A practical approach for the peel stress prediction in the trailing-edge adhesive joint of wind turbine blades

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Abstract. Wind turbine blades consist of thin-walled cylindric and airfoil-shaped structures, which are prone to “breathing” or “pumping” when subjected to cyclic loading. The “pumping” induces a peel stress in the adhesive layer of the trailing-edge bond line. To take account of this peel stress in the design phase, adequate models are required. State-of-the-art blade finite element (FE) models are usually implemented using shell elements. The trailing-edge joint is often represented by solid elements that are connected with the shell elements. The peel stress peak of interest occurs at the edge of the adhesive joint, which is, subject to a singularity, however. This study proposes a practical approach to estimate the peel stress peak in the adhesive joint with the help of the analytical solution which approximates and extrapolates the FE results. Moreover, different modeling techniques are benchmarked in respect of the peel stress prediction.

1. Introduction
The trailing-edge joint in wind turbine blades is subjected to a fatigue loading which results into a multiaxial stress state in the adhesive layer. The magnitude of each stress component depends on the position along the blade length. Cyclic peel stress amplitudes occur especially in the inboard blade region around maximum chord length. Peel stresses are caused by the “pumping” effect (Fig. 1a). [1]

Although the presence of peel stresses can provoke several failure modes of the joint [2], e.g., an adhesive failure or a fiber-tear failure, the present research investigates the cohesive failure in the adhesive layer, called a tunneling crack [1, 3].

The stress exposure, as described by Puck [4], in combination with stress-life curves, e.g., according to Basquin [5], can be used to estimate the load cycles in respect of tunneling crack

Figure 1. “Pumping” deformation of the blade’s trailing-edge panels (a) and peel stress $\sigma_t$ along the width of the adhesive layer (b).
initiation as a consequence of multiaxial fatigue loading, as shown by Krimmer et al. [6]. To this end, all stress components are calculated by means of finite element (FE) simulations to obtain the equivalent stress according to [7], for example.

Rosemeier et al. [1] used a state-of-the-art FE implementation of the adhesive layer with the aid of solid elements which share nodes with the blade shell elements (Fig. 4d and Fig. 2a). Haselbach [8] compared other modeling techniques of the trailing-edge bond line, e.g., a multi-point constraint (MPC) connecting the shell and the solid element (Fig. 4e).

The peel stress peak occurs at the edge of the adhesive represented by solid elements. This edge is subject to a singularity that typically occurs at a geometric or a stiffness discontinuity. That means stress values tend toward infinity when the mesh density is increased. To overcome this singularity, the stress value can be extracted at a defined distance from the edge, c.f. [9]. This method, however, may underestimate the actual stress peak value.

Spies [10] was the first to present a theoretical analysis of the mechanics of elastic peeling. He considered a substrate strip, which is modeled as an elastic beam, embedded into an elastic adhesive layer, and loaded by a peel force which acts perpendicular to the adhesive layer plane, i.e., at an angle of 90°. A similar model was presented by Bikerman [11]. Where Spies’ and Bikerman’s models assume small deflection angles, which is valid for relatively stiff substrate strips, other models take into account large deflections of the strip, viscoelasticity of the adhesive, and peeling angles greater or less than 90°, c.f. [12–14].

Hinopoulus et al. [15] were able to capture the typical peel stress distribution in a relatively detailed solid FE model of a T-peel joint.

In this work, a practical approach is proposed to estimate the peel stress peak in the adhesive layer of an FE model with the help of the analytical solution. Moreover, different FE modeling techniques are investigated on a plate level to provide a thorough representation of the trailing-edge joint subjected to “pumping”. The analytical solution by Spies [10] serves as reference. We focus on suitable models that can be implemented into a state-of-the-art 3D FE blade model at a reasonable computational cost.

2. Methods

2.1. Analytical model

The suction side (SS) and the pressure side (PS) of the trailing-edge joint (Fig. 3a) are modeled using Spies’ [10] assumptions to calculate the peel stress within an adhesive layer that joints a substrate strip with a rigid wall, this peel stress being a consequence of a moment \( M \) and a shear force \( F \) acting at one strip edge (Fig. 3b):

(i) The strip is assumed to be straight.
(ii) The strip extends to infinity in the positive \( x \)-direction.
(iii) Normal stresses in the \( x \)-direction of the strip are neglected.
(iv) The adhesive is homogeneous and obeys Hooke’s law.
(v) The influence of the strip length \( b \) in the \( z \)-direction is neglected.
(vi) The adhesive layer is assumed to be fixed at the lower edge to an absolutely rigid wall.
(vii) Deformations in the $xt$-plane resulting from normal and shear loads in the strip are neglected.

