Fatigue Resistance of High Strength Bolts with Large Diameters

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Abstract
This paper deals with different aspects of the fatigue resistance of high strength bolts with large diameters. After a short overview regarding the European state-of-the-art in standardization and the size effect concerning bolts the results of fatigue tests for high strength bolts M48 are presented. The tests have been carried out under different loading conditions and are compared to the normative regulations. In addition the influence of the geometric size effect has been analysed with numerical methods and fatigue strength calculations have been carried out with the local notch strain approach.
Keywords: Bolts, Fatigue, Tests, FEM

1. Introduction

High strength bolts with large diameters are mainly used in ring flange connections of large and powerful wind energy turbines. This connection is characterized by highly dynamic loadings. Thus the fatigue resistance of the bolts becomes essential. The results of the research project to this topic presented in this paper can be reviewed in detail in the final report from (Berger, Schaumann, Stolle and Marten, 2008).

1.1 Normative situation

In Europe the fatigue assessment for bolts under axial dynamic loading can be carried out according to Eurocode 3 (EC 3) or VDI guideline 2230 (VDI 2230). In the civil engineering standard EC 3 part 1-9 miscellaneous construction details are dealt with. VDI 2230 has been developed especially for the systematic calculation of high duty bolted joints in mechanical engineering. Both regulations do not define the range of application regarding the bolt diameter. However, VDI 2230 is recommended for bolt diameters smaller than 40 mm. In case of EC 3 the relevant S/N-curve detail category 50 is not verified for bolt diameters larger than 36 mm.
Figure 1 shows that the S/N-curves valid for a bolt diameter M48 differ in the beginning as well as in the stress amplitude of the fatigue limit. In general the fatigue resistance for bolts is regulated more conservatively in EC 3 than in VDI 2230.

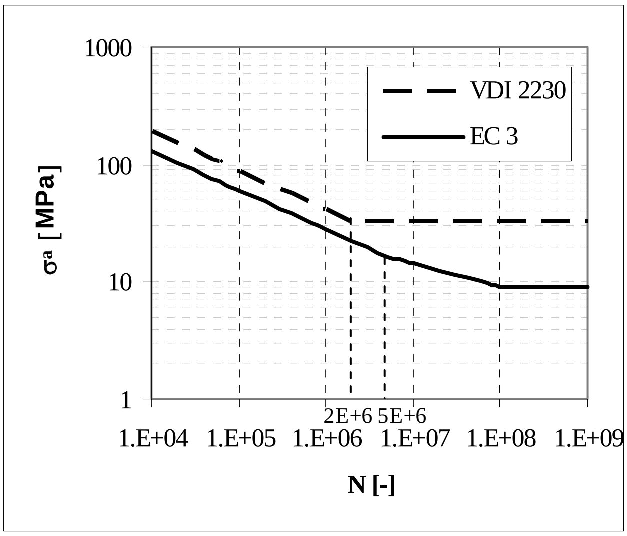

Figure 1. Normative S/N-curves for bolt M48 of EC 3 and VDI 2230

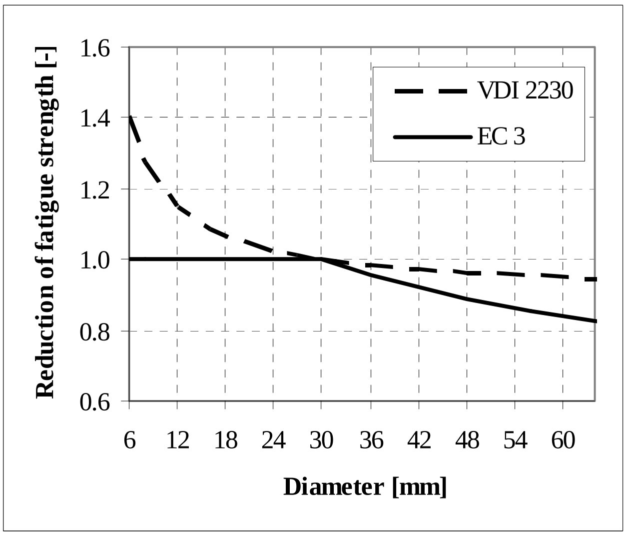

Figure 2. Influence of the diameter on the fatigue strength of EC 3 and VDI 2230

In both standards the fatigue strength depends on the diameter of the bolts. The fatigue limit for bolts rolled before heat treatment σ_{ASV} according to VDI 2230 is calculated directly via the bolt diameter d with Eq. (1). In EC 3 the size influence regarding fatigue strength is taken into account for diameters larger than 30 mm by a reduction factor k_s which decreases the reference value of the fatigue strength σ_C at 2 million cycles of the S/N-curve, see Eq. (2) and (3).

