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FATIGUE LIFE PREDICTION OF COMPOSITE BOLTED JOINTS WITH BOLT FAILURE

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Composite structures are often joined with bolted joints. One common failure mode in fatigue of composite bolted joints is bolt failure. Double lap bolted joints with six fasteners were constant amplitude fatigued at R=-1. The six bolt joints were FE-modeled and the stresses in the fasteners were calculated. Double lap bolted joints with two fasteners were constant amplitude fatigued at R=-0.2. Some of the joints were subjected to repeated overloads. The joints, which were subjected to repeated overloads, had a longer fatigue life than the joints, which were constant amplitude loaded. This behavior is typical for metal fatigue and shows that the fatigue life of the joints is determined by the fatigue life of the fasteners. The average peak tensile stress in the bolts was plotted against the fatigue life of the joints and results for joints with 4 mm and 6 mm bolts collapsed into one scatter band. From this it is possible to predict the fatigue life of bolted joints.

INTRODUCTION

The amount of composites used in aircraft and other advanced applications increase as the cost of composite material decreases. Composite structures are often joined with bolted joints. During the service life of aircraft the joints will be fatigue loaded and it becomes important to predict the fatigue life of joints during design of aircraft. Design against fatigue is often done based on coupon fatigue testing or quasi-static coupon testing to 150% of design limit load. Since the amount of different composite bolted joints in an aircraft is large it becomes expensive to test each type of joint. Therefore, the potential cost saving of a method to predict the fatigue life of bolted joints is large.

Two common failure modes in fatigue of composite bolted joints are bolt fracture and hole elongation. The specimens can also fail due to shear out and net section failure. Bolt failure has been observed in several investigations. Galea and Saunders[1] have tested joints with countersunk fasteners and the failure mode was bolt failure. In another investigation by Saunders et al.[2] of joints with countersunk fasteners bolt failure also occurred. The nucleation site for the fatigue crack in the fasteners appeared to be at the shear plane of the joints. Ramkumar et al.[3] has

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reported fastener failure for countersunk fasteners in bolted joints. Chen[4] observed joints with bolt failure of protruding head bolts. Bolt fatigue failure was most frequent for situations with a large thickness to diameter ratio, probably because of the higher bending moments imposed on the bolts. Trodus[5] and Starikov and Schön[6] have also reported fatigue failure of fasteners.

In a fatigue study bolted joints with different number of bolt rows and lay-up were tested. The joints broke due to bolt failure. If the average bearing stress for the bolts was calculated and used to plot the data the fatigue lives collapsed into a single master curve which could be used for fatigue life prediction, Starikov and Schön[7].

The objectives of the investigation have been to study if bolt fracture of bolted joints can be characterized as composite material failure or as metal failure. In addition fatigue life prediction should be attempted based on classical metal fatigue of fasteners.

EXPERIMENTAL AND NUMERICAL PROCEDURE

The specimens were manufactured from carbon fiber/epoxy, HTA7/6376, composite. A quasi-isotropic stacking sequence was used, $\left[(\pm 45/0/90)_3\right]_s$ for the outer plates and $\left[(\pm 45/0/90)_6\right]_s$ for the central plate. For some specimens thinner laminates were used with the stacking sequence $\left[(\pm 45/0/90)_2\right]_s$ for the outer plates and $\left[(\pm 45/0/90)_4\right]_s$ for the central plate. The nominal ply thickness was 0.13 mm and the geometry of the specimens can be seen in Fig. 1a. The specimen plates were joined with titanium bolts with protruding head, Hexagon type. The 6 mm bolts were tightened to 9 Nm torque or to contact (finger tight). The 4 mm bolts were tightened to 3.5 Nm torque. A lateral support was used to prevent buckling in compression. Teflon plates were placed between the specimen and the lateral support in order to reduce the friction forces and the support was tightened to contact. All tests were done at room temperature and ambient relative humidity. The frequency of the fatigue loading was adjusted such that the temperature of the bolts did not exceed 35 ^{o}C. The peak to peak grip displacement was measured at the beginning of fatigue testing and failure was set to have occurred when the peak to peak grip displacement had increased with 0.8 mm. The constant amplitude fatigue tests were done with a load ratio R=-1, $R=\frac{\sigma_{\min}}{\sigma_{\max}}$.

