

Finite-Element-Based Methods for the Fatigue Design of Bolts and Bolted Joints

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ABSTRACT

Due to the increase of computer performance, modern screw design is more and more based on finite element analyses. Depending on the problem, different possibilities of modelling bolted joints are reasonable. Traditionally, a nominal stress approach is used for the fatigue design of bolted joints. But, in modern lightweight constructions it is more and more common to connect threaded components directly without using a classic bolted connection. In this case a local approach has to be used. In several investigations at the Fachgebiet und Institut für Werkstoffkunde (Chair and Institute for Materials Technology) in Darmstadt and the Robert Bosch GmbH in Schwieberdingen different concepts for a local approach have been tested for the fatigue design of threaded connections. As a result, an overview can be given, which design concept and which model can be properly used for calculations.

INTRODUCTION

Bolted joints have always been one of the most important and challenging fastening elements in the design and construction of components. Especially the fatigue assessment is of vital importance, because notched parts are far more sensitive to cyclic stress than parts without notches. In order to use screws economically regardless of their cyclic sensitivity, an important goal of the design process is the minimization of cyclic stresses in the screw.

As shown in **Figure 1**, the load distribution depends on the joint partners' elastic behaviour and is described by

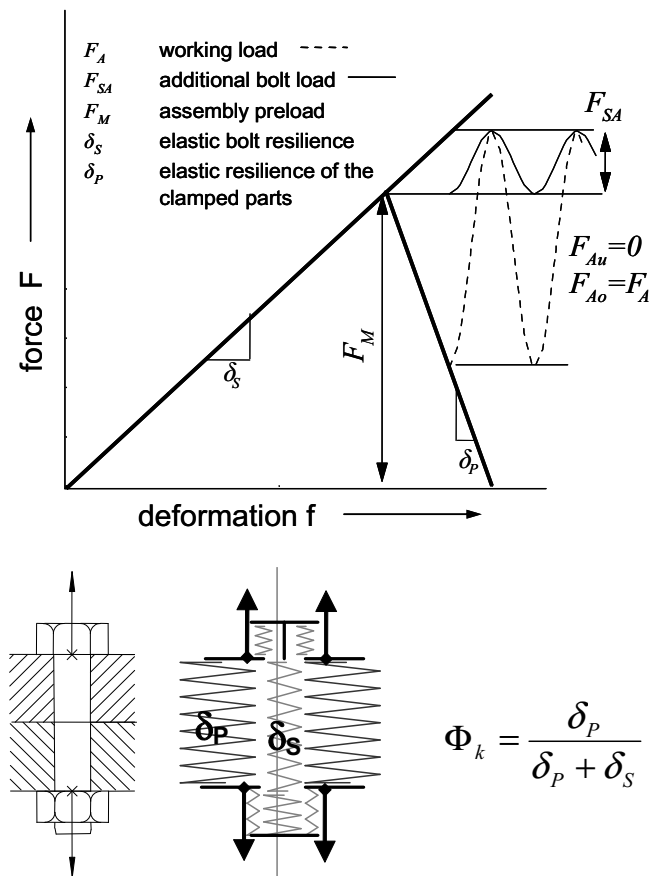


Figure 1: Graphic illustration of the mechanical load bearing behavior of an ordinary bolted joint described by means of a simplified spring model /1/

the resilience factor Φ_k . Until opening of the joint starts, the working load is divided in an unloading of the clamped parts and an additional load F_{SA} of the bolt. Because the bolt resilience is usually greater than the resilience of the clamped parts, the major proportion of the working load is carried by the clamped parts. After the opening of the joint the additional bolt load is increasing. Therefore bolted joints usually are highly pre-loaded during assembly.

The simple spring model of the VDI guideline 2230 /1/ may help to design bolted joints in many cases. Typically the nominal stress in the bolt is calculated to use the nominal stress approach as it is described in the guideline. Similar to every model its use is limited by the contained analytical and empirical approaches to the following:

