DAMAGE TOLERANCE ANALYSES OF BOLT/NUT ASSEMBLIES

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For a reliable integrity analysis of fasteners to be performed it is of paramount importance that accurate knowledge about load distribution and stress intensity factors is available.
In the current study the load transfer (tension and bending) as well as stress intensity factor solutions for various crack locations and sizes have been derived by advanced 3D FE analyses including threaded bolt and nut. In addition, a large number of fatigue crack growth tests with Titanium and Steel fasteners have been performed to verify the solutions.

INTRODUCTION

The crack initiation, propagation and failure behaviour of bolt/nut assemblies subjected to fatigue tensile loads is complicated. However, the state of art in experimental techniques and finite element analyses has reached a level of refinement that allows a detailed study of the load transfer through the bolt shank and nut and also, allows determination of stress intensity distributions along crack fronts.

In this study full 3-D-finite element meshes of metric M8 x 1.0 and M12 x 1.25 bolt/nut assemblies are used to analyse the load transfer and the stress intensity factor distribution. In the FEM mesh quadratic isoparametric elements are used and the $\sqrt{r}$ terms in the displacement field are obtained by collapsing an element face on the crack front and by shifting nodes in the quarter position. The thread on bolt and nut is modelled as helical surfaces in the 3D-space. On the nut the thread run out is also modelled realisticly. Rigid contract of bolt and nut is

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assumed in the contact area on one side of the thread.
After analyses of the load transfer in the undamaged configuration different crack shapes (straight, a/c = 0.645 and a/c = 1.0) and sizes are introduced in the highest loaded thread under the nut. Both the load transfer and the stress intensity factor distribution are analysed for the damaged bolt/nut assembly.

As a part of the experimental programme the bending stresses in the bolt shank were measured using strain gauges. Using a crack front marking technique the evolution of the crack front was registered. From the fracture surface the crack growth rates were determined and using a calibration curve, da/dn versus ΔK, the stress intensity factor distribution was measured for a number of crack fronts. The results obtained for cut and rolled threaded bolts and for different materials are discussed in relation with the FEM results. From the results it is concluded that Young's modulus is not playing a significant role in the load transfer through the undamaged bolt/nut assembly. In the presence of larger cracks the stiffness of the bolt shank plays a dominant role. An analytical bar model is used to account for the bending stiffness of the part of the shank not included in the FEM model.

3-D FEM ANALYSES

Essential in the finite element meshing is the proper modelling of the thread runout of the first thread in the nut including the changing width of the contact area of the run out and the flange of the corresponding thread on the bolt.
The bolt/nut assemblies analysed are Metric M8*1.0 and M12*1.25. The discretized bolt length is 7 times the pitch: 4 threads under the nut and 3 threads to model the free shank part of the bolt. One additional thread is used to model the thread run out in the nut. The mesh generation is extensively discussed in reference 1. In figure 1 the mesh, deformed as a results of a prescribed displacement of the end plane of the shank part, is shown. The displacements of the attachment plane on the nut are suppressed in the direction of the axis of the bolt/nut combination. In other directions unloaded fixation is applied for reasons of numerical stability. Clearly visible in the left part of figure 1 is the end of the thread run out and its effect on the deformations. In the right part of figure 1 the deformed mesh in an intersection with a plane through the axis is shown. Such plots were used to check the mesh design and the boundary conditions.

The finite elements mesh consists of about 7000 higher order elements and the problem size is well over 150.000 degrees of freedom. Even with this refined mesh some simplifications had to be accepted in the model:

- The contact surface of nut and bolt is simulated by rigid coupling of displacements of all nodes in the contact surface. Thus, slip and friction are not included in the model.
- The outer surface of the nut is assumed to be cylindrical in stead of hexagonal. The height of the nut is limited to 4 turns of the thread number plus the thread run out. It was verified that this number is sufficient to distribute the loads in a realistic way over the threads and over the attachment surface of the nut.
- The model is assumed to be linear elastic.

