloading levels achieved and boundary conditions during tightening. Lastly, recommendations for practical application are derived based on the analyses results.

2. Description of method and results

The numerical analysis of a chosen connection is performed in the commercial Finite-Element-Software ABAQUS/CAE. VDI 2230-2 [3] covers the systematic calculation of highly stressed bolted joints. Given in VDI 2230 are so-called modelling classes for Finite-Element applications, pertaining to different levels of detail in modelling the bolted connections. The complexity of modelled joint increases from class 1 to 4. For the given goal, only modelling class 4 with the highest modelling complexity is viable.

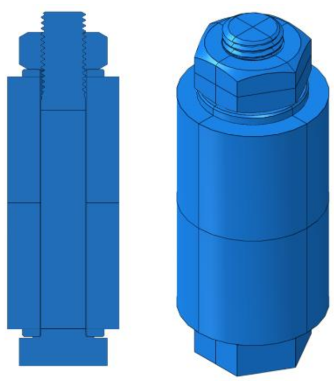

Figure 1: Modelled single-bolt connection

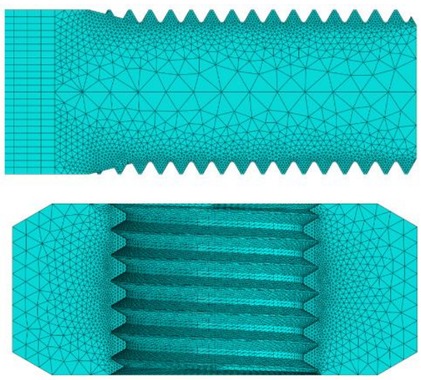

Figure 2: FE-Mesh of the threaded components

In a first step, a chosen connection of the dimension M24x135 HV according to EN 14399-4 [4] is modelled with all geometric details. The connection modelled is shown in Figure 1. Due to the complexity of the geometry of the model, a regular hexahedral (C3D8R) mesh is not possible for most of the threaded parts. For meshing the threads, C3D10M elements with the ABAQUS function of local seeding are used to refine the mesh in regions of expected stress concentrations. These are in particular the threads (flanks and roots) and the bearing surface of the bolt head.

A global mesh size of 4 mm is used with seeding refinements reaching as low as 0.4 mm in the thread roots. All non-threaded components are meshed with the more regular C3D8R elements and component partitions are used to also allow the meshing of the bolt shaft with these more efficient elements as well. The mesh created in the mated threads is shown in Figure 2.

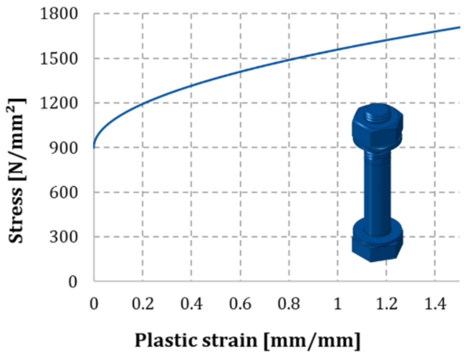

Figure 3: Plastic material derived from nominal material parameters

The bolt material grade is chosen as HV 10.9. Material parameters are taken from EN ISO 898-1 [5] and the created material is applied to the threaded components as well as the washers. The derived material curve is shown in Figure 3. As the plates to be connected are not a relevant part of the analysis, a simple linear elastic material model with an Elasticity modulus of $E = 210\ GPa$ and a Poisson's ratio of $\nu = 0.3$ is sufficient.

The contact modelling of the connection is based on ABAQUS' "general contact" formulation. Based on a defined clearance limit value between surfaces, contact is established automatically. In this kind of contact definition, only a single interaction property can be used. This means, that the same coefficient of friction has to be used for contact modelling in the threaded parts as well as the bearing regions in the model. As most of the components are constrained in the rotational degree of freedom around the centre axis, friction only affects the moving parts. This is highlighted in Figure 4. Additionally, the application of a uniform coefficient of friction for thread- and bearing contacts complies with current state-of-the-art recommendations [3]. In VDI 2230-1 different friction coefficient classes are given for the friction parameters of threads. These classes range between friction coefficients of $0.04 \leq \mu \leq 0.10$ for class A to $\mu \geq 0.30$ for class E. According to VDI 2230-1 a friction coefficient of class B is to be achieved with parameters of $0.08 \leq \mu \leq 0.16$ [6]. In the present model, the initial friction coefficient is chosen based on this recommendation to $\mu = 0.13$.