- Valid for prismatic solids with limited lateral dimensions and the assumptions, that the cross sections remain flat under load and a linear stress distribution exists.
- The resilience factor Φ includes the elastic resilience δ_S of the screw and δ_P of the bolted parts. Both are calculated by the help of simplified approaches, which can be incorrect or inaccurate and seldom be used for real cross-sections.
- Predicting the fatigue limit of screws is very conservative since it is based on uncertain operands (for example special values in /1/ like the distance of the line of action of the axial load from the axis of the symmetrical deformation body a , the distance of the bolt axis from the axis of the symmetrical deformation body s_{sym} or the resilience factor Φ).
- Inhomogeneous stress distributions, which normally occur in components with bolted joints, can only be considered to a very limited extent.
- Especially the evaluation of multi-bolted joints contains so many assumptions that it inevitably leads to a conservative design.
- The nominal approach in the guideline implies that the fracture occurs in the first load-bearing thread turn of the bolt. Other notched regions or a variable load distribution along the bolts longitudinal axis can not be evaluated.

The limitation of the analytical calculation model for bolted joints in the guideline can partially be avoided by means of the finite element method, because stiffness and load transmission within complex shaped and loaded components can be calculated precisely. Therefore stresses within these components can be determined easily, if the corresponding FEM model is only detailed enough.

The result of a detailed FEM model, a model including the thread of the bolt and the contact in all load carrying interfaces, are local stresses and strains which have to

be evaluated. So, the limitations of the simplified analytical model of the guideline, especially referring to the nominal stress approach, can be avoided.

Since the grade of accuracy is strongly depending on the number, shape and local arrangement of the elements in the model, one can imagine that FEM models in general can become quickly non-applicable for daily use. Especially for “normal” bolted constructions the effort to model a bolted joint with the threads, the material behaviour and the frictional contacts in all interfaces is, regarding to the efficiency, unacceptably high (like in /2, 3, 4/ for example). Therefore two topics will be discussed within this article:

- The use of local concepts for the fatigue assessment of threaded connections in general and
- A simplified local approach to use the advantages of the local approach even for “normal” bolted joints.

LOCAL FATIGUE ASSESSMENT OF BOLTS AND BOLTED JOINTS

A fatigue assessment in principle is based on the comparison of failure relevant stresses to the fatigue strength. The failure relevant stresses are either calculated or measured. Depending on the used approach for the fatigue assessment the fatigue strength can be measured on the whole structure, single parts (nominal approach) or material specimens (local approach). Two approaches basically exist for a local fatigue assessment of bolts:

- the local elastic stress approach
- the local strain approach.

For the local elastic stress approach, stresses are computed at the failure relevant local spot of crack initiation with the adoption of a linear-elastic material behaviour. The calculated elastic stresses are evaluated with the fatigue strength determinate from force/stress controlled cyclic tests. Contrary to that, a fatigue assessment according to the concept of local strain is based on the (overall) elastic-plastic strain amplitude in a notch root for example. The evaluated overall strain amplitude comprises both: the linear-elastic and the elastic-plastic ratio /5/. The computation of the local stresses and strains is done in consideration of the material yield behaviour specified by a cyclic stress strain curve. The use of this concept assumes the same crack initiation behaviour for the material in the notch root and a simple material specimen under the same strain load.

Unfortunately the fatigue strength of metallic materials is strongly influenced by variables, such as mean stress /6/ or supporting effect. Both influences have to be included in a fatigue assessment if values of this material

property have to be compared or transferred from smooth or notched specimen to any other notched specimen or component with differing mean load.

The tolerable stress- and strain amplitudes are influenced by the amount and nature of mean stress. Generally, tensional mean stresses tend to decrease the tolerable amplitudes whereas compression mean stresses increase them. Even if the VDI guideline 2230 /1/ disregards the influence of mean stress on the cyclic tolerable nominal stress amplitude S_{aD} of bolts rolled before heat treatment, investigations in /7/ as well as own tests prove such a dependency to a certain degree. Approaches to consider a mean stress influence in the concept of local strain are the so-called damage parameters. A well-known approach is the damage parameter P_{SWT} of Smith, Watson and Topper /8/ which factors the local strain amplitude ε_a and the maximum stress σ_o into the assessment:

$$P_{SWT} = \sqrt{\sigma_o \cdot \varepsilon_a \cdot E} = \sqrt{(\sigma_a + \sigma_m) \cdot \varepsilon_a \cdot E} \quad (1)$$