Load transfer

The solution obtained for the tensile stress in the neck of the thread is shown in figure 2. It is seen that the maximum stress observed in the M8*1.0 mm bolt/nut assembly is 4.68 times the applied remote load. In table 1 results obtained for M12*1.25 mm are given and a comparison is made with the K_t values calculated for the threaded rod loaded in tension (no nut).

Stress Intensity Factor distribution

According to the survey in table 2 cracks were introduced in the FEM meshes. Stress Intensity Factor distributions were calculated for each geometry using the method of virtual crack extension (de Lorenzi Method (2)). The results were checked using a crack opening displacement extrapolation technique. The agreement is good. Some typical results are shown in figures 3 and 4. It is observed that the Young's modulus of the nut material relative to the material of the bolt is of minor importance. This is understood when it is recognized that the major load transfer is through the highly loaded first thread under the nut. The difference in load transfer for the case of nut loading compared to remote loading is obvious. It is known that the high gradient in Y near the free surface (the neck of the thread) is strongly depending on the local geometry of the crack front, so, some care should be taken into account when conclusions are drawn for this part of the crack front.

Most striking is the effect of the boundary conditions on the stress intensity factor. Constant load, simulating that the effective bolt length L/d is infinite gives much higher values then prescribed edge displacements (L/d = 1). This implies that the effective bolt length and stiffness are important parameters in the presence of larger cracks. To account for this effect the part of the bolt shank not modelled in the FEM mesh was described using the analytical bar model shown in figure 5. Continuity conditions at the interface with the FEM model require that the bending moment M and the angle of rotation ψ are the same as in the FEM model. Thus, the following relation can be derived for computation of the bending stress at the interface

$$\sigma_b = \psi_t^{FEM} \sigma_t \Big/ \left[\frac{L - D}{2ED} - \psi_b^{FEM} \right] \tag{1}$$

where ψ_t^{FEM} and ψ_b^{FEM} denote the angular rotation of the interface plane induced by a unit load in tension, respectively, in bending of the FEM model. σ_t is the applied tensile load. Using the FEM solution of the Stress Intensity Factor distribution under bending loads in combination with eq. 1 the effect of bolt shank length can be included. Some of the results are indicated in figure 6. In reference 1 more details are given.

EXPERIMENTAL PROGRAM

An extensive test programme was executed using aerospace quality bolts and nuts. The test parameters are the material (7075T73 Aluminium Alloy Ti-6Al-3V Titanium and A286 steel), size of the thread (M12 * 1,25 and M8 * 1.0), the manufacturing process (rolling or cutting the thread) and the crack shape (straight, a/c = 0.645, a/c = 1.0) and size.

Load transfer

For the undamaged bolt/nut assemblies bending stresses upto 30 percent of the applied load were measured half way on the shank (see Fig. 5). In the presence of a crack the bending stress is highly affected by the crack size.
The former effect is caused by imperfections in the geometry and the alignment of the thread. The latter was predicted properly by the hybrid model discussed in the previous section. In reference 1 details are given.

Determination of Stress Intensity Factor distributions

Precracked bolt/nut assemblies were loaded in fatigue by application of constant amplitude tensile loads. A marker loading technique was used to register the crack fronts at different stages in the fatigue life of the specimens. Using a calibration curve (da/dn versus ΔK), obtained for the same material but known geometry and K-factor solution, the amount of crack growth measured between the markings was converted into K factor distributions. The main results are plotted in figure 6. In the same figure the results of hybrid FEM/Analytical model are given for comparison.

CONCLUSIONS

1. Using the current state of the art technology in FEM analysis a 3D mesh can be created and analysed of a bolt/nut assembly.
2. Measured K factor distributions agree well with predictions based on FEM.
3. The effect of shank stiffness and length is important for larger cracks. A model to describe this effect was applied successfully.

REFERENCES

(1) Koning, A.U. de, Lof, C.J. and Schra, L., Assessment of three-dimensional stress intensity factor distributions for Bolt/Nut assemblies loaded in tension. ESA Contract Report NLR CR 94439 L, 1994.