Specimens with two bolts, see Fig. 1b, were tested with constant amplitude loading at R=-0.2. The same type of specimens was also tested with repeated overloads, see Fig. 2. First the specimen was subjected to 500 cycles at R=-0.2 and 50 kN peak tensile load. Then one overload cycle occurred with a larger tensile component, 65 kN. It was then followed with 4000 cycles at R=-0.2 and 50 kN peak tensile load. After that one overload cycle occurred. Again 4000 cycles followed at R=-0.2 and 50 kN peak tensile load. After which one overload cycle occurred. This was then repeated until failure.

The joints with 6 bolts were modeled with the FEM. This was done with the in-house developed *p*-version program STRIPE, Andersson et al.[8]. In the model, symmetry along the center in the load direction was used to reduce the number of bolts from six to three. Since the joints are double lap symmetry at the center in the thickness direction was also used. As a result one quarter of the joint had to be modeled. The washers at the fasteners were modeled and friction less contact was used at all surfaces in contact. Quadratic elements were used and the model contained 29280 elements. Homogeneous material data were used for all parts in the model and no clearance was present at the bolts.

RESULTS AND DISCUSSION

The average quasi-static strength of specimens with two bolts was 75 kN. The strength was the same in compression as in tension and the specimens failed due to bearing failure. The constant amplitude loading was done at 50 kN and the fatigue life can be seen in Fig. 3. In the same Figure can the fatigue life be seen for specimens subjected to the same constant amplitude loading with a few overloads, see Fig. 2. Three of the specimens had a similar fatigue life as the ones without any overloads. Three specimens had a longer fatigue life than the specimens without any overloads. All specimens failed due to bolt failure. Overloads are known to cause an increased fatigue life in metal fatigue. In composite fatigue they are known to cause a decrease in fatigue life. The fatigue life of composites is often determined by the largest load cycles in a load spectrum.(Nyman et al.[9], Schön and Blom[10], and Schön[11]) This would suggest that the fatigue life of composite bolted joints is determined by the fatigue life of the titanium fasteners and not by the composite. It could have been the composite that were worn down and when it had reached a certain damage level the fasteners could have broken quickly.

In Fig. 4 can constant amplitude fatigue results at a load ratio R=-1 be seen. The results have been potted as applied peak load versus number of cycles to failure. Since the failure mode for nearly all specimens is bolt fracture or hole elongation, which are related to the fasteners, the applied load is related to the load each fastener can transfer at that fatigue life. All specimens have six fasteners.

The specimens with 6 mm fasteners which were tightened to contact, finger tight, had a slightly shorter fatigue life than the specimens with 6mm fasteners which were tightened to 9 Nm torque. Since very little load is transferred by friction when the fasteners are tightened to contact compared to when they are tightened to 9 Nm torque, more load will be transferred by the fasteners in the finger tight specimens. This results in higher stresses in the fasteners and on the hole surface, which result in a shorter fatigue life. Results for both fastener torque levels are on a straight line.

In some specimens the 6 mm fasteners were replaced with 4 mm fasteners. The two specimens fatigue loaded at the highest load level failed due to hole elongation. The remaining specimens failed due to bolt fracture. If the result for the specimen loaded at the lowest load level is excluded the slope of the fatigue curve is similar to that of the specimens with 6 mm fasteners which failed due to bolt fracture. In general, the fatigue curve for specimens with 4 mm fasteners has been shifted to shorter fatigue lives compare with other specimens. This is reasonable since a reduction in fastener diameter should reduce the fatigue life when the failure mode is bolt fracture. The specimen loaded at the lowest load level had a surprisingly long fatigue life. A similar behavior can be seen for the specimen with 6 mm fasteners and 9 Nm torque loaded at the lowest load level. The observations suggest a threshold type behavior.

Some specimens were prepared with thinner composite plates. All, except the specimen loaded at the highest load level, failed due to hole elongation during fatigue loading. The specimen at the highest load level failed due to net-section failure, which is the same failure mode as during quasi-static testing. The slope of the fatigue curve is flatter than for the other joints, which failed due to bolt fracture. Hole elongation is a composite material failure mode, where the material is “worn” away from the hole surface. As the hole “length” increases the grip displacement of the joint increases and the specimen is considered broken when the peak to peak amplitude of the grip displacement has increased 0.8 mm. It has been reported earlier that composite material failure results in a flatter fatigue curve than bolt fracture.[7] It is not surprising that the fatigue life is reduced at a given applied load when the composite plates become thinner. But, at low load levels and long fatigue life the fatigue life of the thin specimens is nearly the same as for the thicker specimens. This shows the importance of which failure mode that determines the fatigue life. Failure of bolts is due to “metal fatigue” which has a steeper fatigue curve than composite fatigue.