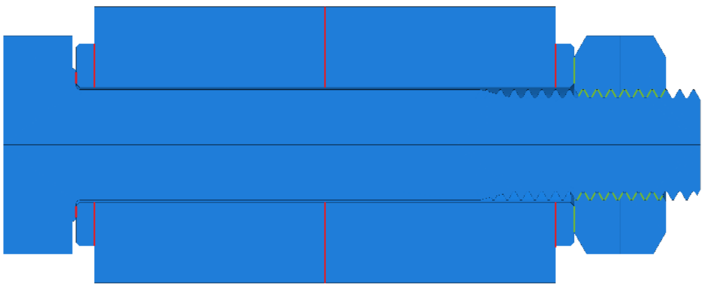

Figure 4: Contact regions in the connection. Regions affected by friction are highlighted in green

In a first simulation the "modified torque method" according to German National Annex DIN EN 1993-1-8/NA is applied on the connection. For the chosen connection of a M24 HV10.9 bolt a tightening torque of $M_t = 800\ Nm$ is required [2]. This torque is directly applied on the model through the use of a reference point coupling the wrench flats. The clamping force is evaluated by integrating the stresses in the cross section in the unthreaded part of the shaft. The resulting clamping force doesn't achieve the required level of preload of $F_{p,C}{}^* = 220\ kN$ [2]. Instead a clamping force of $F_V = 188\ kN$ is observed, falling short by 15%. In subsequent analyses two different possibilities to increase the clamping force are evaluated.

First, the friction coefficient in the model is reduced. A friction coefficient of $\mu = 0.10$ is used in a second analysis. All other system parameters remain unchanged. In this model, the evaluated clamping force reaches $F_V = 229\ kN$ exceeding the required level by roughly 4%. To investigate whether a linear relationship between friction coefficients and achieved clamping force can be observed, further

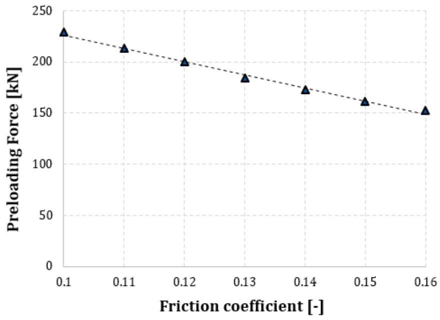

Figure 5: Relationship between friction coefficients and preloading force

analyses are conducted with parameters ranging between $0.10 \leq \mu \leq 0.16$. Figure 5 shows the resulting relationship.

Furthermore, the initial analysis is rerun with an increased tightening torque. DIN EN 1993-1-8/NA indicates a linear relationship between tightening torque and achieved preloading. This relationship is investigated taking into account the observations from the previous parametric study. Since VDI 2230-1 recommends friction coefficients of up to $\mu = 0.16$, it is calculated which tightening torque is required to achieve the general level of preload ${F_{p,C}}^* = 220\,kN$. This is done based on an assumed proportional relationship between both tightening quantities. The required torque is therefore $M_t = \frac{{F_{p,C}}^*}{F_V(\mu=0.16)} \cdot M_{t,code} = \frac{220}{153} \cdot 800 = 1150\,Nm$. Applying this torque moment on the connection with $\mu = 0.16$ leads to a clamping force of $F_V = 222{,}9\,kN$, just slightly above the required level. A linear relationship can be confirmed once again.

A common problem in everyday application of bolted connections is the accessibility of the nut. According to code, the torque has to be applied on the nut only while the bolt head is restrained against rotation. For the case of bolt-sided preloading, a process test is required.

In the next step, the preload is therefore applied by rotating the bolt head and constraining the rotation of the nut. Equal friction coefficients and tightening torques are used to guarantee comparability of the obtained results. The evaluation of thread stresses in the nut thread show that a similar stress level is obtained in both cases. The maximum stresses for bolt-sided preload exceed the standard case by 1.6%. It can also be seen, that the stress in the threads decreases along the thread length, with the run-out threads being nearly unloaded.

The same calculation is rerun with modified boundary conditions. This time, the wrench flats of the nut are only constrained in their outer half. This represents the case of a partly sunken connection in which the nut can't be fully constrained. The thread stresses are now significantly higher than in both previous models, exceeding the maximum stresses by 9% and 11% respectively. In all three modelled cases, the clamping force didn't show any deviations.