$$\sigma_{ASV} = 0.85 \cdot \left(\frac{150}{d} + 45 \right) \tag{1}$$

$$\sigma_{C,red} = k_s \cdot \sigma_C \tag{2}$$

$$k_s = \left(\frac{30}{d} \right)^{0,25} \tag{3}$$

In Figure 2 the influence of the diameter on the fatigue strength is illustrated for the two regulations. The VDI 2230 curve was scaled to M30 in order to compare both codes. Thus values greater 1.0 are possible. For bolt diameters larger than M30 a more conservative approach regarding the size effect is used in the EC 3 than in the VDI 2230.

2. Size effect concerning bolts

The fatigue strength of components decreases with their dimensions. This can be attributed to the size effect. Kloos (1976) subdivided the size effect as follows:

- geometric size effect
- technological size effect
- statistical size effect
- surface-technological size effect

The geometric size effect describes the difference in stress gradient depending on the diameter. For notched components geometrical affinity regarding the notch geometry is essential to seize the pure geometric size effect. In Figure 3 the geometric size effect is shown for a bolt thread. The stress gradient for a large diameter is smaller than for a small one because of a lesser supporting effect. This results in a larger region with high stresses for a large bolt diameter taking the same maximum stress as a basis. As the ratio between the thread pitch and the diameter is not steady but decreases with increasing diameter geometrical affinity is not at hand. Therefore different stress gradients for varying bolt diameters do not only result from the diameter but also from different elastic stress concentration factors. So the geometrical influence on the fatigue strength of bolts is not a pure size effect but a mixture of size and notch effect.

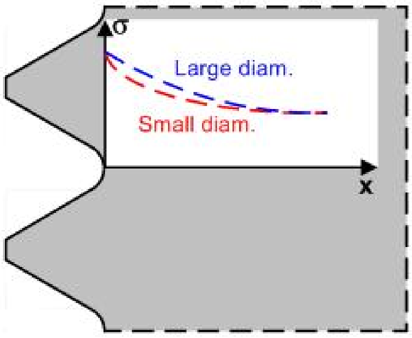

Figure 3. Stress distribution for different bolt diameters

The technological size effect considers fatigue relevant aspects of the manufacturing process. For bolts both the mechanical forming process and especially the heat treatment depend on the diameter. This results in different crystalline structures which have differing fatigue behaviour. Thus it becomes clear that the material on one hand and the manufacturing process on the other hand have to be identical for investigations concerning the technological size effect. As both material and manufacturing process change with increasing diameter it is not possible to quantify or compare technological size effects for bolts.

The statistical size effect takes into account that it is more likely to have damage relevant defects at microstructure level on large components than on small ones. Statistical size effects can be evaluated with the Weibull distribution. Similar to the technological size effect the need of identical material and manufacturing process for varying component dimensions is fundamental for an analysis of a statistical size effect. Therefore statistical size effects cannot be taken into account for bolts.

If the boundary layer of components has been hardened by special treatment it can influence the fatigue strength of the component. The main reason is that residual stresses occur in the boundary layer. The ratio between the residual stressed boundary layer and the core material varies with the thickness of the component, e.g. the bolt diameter. The dependencies between boundary layer, component thickness and fatigue strength is considered by the surface-technological size effect.

The four size effects are not equal in their consequence to fatigue strength. Normally the geometric and the technological size effect have a superior influence in comparison to the other two size effects. But for the reasons mentioned above only the geometric size effect can be investigated sufficiently for bolts.

