THREE–DIMENSIONAL STRESS ANALYSIS OF THE T–BOLT JOINT

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SUMMARY: A three-dimensional finite element model of the joint has been developed. The model has been used to predict the stress field of the laminate in the joint area, assuming homogeneity and linear behavior of the material. The modeled geometry is a T-bolt that joins a coupon of glass fiber–epoxy and a block of steel. The results obtained from the model have been verified experimentally using both electric and optical strain sensors. In the discussion of the results, a comparison between the stresses predicted for the T-bolt and those of an equivalent double lap bolted joint has been performed.

KEYWORDS: Composite Joints, Stress Analysis, 3D Models.

INTRODUCTION

One of the most critical joints in a wind turbine is the joint between the blade root and the hub. While it has a great responsibility in the wind turbine performance and integrity, its inspection is very difficult and forces to stop the turbine. Another singularity of this connection is the huge thickness of the blade root laminate (typically > 60mm).

On the other hand, there is a marked trend in the wind energy industry to build blades as large as possible. Therefore, over the years, there is an increase in the joint loads, but at the same time, the different elements need to be relatively lighter and cheaper.

In this scope, many different solutions have been proposed, but only few of them have demonstrated to be able to fulfill all the requirements of the modern wind turbines and are being used at industrial scale. One of the most successful of these solutions is the so-called T-bolt connection, shown in figure 1. It consists mainly of the nut, i.e. a steel cylinder placed transversally to the blade root laminate, and a pre-stressed screw with one tip screwed to the nut and the other one connected to the rotor hub by means of standard washers.
One of the main characteristics of this kind of joints is that, due to the high pre-stresses introduced in the screw, there is always compression under the nut, no matter whether the external load is introduced by tension or compression. Therefore, the dangerous loosening of the joint, due to reversed loads, is avoided.

There are some important differences between T-bolt joints and classical overlapped joints, among others:

- The laminates are extremely thick: typically 55-80 mm.
- The stress distribution in the bearing zone is clearly non-uniform through the thickness.
- When loaded in tension, the pretension of the screw creates a load path different to those usually found in overlapping joints (a kind of tension + compression by-pass).
- Usually it is not possible to include any clamping pretension in the nut.

Those differences make unfeasible to use the results obtained from overlapped joints to predict the T-bolt joint strength. Despite this fact, there are only a few published works about the T-bolt connection [1]. Therefore, new efforts are required in order to characterize the T-bolt behavior and its strength.

**Finite element model**

In order to characterize the stress distribution in the T-bolt a parametric FE model has been developed. It allows an easy generation of 3D FE models, for any particular set of dimensions and material properties, by simply editing a text file containing the different physical parameters as well as the meshing densities.

The model has been implemented using the program MSC MARC.
The geometry presented does not correspond to a blade root, but to the test specimen shown in figure 2. This specimen has been used to verify the numerical results and will be used to investigate the failure modes of this kind of joints. The specimen consists of a laminate 37 mm thick, joined to two steel blocks by means of two equal T-bolts.

![Test coupon diagram](image)

**Fig. 2 Test coupon**

The laminate in the vicinity of the joint has been modeled as an homogeneous orthotropic material with elastic properties equivalent to those of the laminate. Although using this assumption interlaminar stresses are neglected, it is not possible to use a “one or more” element per layer approach, due to the enormous number of layers involved: about 280 for a laminate 60 mm thick.

Due to symmetry considerations, only 1/8 of the laminate part of the coupon has been modeled. The model uses eight node hexahedral elements with bilinear shape functions (element 21 from the MARC element library). This element class was chosen instead of more accurate quadratic elements, because of the problems arising from the use of them in contact simulations.

The presence of the screw hole makes not possible to mesh the T-bolt geometry using simple radial structured meshes similar to those used for double lap bolted joints (figure 3a). Thus, automatic unstructured mesh generators were tested, but it was not possible to have a precise control over the mesh quality in the different areas of the model.

