# STRESS AND FAILURE ANALYSIS OF ADHESIVELY BONDED TUBULAR JOINTS SUBJECTED TO BENDING LOADING

Biabangard Oskouyi, A, Stack, M, and Mertiny, P*
Department of Mechanical Engineering, University of Alberta
Edmonton, Alberta, Canada, T6G 1H9

* Corresponding author: pmertiny@ualberta.ca

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## ABSTRACT

Joining of components made from composite materials is one of the most challenging tasks in engineering design of composite structures. Material and geometry discontinuities introduced by joints can be a location for stress concentrations and stress singularities, making the structure susceptible to static and fatigue failure. Consequently, joints play a dominant role in the overall integrity of composite structures. In this study, the load capacity and failure of tubular adhesively bonded joints subjected to bending and torsional loading are investigated. A three-dimensional non-linear finite element model was developed to examine the effects of design parameters on the stress distribution within the joint. Design parameters include the adhesive thickness and adherend overlap length. This study was a key step toward the design of adhesively bonded joints for drive shafts subjected to substantial bending loads.

## 1 INTRODUCTION

The engineering design of joints in fiber reinforced polymer composite (FRPC) structures is a critical and challenging task. Joints introduce material and geometry discontinuities which can be the location for stress concentrations making the structure susceptible to static or fatigue failure. Therefore, the overall integrity of the composite structure is often governed by the performance of joints and not by the adherends [1]. Elevated stress fields and failure of joints have been addressed in several research works and technical reports. Mechanical fastening and adhesive bonding are the two main joining methods that are widely employed in the composite industry. Weakening the structure by drilling apertures and increasing the risk of stress concentration and fatigue failure are the main drawbacks associated with mechanical fastening [2]. Furthermore, mechanical fastening may cause laminated structures to be susceptible to delamination, especially during machining processes. Mechanical fasteners may also initiate galvanic corrosion (e.g. carbon fibers and steel fasteners). Adhesively bonded joints, on the other hand, often exhibit excellent fatigue life and structural performance and are of low cost, which makes them preferable to mechanically fastened joints. In principle, adhesive bonding provides for smooth a stress distribution, and the structural efficiency and integrity of adhesive bonded joints are superior compared to mechanically fastened joints [1]. Tubular adhesively bonded joints are frequently employed by engineers in the design of fiber reinforced composite drive shafts or as a coupling for piping, and as such, tubular adhesively bonded joints are present in wide range of applications, including in the automotive, oil and gas, and aerospace industry. The majority of research contributions on adhesively bonded joints in the technical literature has focused on flat components, and comparatively few works are available that address adhesive bonding for the assembly tubular composite structures. Kim et al. used a single lap tubular adhesive joint to connect a one-piece hybrid carbon/glass/epoxy composite shaft to aluminum yokes. They developed a finite element model and conducted experiments to investigate the static torque transmission capacity of the tubular adhesively bonded joints. They showed that the torque transmission capacity of shafts increases as the bonding length is increased. However, with exceeding bonding length a diminishing effect on load transmission capability of the shaft was observed [3]. Kim et al. further investigated the effect of governing parameters such as adherend surface roughness and adhesive thickness on fatigue characteristics of tubular adhesively bonded joints. They reported 2μm and 0.15mm as optimal values of surface roughness and adhesives thickness, respectively. They also demonstrated that double lap joints have a 20% higher torque capacity. Surface treating double lap joints and providing double lap joints with scarf edges also increased the static torque capacity by a similar amount [4]. Hipol developed a finite element model to investigate the stress distribution in tubular lap joint of a steel tube adhesively bonded to a FRPC shaft subjected to torsion. It was shown that the maximum shear stress in the adhesive layer is governed by the stiffness ratio of the adherends, and that the peak shear stress can be lessened by reducing the stiffness imbalance. An interesting finding in Hipol's study is that the maximum shear stress in the adhesive layer is not affected by the bonding length [5].

To the authors' knowledge, research works available in the technical literature conducted on the structural integrity of tubular adhesively bonded joints are restricted to studying the structural performance and stress distribution of joints subjected to pure torsion. The technical literature has been mute with regard to investigating the integrity of tubular adhesive bonded joints subjected to bending loads, probably since this loading scenario has only limited significance in terms of stress distribution and structural performance of drive shafts in practical engineering applications. In current research work a numerical model was developed to investigate the stress distribution and structural integrity of a FRPC shaft subjected to bending that is connected to a steel shaft using adhesive bonding. A three-dimensional finite element model was employed to explore the effect of governing parameters, such as adhesive thickness and overlap length on stress distribution and structural performance of tubular bonded joints. Stress distribution and stress concentration were comprehensively investigated, which play a significant role in understanding the characteristics of tubular joints subjected to bending loadings.