3. Fatigue Testing

As the knowledge of the fatigue behaviour regarding large bolt diameters is insufficient fatigue tests were carried out for high strength bolts M48. The thread of the bolts was rolled before the heat treatment.
Very often the bolt axis lies eccentric to the outer line of force of a connection. A typical example for this situation is the bolted flange connection. Here the bolt is loaded by combined axial forces and bending moments. Therefore tests were performed for three different loading conditions.

a. pure axial loading
b. pure bending moment loading
c. combined axial + bending moment loading

3.1 Boundary conditions

For the pure loading conditions fatigue tests were carried out at fatigue limit level as well as in the high cycle fatigue range. The objective was to verify or extend the scope of regulations in EC 3. The tests were performed with bolts from three different manufacturers. Against normal procedure all test specimens were treated as one sample. On one hand the considering of multiple batches led to a greater variance. On the other hand by this way the validation of the S/N-curve in EC 3 is free of possible fabrication influences.
Depending on the loading condition different high frequency testing machines according to Figure 4 and 5 were used for the loading conditions a. and b.. These two test series were performed with the same testing frequency of 60 Hz to eliminate a possible frequency influence.

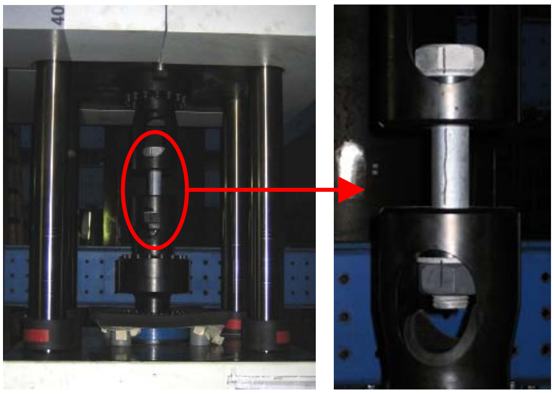

Figure 4. Testing machines for axial loading condition

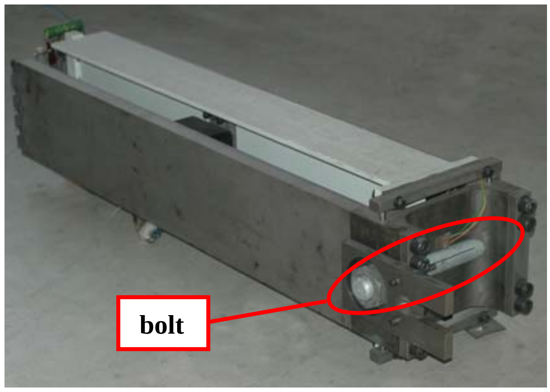

Figure 5. Testing machine for bending moments (Pat.-Nr. 102 04 258)

For combined loading condition prestressed high strength bolts M48 were tested in ring flange segments. The test setup is shown in Figure 6, the flange segment dimensions are listed in Table 1.

Table 1. Flange segment dimensions for fatigue tests

a [mm]	b [mm]	s [mm]	t [mm]	c [mm]
120	65	40	100	135

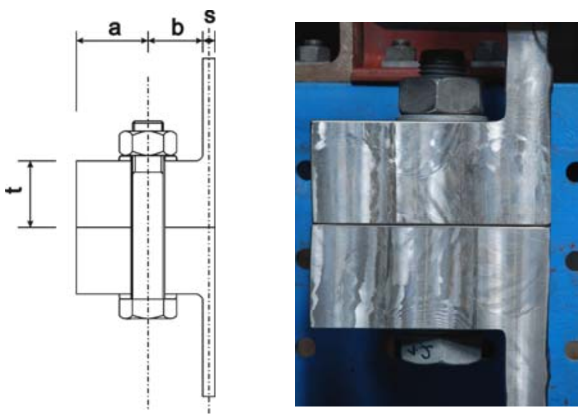

Figure 6. Flange segment connection for fatigue tests

For the flange segment tests (load condition c.) a servohydraulic testing machine was used. Due to the machine capacity the testing frequency was limited to 5 Hz, which makes these tests very time consuming. For this reason the fatigue limit was not determined. Tests were carried out in the low and high cycle fatigue range.

The main parameters of the three different test series are listed in Table 2.