\[ \sigma_t = \sigma_0^t \, e^{-\frac{x}{c}} \left( \cos \frac{x}{c} - \frac{M}{M + F_c} \sin \frac{x}{c} \right), \]
\[ \sigma_0^t = \frac{M + F_c}{b \, c^2}, \]

where $c = \sqrt{\frac{t_a B}{2E_a}}$. $B$ denotes the bending stiffness of the substrate in the transverse $s$-direction about the $z$-axis, which is equivalent to the matrix element $D_{22}$ of the bending stiffness matrix, as explained in [16]. $E$ denotes Young’s modulus and $t$ the thickness, and index $s$ stands for substrate strip and index $a$ for adhesive. In the case of isotropic substrates, (3) simplifies to:

\[ c = t_s \sqrt{\frac{1}{6} \frac{E_s}{t_s}}. \]

The positions of the extrema are calculated by setting the first derivative of (1) equal to zero and solving for $x$:

\[ x_n = 2c \left( n \pi + \arctan \left( \frac{1 - \frac{M}{M + F_c}}{1 + \frac{M}{M + F_c}} \right) \right), \]

where $n \in \mathbb{Z}$. $x_0$ defines the distance from the edge to the first extreme value, i.e., a minimum or maximum point, and $x_1$ defines the beginning of the flat plateau (Fig. 5a).

Figure 3. Cross-section view of the trailing-edge joint subjected to a peel force (a); diagrammatic representation of Spies’ model (b), based on [10].
2.2. Finite element plate models

Four two-dimensional sandwich-like plate models were implemented in ANSYS APDL (ANalysis SYStem Parametric Design Language) [17] to generically represent the trailing-edge joint:

- A high fidelity “n-sliced solid” model wherein the nth layer of the bond line laminate is represented by a homogeneous SOLID186 element (Fig. 4b).
- A “3-sliced layered solid” model wherein the adhesive layer is represented by a homogeneous SOLID186 element covered on both the PS and the SS with a layered SOLID186 element, which is attributed with the properties of the respective bordering laminae via the SECDATA command (Fig. 4c).
- A “solid/shell share” model wherein a homogeneous SOLID186 element represents an adhesive layer which is as thick as the whole laminate, and two SHELL281 elements represent the bordering laminae (Fig. 4d). The shell elements share their nodes with the solid. The shell nodes are offset with respect to the outer surface in such a way that the laminae overlap the solid.
- A “solid/shell MPC” model wherein a homogeneous SOLID186 element represents the adhesive layer, and two SHELL281 elements represent the bordering laminae (Fig. 4e). The shell nodes are offset with respect to the outer surface in such a way that the laminae protrude into the gap between the solid and shell. Multi-point constraints (MPC) bond the surfaces of the solids and shell with each other.

The SS edge along the z-direction is constrained in all translational degrees of freedom. A rigid body element of type CERIG connects the PS edge with a master node, which can translate in the t-direction. A unit load $F$ is applied to the master node.

The bordering laminae on the SS and PS sides overlap the adhesive layer over the length $l = 1/3a = 0.33$ m. The adhesive layer is meshed between the two sides over a width of $2/3a = 0.67$ m, a length of $b = 1.0$ m, and a mesh size from 16 to 64 elements along the width. The element size in the longitudinal z-direction is 100 mm.

![Figure 4](image_url)
2.3. Mesh density and peak stress calculation
The mesh density along the width of the adhesive layer increases toward the inner edge where the peel stress peak of interest is expected. The required distance $x_0$ from the edge to reproduce the characteristic peel stress distribution is estimated with the help of the analytical solution (5).

The peel stress distribution is derived from nodal FE stresses along the SS and PS edge along the width ($x$-direction) of the adhesive layer. By varying $M$, $F$, and $b$, the analytical equation (1) approximates the peel stress distribution of the FE model using the least-squares method [18]. The first node at the inner edge is neglected since it is subject to a singularity: The higher the mesh density, the more the first node tends toward infinity. Finally, the peel stress peak of the approximated solution obtained with (2) is used as the result.

In particular, for the approximation of the “solid/shell share” FE results, the adhesive thickness $t_a$ which is input for $c$ according to (4) is corrected in order to take into account the decrease in stiffness of the elastic embedding, which results from the overlap of the adhesive and the substrate:

$$t_{a,cor} = \frac{1}{2}t_a + t_s$$  \hspace{1cm} (6)

After approximating the “solid/shell share” FE results with the corrected thickness according to (6), the peel stress peak $\sigma_0^t$ obtained with (2) is again corrected in order to obtain comparable results to those of the analytical solution. This correction is derived by relating the stress peak $\sigma_{0,cor}^t$ according to (2), which is obtained by taking the corrected thickness $t_{a,cor}$ according to (6) into account, to the stress peak $\sigma_0^t$ obtained by taking the actual thickness $t_a$ into account:

$$\sigma_{t,cor}^0 = \sigma_0^t \sqrt{\frac{t_a + 2t_s}{t_a}}$$  \hspace{1cm} (7)

3. Results
3.1. Sensitivity to input parameters
A simple reference steel-adhesive joint is subjected to a sensitivity study using the analytical model. To this end, the input parameters of the reference (Fig. 5a) are varied. Their impact on the peel stress peak and the edge-distance to the extreme value $x_0$ is investigated. The decrease in stiffness of the substrate $E_s$ toward typical values of uni-directional and bi-directional glass fiber-reinforced polymers (GFRPs) increases the peel stress peak and shifts $x_0$ closer to the edge (Fig. 5b). The same effect is observed when either the substrate thickness $t_s$ or the adhesive thickness $t_a$ is decreased (Fig. 5c and Fig. 5d). The opposite effect is observed when the thickness increases. An increased peel moment $M/b$ increases only the stress peak but has no effect on $x_0$ (Fig. 5e and Fig. 5f). The peel force $F/b$ increases by a factor of ten, slightly increases the peak, and slightly shifts $x_0$ away from the edge.