![Mesh diagrams](image)

**Fig. 3 a) Radial structured mesh. b) T-bolt.**
Therefore, an automatic structured meshing procedure for the T-bolt geometry has been developed. The mesh density in the different areas of the model is controlled by several simple variables that allow to obtain meshes with high element densities only where high stress gradients are expected (figure 3b). Using this procedure, it is possible to obtain accurate results with less than 1/5 of the nodes needed in unstructured meshes.

The nut has been modeled as a rigid cylindrical surface because of the small influence of considering its stiffness on the results (the stiffness ratio between steel and the considered laminate is about 210 to 18). Therefore, the contact procedure checks whether each node in the nut hole surface penetrates the cylindrical surface. Once contact is detected, the degrees of freedom are transformed to a local system, and a constraint is imposed such that: \( \Delta u_{\text{normal}} = v \cdot n \), where \( v \) is the velocity of the rigid surface and \( n \) is the normal to it. The friction forces between the nut and the laminate have not been considered.

Due to the pretension of the screw, the increment \( (P_b) \) of the screw tension \( (F_b) \) is only a fraction of the applied external load \( (P) \). This fraction depends on the geometry of the joint and on the stiffness ratio between the screw and the part of the laminate between the nut and the steel block. Therefore, the relative displacement between the nut and the steel block should be calculated for each load case, taking into account the applied external load \( (P) \), the initial pretension of the screw \( (F_0) \), the screw stiffness \( (K_b) \) and the laminate stiffness \( (K_m) \). Figure 4 shows the relations between the different parameters in the so-called rigging diagram.

\[
F_b = P + F_m \\
F_b = F_0 + P_b \\
F_0 - P + F_m \\
F_0 - F_0 + P_b
\]

**VERIFICATION**

In order to verify the F.E. model, a single specimen has been tested both numerically and experimentally. The geometry of the test specimen is shown in figure 2 and its dimensions are shown in the following table.

<table>
<thead>
<tr>
<th></th>
<th>D (mm)</th>
<th>d (mm)</th>
<th>w (mm)</th>
<th>t (mm)</th>
<th>L (mm)</th>
<th>e (mm)</th>
<th>tf (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sp1</td>
<td>25</td>
<td>17.5</td>
<td>100</td>
<td>37 (avg.)</td>
<td>300</td>
<td>60</td>
<td>60</td>
</tr>
</tbody>
</table>

Regarding to the material, the specimen was made of a glass fiber - epoxy laminate \((0,+45,-45,90)_{42}\), with a fiber volume fraction \( V_f \approx 35\% \). The elastic properties of the laminate have been estimated assuming that all layers in a small laminate region are under the same in–plane strains \((\varepsilon_{xx}, \varepsilon_{yy} \text{ and } \varepsilon_{xy})\) and also the same out–of–plane stresses \((\sigma_{zz}, \sigma_{yz} \text{ and } \sigma_{xz})\). The
resulting values are shown in the next table:

<table>
<thead>
<tr>
<th>$E_x$  (MPa)</th>
<th>$E_y$  (MPa)</th>
<th>$E_z$  (MPa)</th>
<th>$v_{xy}$ (-)</th>
<th>$v_{yx}$ (-)</th>
<th>$v_{zx}$ (-)</th>
<th>$G_{xy}$ (MPa)</th>
<th>$G_{yz}$ (MPa)</th>
<th>$G_{zx}$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>18130</td>
<td>10315</td>
<td>7200</td>
<td>0.346</td>
<td>0.385</td>
<td>0.114</td>
<td>4776</td>
<td>2398</td>
<td>2658</td>
</tr>
</tbody>
</table>

The strains have been checked both at the laminate surface, at a point placed 10 mm under the bearing area of the nut (points G1 and G2 in figure 5) and inside the screw hole at a distance of 18 mm from the nut (point Op1). The points on the laminate surface have been checked using electrical strain gauges, whereas, at point Op1, an optical strain sensor based on Bragg gratings has been used. A description of the optical measurement technology can be found in [2].