Since joints, which fail due to bolt fracture during fatigue loading, show metal characteristics it might be possible to predict fatigue life of those joints with classical metal fatigue of the fasteners. The peak tensile stress in the fasteners was calculated and normalized, see Fig. 5. For all three joints the first bolt row had the highest tensile stress. The middle row, bolt row number two, all had lower tensile stresses than the first bolt row. In the joints with 6 mm fasteners the third bolt row had nearly as high stress in the fasteners as the first bolt row. For the joint with 4 mm fasteners the tensile stresses in bolts in bolt row number three decreased further. In fatigue experiments using the joint with 6 mm bolts tightened to 9 Nm and 3.12 mm thick outer plates it was found that bolts in the first bolt row often were the first to break followed by bolts in the third bolt row. The second bolt row had least number of failed bolts, Schön and Nyman[12].

During fatigue loading hole wear will occur and the wear particle will fill any space between the fasteners and hole surface. Thereby, reducing the grip displacement of the joints, Starikov and Schön[7][13]. This will probably cause redistribution in load transfer between the bolts in such a way that the bolts transfer close to the same amount of load.[7] This would probably also cause the peak stress in the different bolts to become more even. Therefore, the average peak tensile stress for the bolts was used to predict the fatigue life. The stresses in the bolts was calculated at one load level and then the stresses at other load levels were calculated assuming a linear dependence with load. Contact problems are nonlinear but at the high load levels used in the joints the contact area will probably not change greatly with load. Making the stresses in the fasteners close to linear with respect to applied load.

In Fig. 6 has the average peak tensile stress in the bolts been plotted against number of cycles to failure for tested joints. Since the average tensile peak stress for the joints with finger tight fasteners and with fasteners tightened to 9 Nm have been calculated using the same FE-solution they will, relative to each other, look the same as the original data in Fig. 4. In Fig. 4 there was a large difference between the results for joints with 6 mm and 4 mm fasteners. But, in Fig. 6 they are on top of each other. The two specimens with 4 mm fasteners who have the highest average peak stress in the fasteners broke due to hole elongation followed by bolt failure of the bolts. All the remaining joints with 4 mm and 6 mm fasteners and 3.12 mm thick outer plates broke due to bolt failure. The joints with thin plates all failed due to hole elongation. This means that the bolt holes were “worn out” before the bolts would break. As a result they are to the left of the bolt failure curve in the Figure. A thinner joint reduces the bending stresses in the bolts. The joints will break due to the failure mode that has the shortest fatigue life. In the Figure rotating beam fatigue data, R=-1, for Ti-6Al-4V, the same titanium alloy as in the fasteners, has been included.(Yoder et al.[14] and Saritas et al.[15]) The fatigue data have been read from graphs. The titanium fatigue data agrees well with the average peak stress in the bolts. Using the fatigue curve in Fig. 6 it is possible to predict the fatigue life of joints with bolt failure loaded at R=-1. For other load ratios a new fatigue curve need to be measured. Galea and Saunders[1] fatigue tested joints with countersunk fasteners at room temperature and at hot/wet environment. In both cases specimens failed due to bolt fracture. In the case of hot/wet environment the fatigue life was significantly reduced. If the fatigue life had been governed by metal fatigue of the fasteners the fatigue life would have been independent of the environment. Fractographic work found significant hole wear and damage of the composite at the hole surface, which are composite material related damage mechanisms. This suggest that in the transition region between hole wear and bolt fracture it is not sufficient to calculate the stresses in the bolts with a quasi-static FE-model which do not take into account the changes in geometry of the bolt hole during fatigue loading. For the specimens with 6 mm fasteners tightened to 9 Nm tested at low load levels it has been found that the hole wear and composite damage at the hole is small.[13]

CONCLUSIONS

If the amount of hole wear is small and the joints fail due to fastener fracture the joint shows characteristics of “metal fatigue”. For those joints the fatigue life of the joints can be predicted by calculating the peak tensile stress in the bolts.