## 2 Finite Element Analysis

The finite element analysis, which focused on the composite adherend-adhesive interface, was conducted using ANSYS Mechanical APDL 17.1 [6]. Both the outer composite shaft and the adhesive were modeled in cylindrical coordinates using a bottom-up approach as depicted in Figure 1. This method facilitated using elements with a high quality mesh and a desired aspect ratio of unity (Figure 2). Both the composite shaft and the adhesive were modeled as isotropic materials. The adhesive was assigned a modulus of 10.0GPa and a Poisson's ratio ($\nu$) of 0.4. The bending stiffness of the FRPC adherend was equivalent to that of an E-glass epoxy shaft with an inner radius of 50mm and wall thickness of 20mm. Note that a comparatively large diameter shaft assembly with significant wall thickness was investigated in this study. The contact surface between the adhesive and composite shaft was modelled by producing two sets coincident nodes. One set belonged to the nodes corresponding to the composite shaft, the other to the adhesive. The coincident nodes were coupled to simulate the effect of the adherend-adhesive interface. In the analysis, three-dimensional, eight-node elements designated as SOLID185 were used for both the adherend and the adhesive. The adherend was modeled with a single layer of elements while the adhesive had two layers of elements with a length that was half of the adhesive thickness. To reduce computational cost, symmetric boundary conditions were applied. Considering the comparatively high stiffness of the inner steel component, displacement boundary conditions in all directions were applied on the innermost surface of the adhesive nodes as depicted in Figure 3, preventing the displacement of the adhesive surface in all directions. As such, this boundary condition simulated an effective bond between the inner adhesive surface and the steel component, and the outer composite shaft acted as a cantilever for a load applied at the opposite shaft end. This allowed for the effects of bending on an adhesive to be evaluated using the finite element analysis.

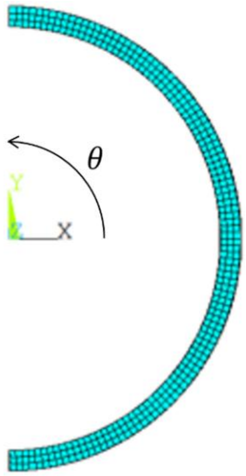


Figure 1. Coordinate system of FEM model.

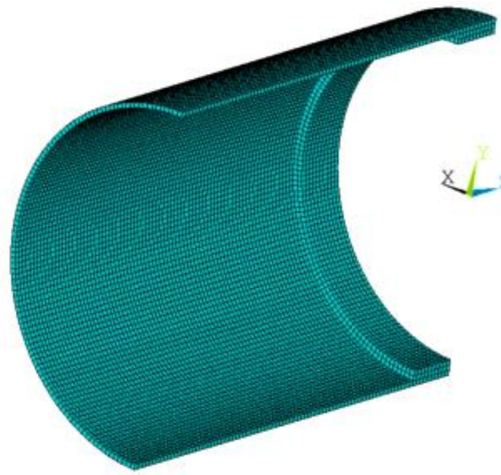

Figure 2. 3D view of model of adhesive and shaft.

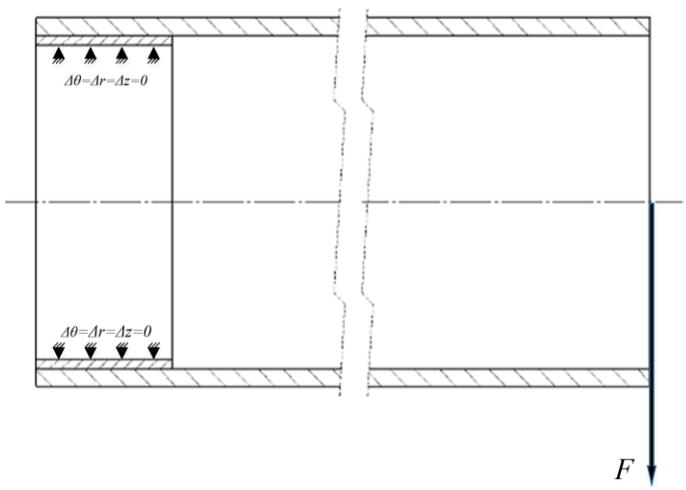


Figure 3. Section view of adhesive and adherend.