(2) Bakker, A., The three-dimensional J-Integral. Ph.D. Thesis, Delft University of Technology, 1984.

(3) Toribio, J., Sanchez-Galvez, V., Astiz, M.A., Stress intensification in cracked shank of tightened bolt. Theoretical and Applied Fracture Mechanics 15, pp. 85-97, 1991.

TABLE 1 - Stress concentration factors calculated by 3D-FEM for the undamaged thread. Remote stress based on basic minor (core) diameter d: $s = \sigma_o = 4P/\pi d^2$ and the bolt load P

Undamaged metric threaded rod	p/d	Stress concentration factor K_T Threaded rod loaded in tension	Threaded rod loaded via the nut
M 8*1.00	0.125	1.80	4.68
M12*1.25	0.104	2.32	5.20

TABLE 2 - Matrix of cases used for stress intensity factor computation

Type of thread	Shape of the crack front	Size of the crack a [mm]	relative size*) of the crack a/d
M12*1.25	straight	0.68	0.06
		1.50	0.14
		3.00	0.29
		4.67	0.45
	a/c = 0.645	1.06	0.10
		2.40	0.23
		3.60	0.34
		4.80	0.46
	a/c = 1.000	1.20	0.11
		2.05	0.20
		3.55	0.34
		4.60	0.44
M8*1.00	a/c = 0.645	0.90	0.13
		1.50	0.22
		2.29	0.34
		3.15	0.46

*) d based on nominal projected core diameter d = 10.466 mm (M12) or 6.773 mm (M8).

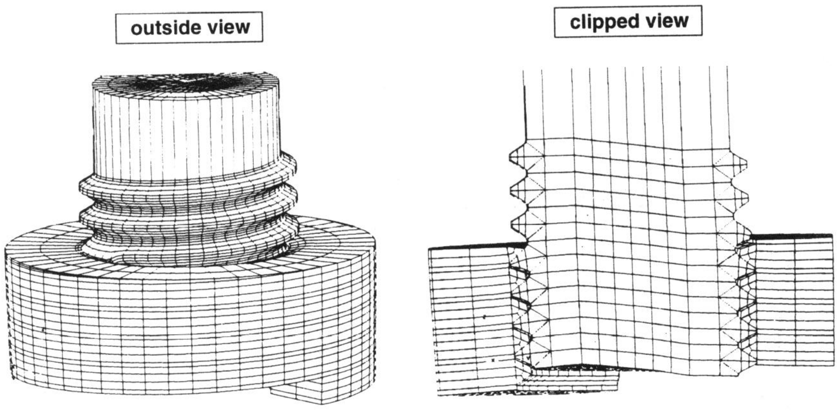

Figure 1. The mesh deformed under remote loading

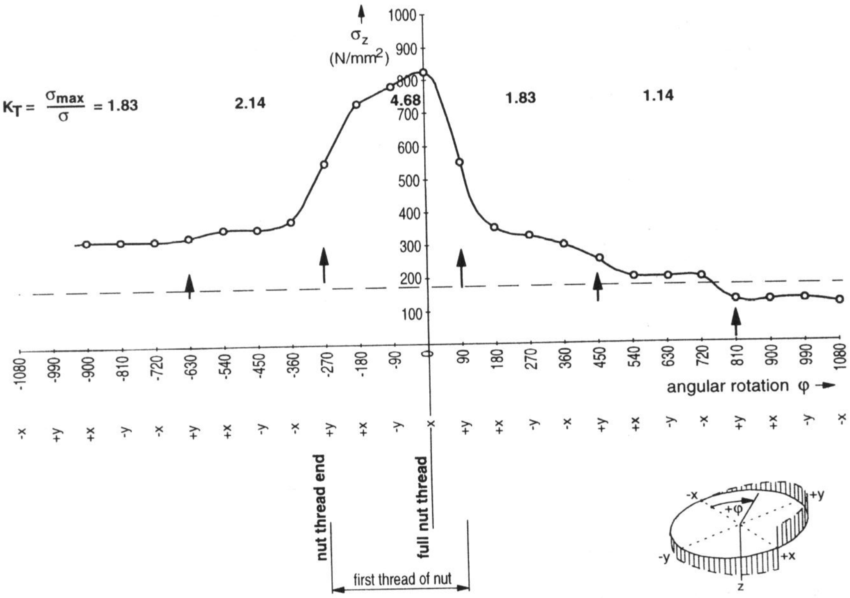

Figure 2. Calculated tensile stress in the neck of the thread (nut loading, σ = 175 MPa, M8×1.0 mm)