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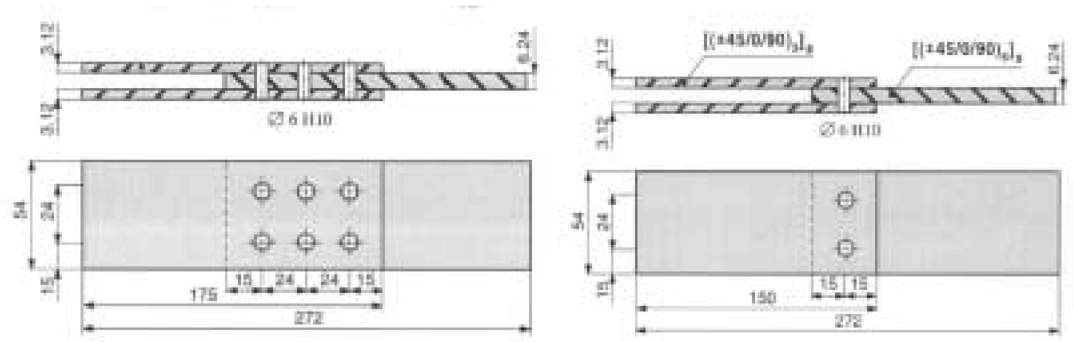

FIGURE 1 Schematic of bolted joints. a: Bolted joint with six protruding head fasteners, 6 mm and 4 mm. Some joints with thinner composite plates, 2.08 mm outer plates and 4.16 mm central plate. First bolt row to the right and third bolt row to the left. b: Joint with two fasteners.

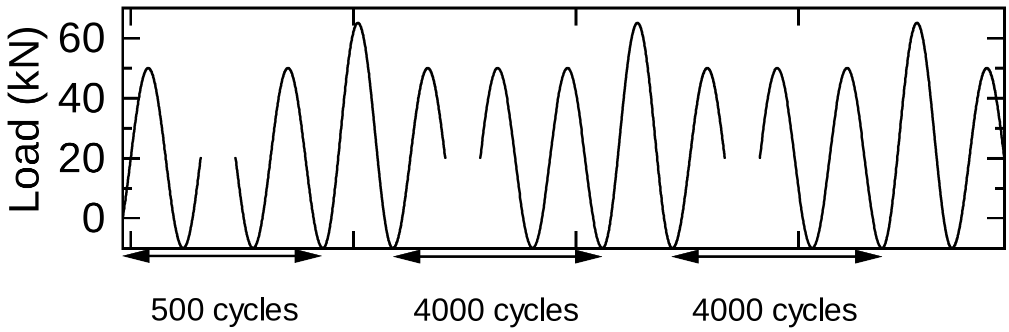

FIGURE 2 Schematic of load spectra with repeated overloads.

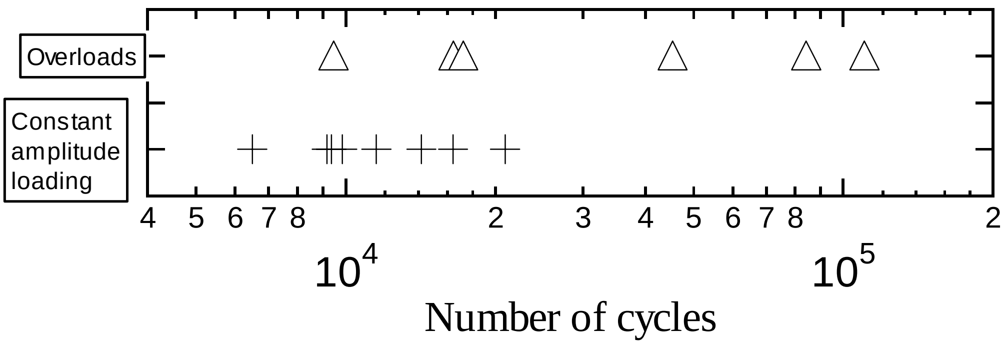

FIGURE 3 Fatigue life of constant amplitude loaded specimens and specimens subjected to repeated overloads.

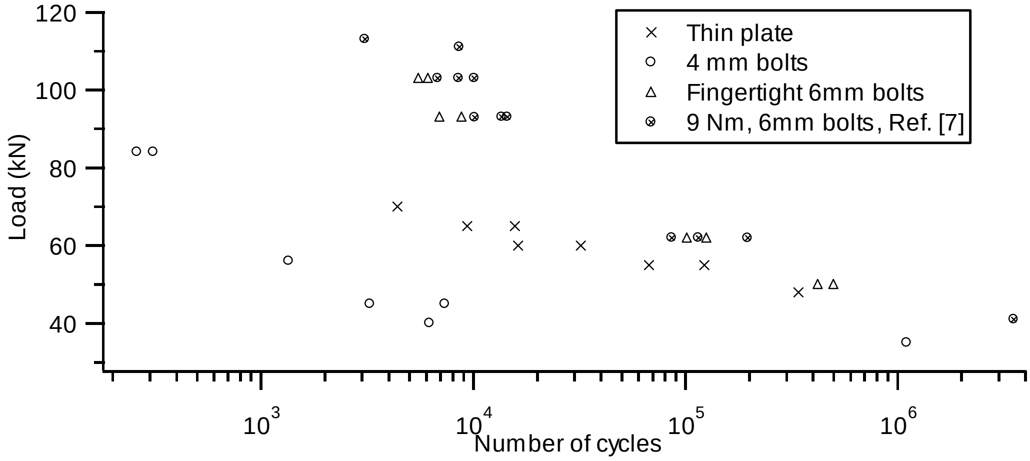

FIGURE 4 Fatigue life of bolted joints.