Table 2. Fatigue tests parameters

	Loading condition		
	Axial loading	Bending moments	Combined loading
Test specimens	M48 10.9 HV, hot galvanized	M48 10.9 HV, hot galvanized	M48 10.9 HV, hot galvanized
Applied preload	200 kN	500 kN	930 kN
Test frequency	60 Hz	60 Hz	5 Hz
Number of tests for fatigue limit level	30	30	-
Number of tests in high cycle fatigue range	20	20	7
Failure criterion	Rupture	Initial crack	Rupture

The normative preload of 930 kN for high strength bolts M48 10.9 had to be reduced for the pure load condition tests because of limited capacities of the applied high frequency testing machines. As long as the mean stress is higher than 30% of the yield stress the influence of a lower preload should be negligible. According to Schneider (1991) bolts with threads rolled before heat treatment are approximately insensitive to the mean stress. However, only 15% of the yield stress could be applied as preload for the axial loading condition tests. The low mean stress could lead to a larger scatter. With respect to the low and high cycle fatigue range a low mean stress could also result in higher load cycles because the maximum stress is relevant for the crack approach.
The normal failure criterion for bolts in fatigue tests is rupture. For the bending moment loading condition this failure criterion was not possible due to the load application control. Therefore initial crack had to be chosen as failure criterion for this loading condition. Thus the achieved load cycles for the bending moment tests are conservative.

3.2 Test results
A comparison between the test results under axial loading and the relevant S/N-curves of EC 3 and VDI 2230 is presented in Figure 7.

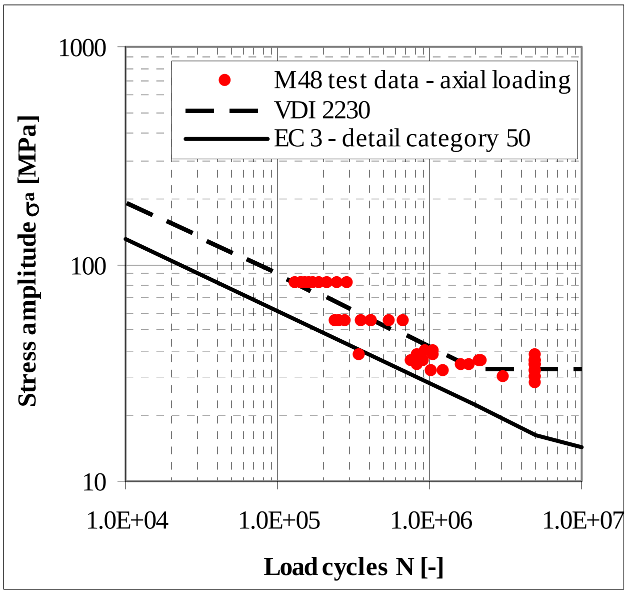

Figure 7. Comparison between scatterband of tests and normative S/N-curves

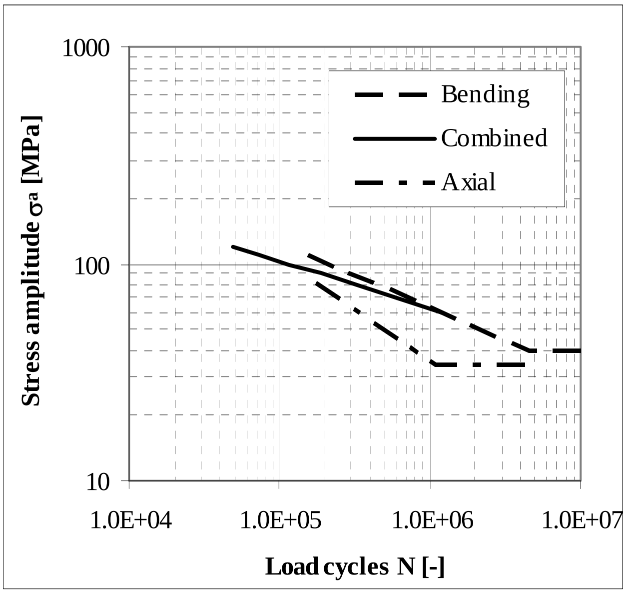

Figure 8. Comparison between the three test series – survival probability 50%

As Figure 7 shows all test results lie above the reduced detail category 50 of the Eurocode 3. The S/N-curve of VDI 2230 is not conservative. The statistical evaluation of the tested stress levels (not shown here) for a survival probability of 97.7% according to EC 3 confirms that the reduced detail category 50 is sufficient for high strength bolts M48.