Figure 5 Measurement points

Figure 6 shows the results obtained experimentally, without screw pretensioning, compared to the calculated with the F.E. model. As the F.E. model is symmetric, points G1 and G2 have the same numerical strain, that has been compared with the mean of the measures of both gauges. There is an error of about 15% between experimental and calculated values. This error may be due to some delaminations found in the edge of the nut hole, developed during manufacturing. These delaminations may lead to a relaxation in the stresses under the nut. Moreover, the estimated elastic properties of the laminate have not been verified experimentally.

Despite this error level, a good correlation can be observed between the results obtained in the different points.

Figure 6 Experiment – FEM comparison
SIMPLE STRESS ANALYSIS

In the following paragraphs, some results are presented and discussed for a T-bolt geometry identical to the used for verification, except on the width (w), that has been taken as 50 mm. This set of values leads to the geometry ratios (w/D, d/D, e/D…) typical of current blade root designs.

For double lap bolted joints, when there is no clearance between the bolt and the laminate, the stress distribution is not dependent on the applied load [3]. But, according to figure 4, due to the pretension of the screw (F₀), the bearing load (F₀) will not equal the external load (P) until the opening of the joint (P ≥ P₀). In fact, for a given initial pretension (F₀), the ratio F₀/P will continuously vary from ∞ to 1, as P increases from 0 to Pₖ. Thus, although the stress distribution might not be dependent on the applied load (P), it will always depend on the F₀/P ratio. This fact is illustrated in figure 7, where the x component of strain in the middle sections (through planes x-y and z-x) of the laminate are presented for a case with initial pretension F₀=35 kN. The external load (P) is 19 kN in figure 7a, and 65kN in figure 7b, corresponding the first one to an F₀/P ratio of 2, and the second one to a wide open joint (F₀/P=1). While in the first case, the maximum compressive strain almost doubles the tensile one, in the second, maximum compressive and tensile strains are almost balanced.

Therefore, it was necessary to define which F₀/P ratios should be studied. Obviously, the most interesting ratios will be those at which failure is more likely to occur. Thus, for a static analysis, the load at which opening of the joint occurs (F₀/P=1) has been chosen, due to the following facts:

1. In the current T-bolt design the screw acts as a “fuse”, because it is the element of the joint easiest to replace.
2. If the former design criterion is used, and the fracture of the screw occurs before the opening of the joint, the maximum external load supported would be less than the maximum allowed by the screw.
3. Another important design criterion states that, in the case of laminate failure, it is always preferable a bearing failure to a net tension failure, because of the catastrophic nature of the latter. The situation where the net tension failure is more likely to appear is

![Fig. 7 a) Strains, x comp., F₀=35 kN, P=19 kN  b) Strains, x comp., F₀=35 kN, P=65 kN](image-url)
at joint opening. Therefore, if the joint is designed in order to force bearing failure at joint opening, net tension fracture would not occur in any other situation.

Despite its more sophisticated geometry, it is not difficult to compare the T-bolt joint to a typical double lap joint. Therefore, our first analysis consists in a comparison between the strain and stress fields of both joints in a similar load situation, with the aim of take advantage of the knowledge already available on double lap joints [4].

Figures 8 to 13 show the results for a load of 65 kN without any pretension. Both models (T-bolt and double lap) have the same geometry, except, of course, for the presence of the screw hole in the case of the T-bolt.

The comparison of in–plane behaviors is based on strains ($\varepsilon_{xx}$, $\varepsilon_{yy}$ and $\varepsilon_{xy}$), while the out–of–plane comparison it is based on stresses ($\sigma_{zz}$, $\sigma_{yz}$ and $\sigma_{zx}$). If the basic set of four layers (0,45,-45,90) is thin enough, we can assume that the laminate behaves like an homogeneous orthotropic material. Therefore, the in-plane strains will be the same for all layers, while the stresses will vary from one to another. Conversely, the out–of–plane common values will be the stresses.