Metallic notched specimens tolerate locally higher cyclic stresses and strains than smooth specimens. This is described by the supporting factor n_σ which is the ratio of the stress concentration factor K_t and the fatigue notch factor K_f and. The amount of support strongly depends on the local stress gradient, the strength and ductility of the used material and the material class. A famous example is the gradient concept of Siebel/Stiebler /9/. The influence of a local stress concentration is estimated with the so-called referred stress gradient χ^* ,

$$\chi^* = \frac{1}{\sigma_{\max}} \left(\frac{d\sigma}{dx} \right)_{\max} \quad (2)$$

calculated from the local stress slope with reference to the highest stress level $\sigma_{\max}$ in the notch. This concept is included in the FKM guideline *analytical strength assessment* /10/. The supporting factor is defined as

$$n_\sigma = 1 + \sqrt[4]{\chi^* \cdot \text{mm} \cdot 10^{-(a_G + \frac{R_m}{b_G \cdot \text{MPa}})}} \quad (3)$$

with the special material parameters a_G and b_G . A modification of the supporting factor for a stress gradient in the range 10 mm^{-1} and 100 mm^{-1} is described in /11/.

USE OF LOCAL CONCEPTS

The application of the local concept for a fatigue assessment of bolted joints is presented. By means of a comparison of local strength values calculated with finite element analyses from nominal fatigue limits measured on bolted structures (model bolted joints), single parts (bolts) and material specimens.

As pictured in **Figure 2** the tested bolted structure consists of a rotation-symmetric flange and nut, screwed together by a bolt rolled before heat treatment with a

dimension M10 and a strength grade of 10.9. The tested parts of flange and nut consisted either of heat treated steel (42CrMoS4) or the aluminium wrought alloy (AW-6082) and were assembled in a torque controlled tightening process. The torque is controlled by measuring the elongation of the bolt. Overall, 6 series of bolted joints with clamped parts differing in their tightening torques (M_A) and materials have been tested for their corresponding fatigue limit. **Table 1** gives an overview of the test results with values for the minimal load F_U , the cyclic bearable testing force amplitude $F_{a50\%}$, the range of scatter $1/T_S$ and the standard deviation s . All series consisted of 25 test parts and were tested for cyclic failure with test frequencies between 120 Hz and 150 Hz. The tests for fatigue limit were done by the staircase method with a fatigue limit defining number of 5×10^6 cycles. In the same way, series of single bolts and round material specimens have been tested. The corresponding characteristics and results are listed in **Table 1**.

Table 2 describes the characteristics of tested specimen. The listed stress concentration factors K_t were either calculated with a detailed FEM-model (index *FEM*) or analytically following /10/ (index *FKM*). **Table 3** shows the positions of failure in the cyclic tested bolted joints and bolts. The majority of failures appeared in the first bearing thread and the transition between shaft and thread (shaft/thread). Some of the tested screws in the bolted joints failed in the transition between head and shaft (head/shaft). Numerical examinations in /12/ have shown, that this resulted from the process of tightening which partially lead to slightly eccentrically tightened bolted joints. A direct dependency of the location of failure and the number of cycles to failure could not be found.

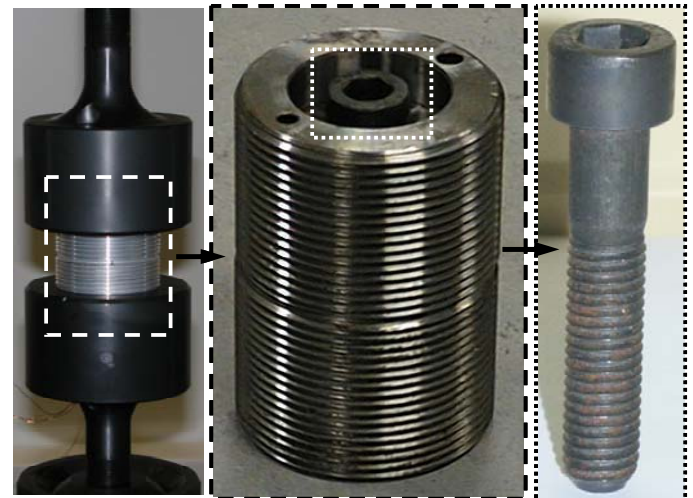


Figure 2: Left to right: test device, tested structure (bolted joints) and test part (single bolts)

With the help of finite element analysis the measured fatigue limits $F_{a50\%}$ were converted to local fatigue limits.

Figure 3 shows exemplary the finite element model of the bolted joint. According to examinations in /12/, the