In Figure 8 the statistical evaluations of the three test series are presented for a survival probability of 50%. The slopes of the curves are given in Table 3.

Table 3. Slopes for the S/N-curves in Figure 5 (survival probability 50%)

Load condition	Axial loading	Bending moments	Combined loading
Slope m	2.1	3.3	4.6

If the different slopes are not considered at first the S/N-curve for the combined load condition runs between the curves for pure loading condition. Thus the pure load conditions act as an upper and lower limit for the location of the combined curve. As the failure criterion of the bending moment tests was "initial crack" the upper border is sharper than with a rupture failure criterion.
However, the slopes of the three curves are quite different. Normally fatigue tests with bolts under axial loading lead to a slope of about m = 3 in the high cycle fatigue range. A reason for the larger slope here is the testing of bolts from different manufacturers as one sample. This leads to a larger scatter especially on the lower high cycle fatigue level and consequently to a change in slope. Another reason for the steeper slope of the curve can be the low mean stress during the tests. A low preload can cause higher load cycles for the upper high cycle fatigue range because the maximum stress that is responsible for crack growth is lower. The typical slope of S/N-curves for bolts under bending moments in the high cycles fatigue range is not well established due to a lack of tests. For anchor bolts under bending moments Frank (1980) determined also a slope of m = 3.3. Regarding combined load situations Petersen (1998) investigated the fatigue strength of thinner M20 flange segment connections. He determined a slope of m = 2.6.
The institute of material science of the Technical University of Darmstadt determined additional S/N-curves for high strength bolts M16 and M36 (for details regarding these tests see Berger, Schaumann, Stolle and Marten, (2008)). In Figure 9 the different fatigue limits for a survival probability of 97.7% are illustrated together with the normative fatigue limit of EC 3 and VDI 2230.

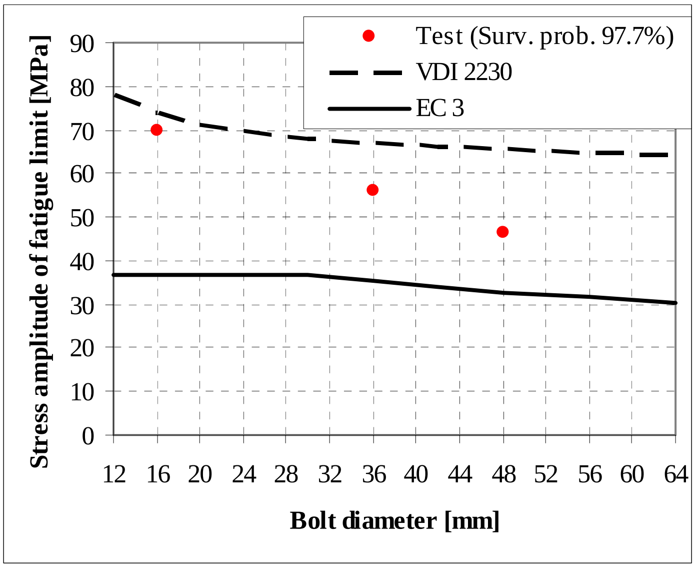

Figure 9. Determined fatigue limit (survival probability 97.7%) of different bolt diameters and normative fatigue limit (according to Berger, Schaumann, Stolle and Marten (2008))

The figure shows that the VDI guideline overestimates the fatigue limit of high strength bolts. On the other hand the fatigue limit taken from the EC 3 is partially very conservative especially for smaller bolt diameters. The figure conveys the impression that the fatigue limit falls linear with increasing diameter. But in fact the fatigue reduction over the bolt diameter is quite similar to the curve run of the VDI 2230. Here the false impression arises from the large scatter of the M48 test results due to the different fabrication lots. This leads to an underestimated fatigue limit for high survival probabilities.