3.2. Effect of modeling technique
Three FE plate modeling techniques are investigated in respect of the prediction of the peel stress peak and are compared to the analytical model (Fig. 6a). The reference steel-adhesive joint is implemented with a mesh density of 64 elements along the adhesive layer width.

The approximation of the “3-sliced layered solid” model (indicated as “fit” in Fig. 6a) is closest to the analytical peak stress ($-10.4\%$), followed by the “solid/shell MPC” model ($-11.2\%$). The fit of the “solid/shell share” model exhibits the lowest peak ($-22.3\%$). When the peak stress of the approximated stress distribution is corrected according to (7), the “solid/shell share” model reveals the peak closest to the analytical solution, which is overestimated by 4.2%.

The first node in all FE models overestimates the analytical solution. Discontinuities are observed in the nodal FE values of the “solid/shell MPC” model at $x \approx 2.5$ mm and $x \approx 5.5$ mm.
Figure 5. Peel stress distribution along the adhesive layer (a); sensitivity to stiffness fraction $E_s/E_a$ (b), substrate thickness $t_s$ (c), adhesive thickness $t_a$ (d), peel moment $M/b$ (e), and peel force $F/b$ (f).
3.3. Effect of mesh density
The element discretization of the “solid/shell share” model is varied between $N_e = 16$ and $N_e = 64$ elements along the adhesive layer width, i.e., in the $x$-direction (Fig. 6b). An element spacing ratio of $2N_e/1$ between the element width at the edge ($x = 0$ mm) and the center ($x = 333$ mm) is set up such that the mesh density increases toward the edge.

At least three nodal results within the distance $2x_0$ from the edge, including the first node, are required to capture a stress peak that is close to the analytical solution using the method of approximation and extrapolation.

The higher the mesh density, the better the match between the analytical peel stress distribution and the modeled one. This trend is confirmed when the peak stress is corrected. The peak stress obtained from the first node, however, increases with the mesh density toward infinity. This is attributed to a singularity.

![Figure 6. Peel stress distribution along the adhesive layer width; comparison of modeling techniques (a); mesh density study for “solid/shell share” model (b).](image)

4. Discussion
The approach proposed in this paper circumvents the dilemma that the position of the peel stress peak value of interest coincides with a singularity in an FE model: An analytical model approximates the nodal stress results and extrapolates the stress peak of interest. Since the force and moment acting at the substrate edge cannot be extracted directly from a full 3D blade model with the modeling techniques presented in this work, the approach varies $M/b$ to fit the stress peak and the amplitude of the valley (Fig. 5e), and $F/b$ to fit the distance of the valley from the edge $x_0$ (Fig. 5f).

When the peel stress peak in the “solid/shell share” model-fit is corrected, this peak is closest to the analytical solution when compared to the other models. Moreover, this modeling technique is implemented with the least effort in a state-of-the-art full 3D FE shell model. Therefore, this model seems suitable for the design analysis of a trailing-edge joint of a wind turbine blade.

A sufficient number of elements need to be placed between the edge and the materially and geometrically dependent parameter $x_0$, which is estimated using the analytical solution (5).

The proposed approach represents the more conservative option to predict the stress peak in comparison to the method of extracting the stress value at a defined distance from the edge, as used in [9]. Moreover, this approach is computationally more efficient since the mesh density
can be determined with the analytical model by calculating the distance of the extrema from the edge using (5).

The assumption made in [1] that the peel stress distribution and peak may be affected by the connection between the shell and solid elements, i.e., that the rotational degrees of freedom of the shell elements are not adequately transferred to the solid elements, is disproved in this work. All modeling techniques benchmarked in this work are able to capture the stress response of the analytical solution with the proposed approach.

5. Conclusions
In this work, we propose and verify a practical approach to estimate the peel stress peak at the inner edge of the trailing-edge joint with the help of an analytical model which approximates and extrapolates nodal peel stress results. The approach circumvents the dilemma that the position of the peel stress peak value of interest coincides with a singularity in an FE model. The computational efficiency of this approach can be optimized by calculating the distance of the extrema analytically, and thus the region required to have a higher mesh density. Moreover, it is suggested that the “solid/shell share” modeling technique can be implemented with the least effort in a state-of-the-art full 3D FE shell model when compared to the other techniques presented.

In a future work, the analytical model can be extended to consider asymmetric adhesive joints. Furthermore, this approach can be applied to other loading situations of the adhesive joint such as normal or shear stressing.

Supplemental material
The data used for the figures are available at https://doi.org/10.5281/zenodo.3937258 [19].

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