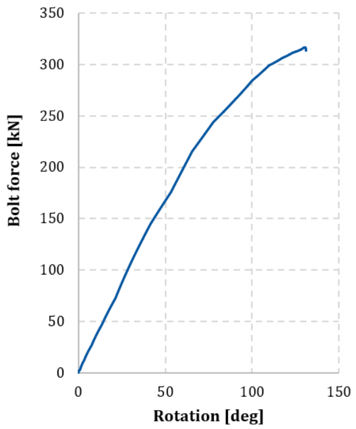

Figure 6: Bolt force rotation curve of M24x135 HV

Next up, a new analysis is run based on the "combined method" of EN 1090-2. The first step of this method is again torque controlled. To simplify the procedure in the numerical model, this first step is simulated in a preliminary analysis and the obtained angle of rotation is measured. The actual tightening is then applied by adding up the measured angle from the first step and the required additional angle from the second step according to EN 1090-2. As the rotation angle is independent of the friction coefficient, it is of reduced significance in this analysis. The evaluation of the clamping force shows a preload exceeding the required level by 27.5%. This is due to the geometry of the model, as the first tightening step already leads to a preload of $135\,kN$. In a real model with all of its imperfections, the first tightening step is required to guarantee a snug fit of the nut to the plates. Figure 6 shows, that the modelled connection adopts a non-linear behaviour after exceeding a preloading level of ca. $F_V = 220\,kN$. This

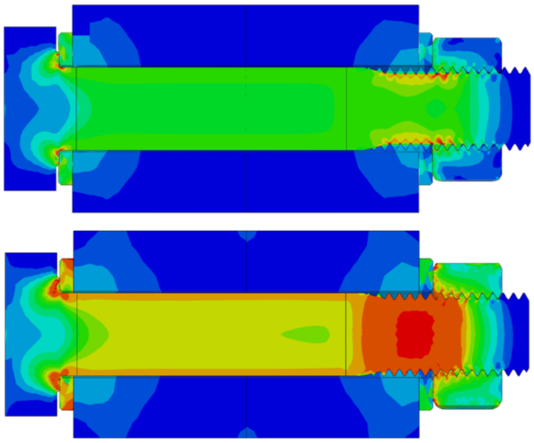

Figure 7: Stress plots of the modified torque method (top) and combined method (bottom)

is due to increasing plastic deformations of the most excessively loaded threads. This characteristic is typical for connections preloaded according to the used procedure. Figure 7 shows a comparison of the stress plots generated by both preloading operations considered in this thesis. As the target level of the combined method is ca. 14% higher than that of the modified torque method, higher stresses are to be expected [1].

3. Potential for application of results

In the analysis it was shown, that a linear relationship between friction coefficients and preloading force exists. Currently, one of the leading causes of premature fatigue failure of bolted connections is insufficient preloading of a connection. Since the experimental determination of clamping forces present in a connection is often omitted due to economic reasons, the preloading of a connection often leads to a "black-box" type scenario with unknown connection parameters. Based on the performed research, the preloading of a connection can be determined numerically without the additional need of experimental tests. This can lead to a significantly increased service life of preloaded connections while subsequently contributing greatly to operational safety of these joints. The "combined method" is shown to be less sensitive to changes of system parameters and more robust in achieving the required levels of clamping forces across different cases investigated. It is therefore recommended for future use in all bolted connections with structurally relevant preload. The "modified torque method" is only recommended for cases in which the preload is only intended to increase the serviceability performance of a structure, as it generally is more sensitive to changing parameters. The application of "off-the-code" tightening conditions is shown to have a significant effect on thread stresses. Special consideration is therefore needed in those cases.

4. References

[1] EN 1090-2, Execution of steel structures and aluminium structures – Part 2: Technical requirements for steel structures, 2018.

[2] EN 1993-1-8/NA, National Annex – Nationally determined parameters – Eurocode 3: Design of steel structures – Part 1-8: Design of joints, 2010.

[3] VDI 2230 Part 2, Systematic calculation of highly stressed bolted joints, 2014.

[4] EN 14399-4, High-strength structural bolting assemblies for preloading - Part 4: System HV - Hexagon bolt and nut assemblies, 2015.

[5] EN ISO 898-1, Mechanical properties of fasteners made of carbon steel and alloy steel – Part 1: Bolts, screws and studs with specified property classes – Coarse thread and fine pitch thread, 2013.

[6] VDI 2230 Part 1, Systematic calculation of highly stressed bolted joints, 2015.