4. Numerical investigations

4.1 Analysis regarding stress concentration and notch factor

The geometrical size effect can be analysed with numerical methods. With rising diameter the thread of high strength bolts becomes more sort of a fine pitch thread. The rising in notch sharpness can be expressed by the elastic stress concentration factor (SCF) K_t, which is the ratio between the maximum stress σ_{max} and the nominal stress σ_{nom} at linear elastic material behaviour, see Eq. (4). The nominal stress is calculated with the core diameter without consideration of a notch.

$$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \tag{4}$$

Formulas for the analytical calculation of the stress concentration factor of notched round bars can be found in literature. The problem is that these formulas do not consider the specific load transfer between the nut and the bolt thread. In consequence these analytical calculated stress concentration factors are too low. Realistic stress concentration factor can only be calculated with the finite element method (FEM).

The stress concentration factors presented in Table 4 have been calculated with a two dimensional axissymmetric FE-model according to Figure 10.

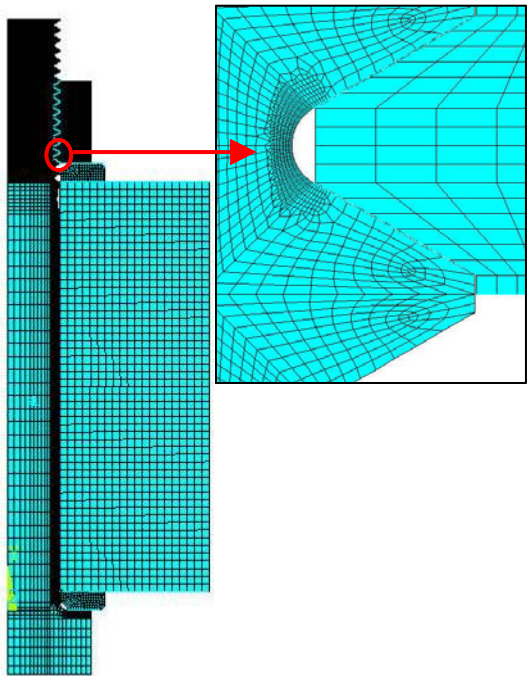

Figure 10. 2D-axissymmetric FE-model

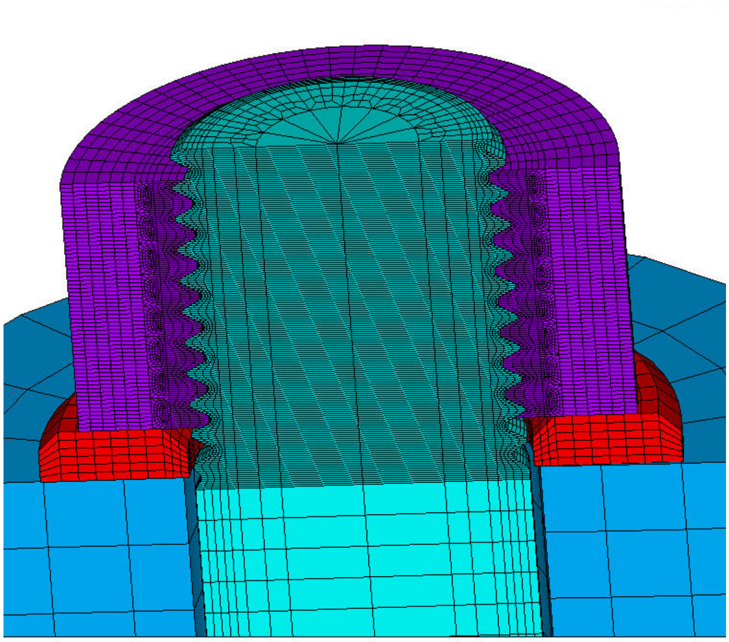

Figure 11. 3D FE-model (1. step, only half of the model showed)

Table 4. Numerically calculated stress concentration factors for different bolt diameters

Bolt diameter	M16	M20	M24	M30	M36	M48	M64
SCF (2D)	4.0	4.0	4.1	4.2	4.3	4.3	4.4
SCF (3D)	-	-	-	-	-	4.6	-

A possible influence of the thread pitch to the SFC was verified for a bolt M48 with a three dimensional model according to Figure 11. The modelling technique of the 3D-model bases on Fukuoka and Nomura (2008), the calculated SCF was determined with a submodel in a second step. The SCF determined with the 3D-model is slightly higher than the one identified with the 2D-model. However, the difference between the 2D- and the 3D-SCF is not vital so the calculations with the 2D-axissymmetric model without consideration of the thread pitch are sufficient.