![Fig. 8](image1.png) a) Strain, x direction, T-bolt.  b) Strain, x direction, double lap joint

![Fig. 9](image2.png) a) Strain, y direction, T-bolt.  b) Strain, y direction, double lap joint
Fig. 10  a) Stress, z direction, T-bolt.   b) Stress, z direction, double lap joint

Fig. 11  a) Shear strain, xy direction, T-bolt.   b) Shear strain, xy direction, double lap joint

Fig. 12  a) Shear strain, yz direction, T-bolt.   b) Shear strain, yz direction, double lap joint
For all of the stress and strain components there is a high stress concentration at the intersection between the screw hole and the nut hole. These very local extreme stresses have no influence on the joint strength. Therefore, in the following discussion, we will always discard the results in this area.

In general, the main stress and strain concentration areas (apart from the above mentioned) are the same for both joints. Therefore, three main stress zones should be analyzed: the net tension zone (figure 8), the bearing zone (figures 8 and 9) and the shear–out zone (figure 11).

Comparing both models, the following conclusions can be extracted:

- In the net tension zone, stresses and strains are approximately the same in both designs (figures 8 and 9).
- The compressive maximum strain in the T-bolt is about 1.75 times the maximum for double lap joints (figure 8). Part of this increment can be explained by the bearing area reduction caused by the screw hole ($A_{\text{bolt}} = A_{\text{overl.}} / 1.35$).
- In the bearing zone, the tensile stress ($y$ direction) is almost 2 times higher in the T-bolt (figure 9). This strain, although is not directly responsible for the bearing failure, reduces the laminate strength, producing matrix cracks and fiber failure in the 90º layer.
- The shear-out zone is affected in two ways by the geometry of the T-bolt: first, there is a small increase in the maximum $xy$ shear strain and, second, there is a shifting in the position where the maximum shear strain is observed (figure 11). In the T-bolt the maximum shear strain is shifted towards the bearing area. The ratio between maximum shear strains is about 1.4, and corresponds mainly to the reduction of the shear section due to the screw hole (about a factor of 1.5).
- There is no significative increasing in the maximum values of the stresses in the $z$ direction compared to double lap joints (figure 10).
- $yz$ and $zx$ shear stresses appear in the neighborhood of the screw hole (figures 12 and 13).
CONCLUSIONS

A three dimensional FE model has been developed for the prediction of the stresses and strains at a T-bolt joint. The model has been parameterized in order to allow an easy modification of the different geometric and elastic parameters.

A simple experimental verification of the model has been conducted. Although the absolute values in the bearing zone do not completely agree with the calculated ones (15% of error), good correlation was found for the variations between different measurement points.

Although for each ratio $F_b/P$ there is a different stress distribution over the laminate, only the case with $F_b/P = 1$ has been analyzed, because it is the most critical for the strength of the joint.

Comparing the T-bolt stress distribution with that of a double lap bolted joint, it is found that the net tension stresses are quite similar in both designs, while the stresses in the bearing zone are between 1.75 and 2 times higher in the T-bolt. Those increasing factors can not be explained entirely by the bearing surface reduction caused by the screw hole. Due to the higher bearing stresses, a lower joint efficiency is to be expected. On the other hand, the highest possible efficiency would appear at the transition between net tension and bearing failure, and this will occur at a much lower w/D ratio than in the case of double lap joints. Finally, some out-of-plane shear stresses have been found in the T-bolt model that are not present in double lap joints. These stresses may contribute to the initiation of a delamination process, but it is not possible, with the current model, to determine the relevance of these stresses compared to the interlaminar stresses in the same area.

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REFERENCES