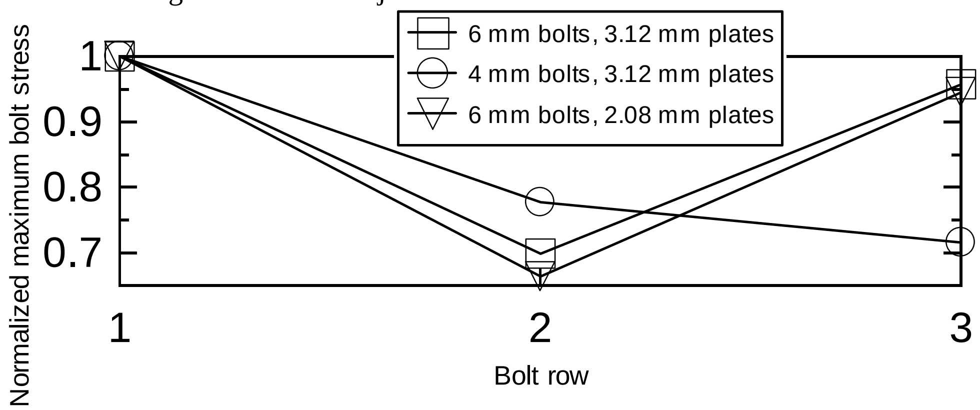

FIGURE 5 Normalized peak tensile bolt stress in six bolt joints.

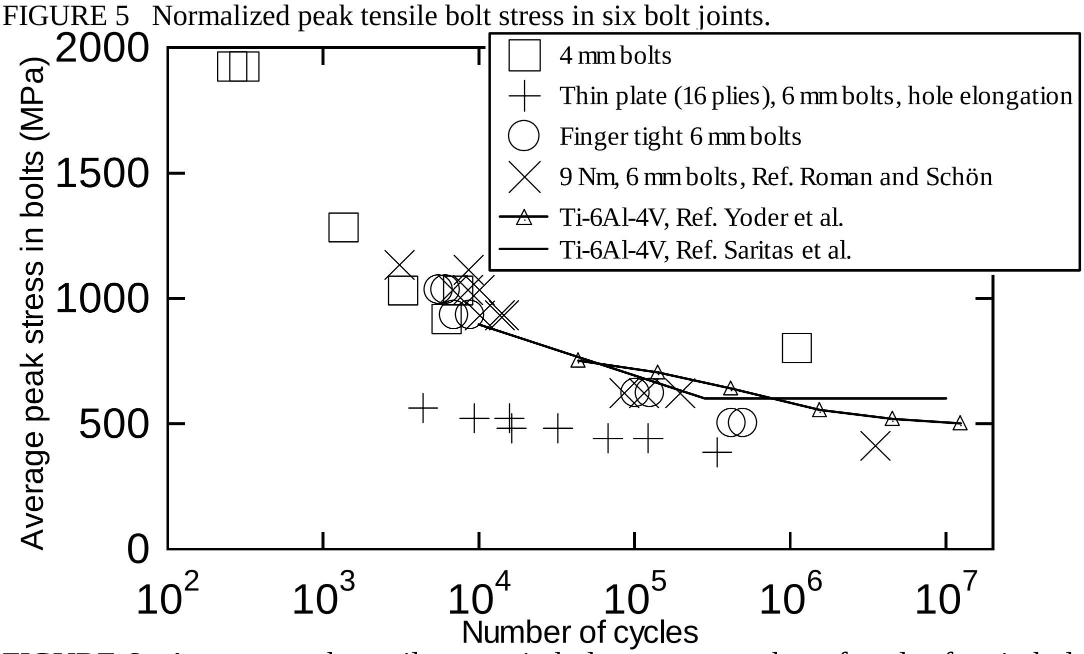

FIGURE 6 Average peak tensile stress in bolts versus number of cycles for six bolt joints.