The stress concentration factor gives a hint regarding the fatigue strength and is needed for local fatigue concepts. However, the notch factor K_f has a direct relationship to the fatigue strength and describes the ratio between the fatigue limit of a notched specimen $\sigma_{D,Kt>1}$ and the same specimen without a notch ($\sigma_{D,Kt=1}$). The notch factor K_f can be calculated with the stress concentration factor and the notch sensitivity factor n as shown in Eq. (5):

$$K_f = \frac{\sigma_{D,K_t=1}}{\sigma_{D,K_t>1}} = n \cdot K_t \tag{5}$$

The notch sensitivity factor n depends on the relating stress gradient $\bar{\chi}$ and the tensile strength R_m. A formula Eq. (6) for n is given in the German FKM-guideline:

$$n = 1 + \sqrt[4]{\bar{\chi}} \cdot 10^{-\left(0.5 \cdot \frac{R_m\,[MPa]}{2700}\right)} \tag{6}$$

The relating stress gradient $\bar{\chi}$ can be calculated with the supporting effect model of Siebel and Stieler using Eq. (7):

$$\bar{\chi} = \frac{1}{\sigma_{max}} \cdot \left(\frac{d\sigma}{dx}\right)_{max} \tag{7}$$

The notch factor increases with the bolt diameter. In Figure 12 the percentaged decrease of the reciprocal value of the notch factor and the fatigue limit according to VDI 2230 are compared to each other over the diameter. Both curves are scaled to the bolt diameter M12.

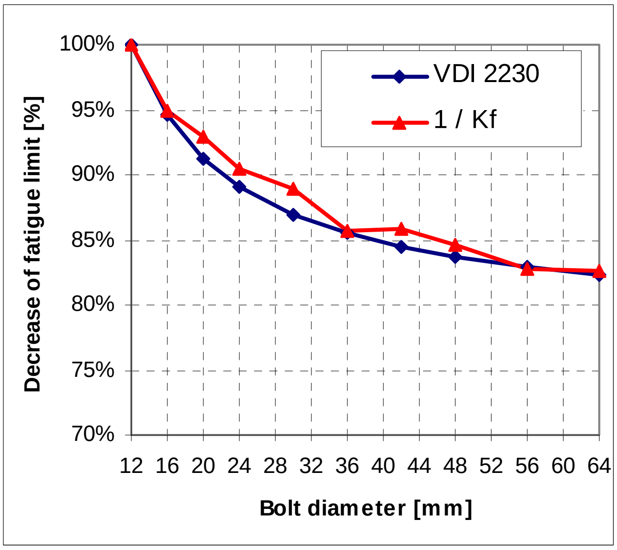

Figure 12. Comparison between percentage decrease of fatigue limit after VDI 2230 and reciprocal value of the notch factor

Figure 12 shows fair correlation between both curves. So, with knowledge of the fatigue limit for small bolt diameters the fatigue limit of large bolt diameters can be estimated reasonably good also with numerical tools.

4.2 Analyses with local notch strain approach

If also the fatigue limit of small bolt diameters is unknown a theoretical fatigue strength until initial crack can also be calculated with the local notch strain approach. With the damage parameter P the basic strain woehlercurve, which is valid for a stress ratio R = -1, can be transferred for mean stresses. As the stress condition in the bolt thread is multiaxial nature for the following calculations the modified damage parameter P_{SWT} from Socie (1986) for multiaxial loading conditions according to Eq. (8) was applied:

$$P_{SWT} = \sqrt{(\sigma_{a,1} + \sigma_{m,1}) \cdot \varepsilon_{a,1} \cdot E} \tag{8}$$

In Eq. (8) σ_a is the stress amplitude, σ_m the mean stress, ε_a the strain amplitude (all in principal direction) and E the Young's Modulus. The knowledge of the cyclic material behaviour is fundamental for calculations after the local notch strain approach. As cyclic material data are not available for many materials Boller and Seeger (1987) developed the Uniform Material Law (UML) which can be used instead. In Figure 13 the P_{SWT}-woehlercurve has been calculated with the UML and with the material 34CrNiMo6, which has been used for the tested hot galvanized bolts M48.