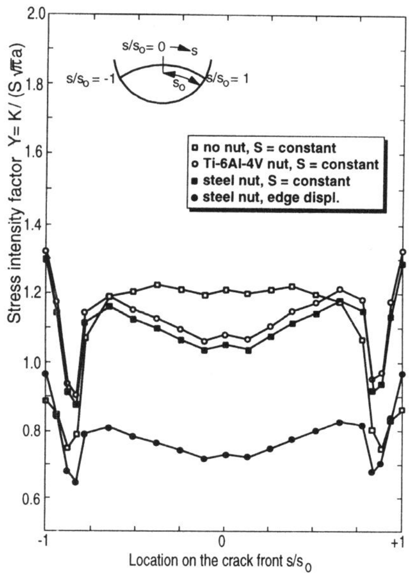

Figure 3. *3D-FEM results obtained for the stress intensity factor distributions along a crack front (a = 3.6 mm, a/c = 0.645) in a Ti-6Al-4V bolt M12 × 1.25, S = 100 MPa*

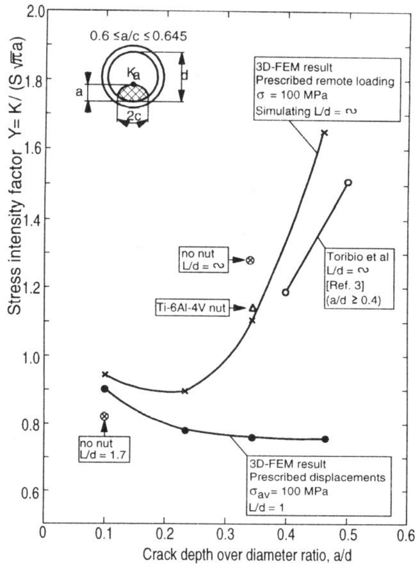

Figure 4. *Stress intensity correction factors Y computed for an M12 bolt/nut assembly. Steel nut unless indicated otherwise. Ti-6Al-4V bolt, effective length L/d*

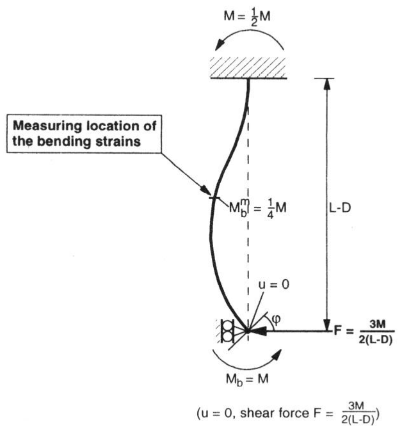

Figure 5. *The cylindrical bar model of the part of the bolt shank not included in the FEM model. M_b is the bending moment at the interface with the FEM model*

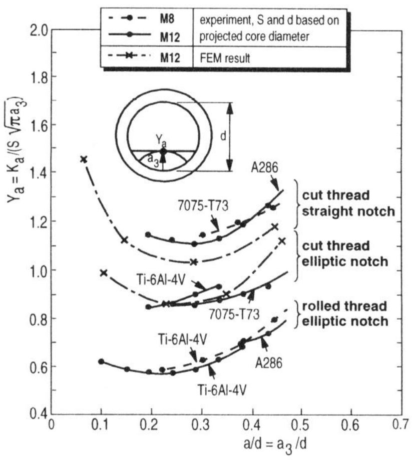

Figure 6. *Stress intensity correction factors determined for the centre point of the crack front. Final results after evaluation*