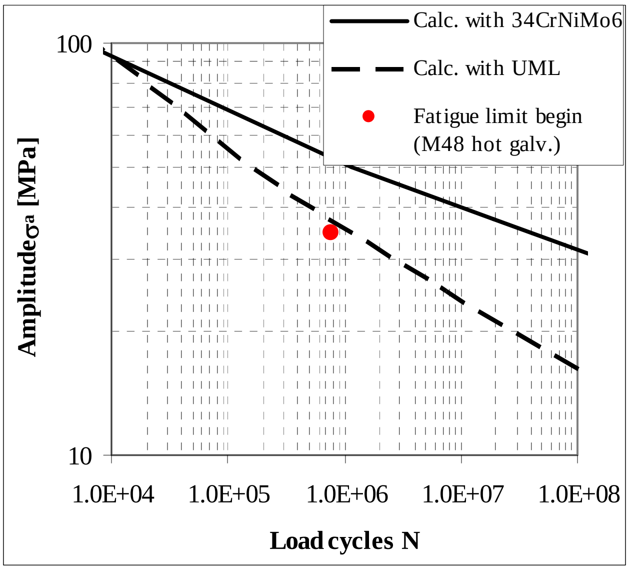

Figure 13. P_{SWT}-woehlercurves for different materials and determined fatigue limit for M48 from tests (survival probability 50%)

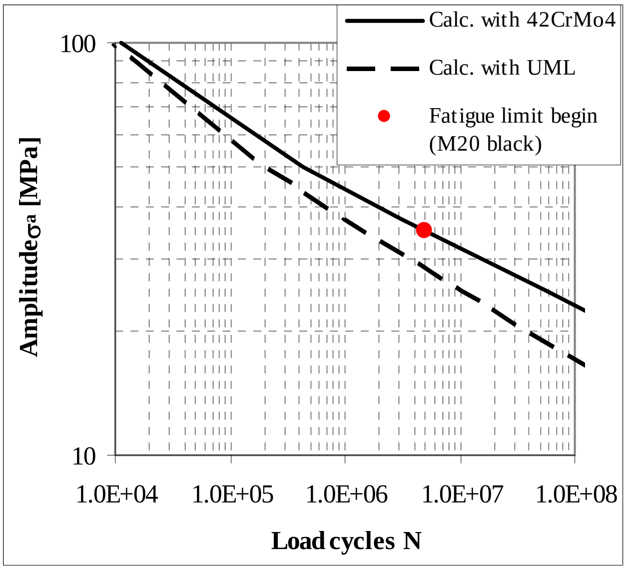

Figure 14. P_{SWT}-woehlercurves for different materials and determined fatigue limit for M20 from tests (survival probability 50%)

Figure 13 gives the impresssion that the calculation with UML is far better than the one with the real material 34CrNiMo6. But before the background that the UML bases on a scatter band of a large number of different materials (including the material 34CrNiMo6) a calculation with UML cannot be as precise as with the real material. There are two possibilities for this circumstance here: First, as already mentioned above the testing of different fabrication lots within one sample and the low mean stress leads to a more conservative fatigue limit and beginning point. Second, hot galvanized bolts have a lesser fatigue strength than black bolts. The influence of the coating has not been considered in the calculation. The comparison in figure 14 between the calculated and the determined fatigue limit of black bolts M20 from one lot with normative preload shows a better correlation for the real material data than for UML.
For most applications the UML is sufficient for calculation. However, in cases with very high mean stresses like in prestressed bolted connections the usage of the UML leads to larger variations for the fatigue strength in comparison to calculations with the real material data. For a more realistic estimation of the fatigue strength of highly prestressed bolts the strain woehlercurve for the specific material must be at hand.

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