# 3 RESULTS AND DISCUSSION

Material and geometric discontinuities may initiate stress concentration and singularities in adhesively bonded joints, which are the main limitations in the application of adhesive bonding. Different design solution and parameters, such as overlap length, adhesive layer thickness, and adherend and adhesive stiffness dissimilarities, should be studied by engineers to mitigate stress concentration and singularities, leading to a more uniform stress distribution. In the following, the developed finite element model was employed to investigate the effects of tubular adhesively bonded joint design parameters on stress gradients and concentrations. Stress component in cylindrical coordinates ($\sigma_{rr}$, $\sigma_{\theta\theta}$ and $\sigma_{r\theta}$) were calculated and normalized to a nominal stress $(\sigma_{yy})_{avg}$ given by Equation (1).

$$(\sigma_{yy})_{\text{avg}} = M(2L_{\text{a}}^2 R_{\text{in}})^{-1} \tag{1}$$

where $M$, $L_{\text{a}}$ and $R_{\text{in}}$ are the bending moment applied to the joint, the overlap length or adhesive length, and the inner radius of the adherend, respectively. In the following reporting of analysis results, stresses in the midplane of the adhesive are reported.

### *3.1 Effect of Overlap Length*

Normalized stress values were calculated for tubular adhesively bonded joints with overlap lengths of 3mm, 5mm, 10mm and 15mm. The adhesive thickness, $t_{\text{a}}$, and adherend inner radius were fixed at 0.5mm and 25mm, respectively. $\sigma_{rr}$, $\sigma_{\theta\theta}$ and $\sigma_{r\theta}$ at the adhesive midplane and for a cylindrical coordinate $\theta$ of 90° (i.e. the location with the further distance from bending midplane) were plotted versus the normalized distance measured from the free edge of the outer adherend, as shown in Figures 4 to 6, respectively. For details on the employed coordinate system, see Figures 1 and 2. The graphs indicate that tubular adhesively bonded joints with longer overlap length are more susceptible to stress concentrations and stress singularities. For example, when $L_{\text{a}}/t_{\text{a}} = 30$ the stress gradient is significantly higher at the free edge of the adhesively bonded joint as compared to lower $L_{\text{a}}/t_{\text{a}}$ ratios. Consequently, an engineering designer needs to pay attention when deciding on the overlap length of adhesively bonded joints subjected to bending loads. Adhesively bonded joints with greater overlap length may be beneficial in terms of decreasing the applied average stress but they are also susceptibility to stress risers and consequently failure. Stress data for a cylindrical coordinate $\theta$ of 0° (i.e. at the neutral plane of the shaft in bending) are given in Figure 7. As expected, stresses at the adhesive midplane are negligible in this case. It is thus confirmed that the tubular joint experiences stresses that are most critical to structural integrity at the furthest distance from the bending midplane.

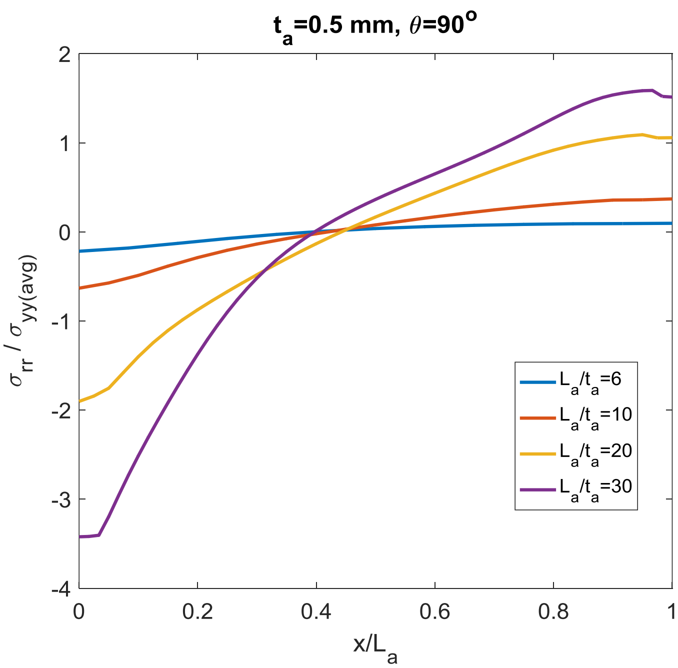


Figure 4. Effect of overlap length on radial stress versus normalized distance from the free edge of the outer adherend.

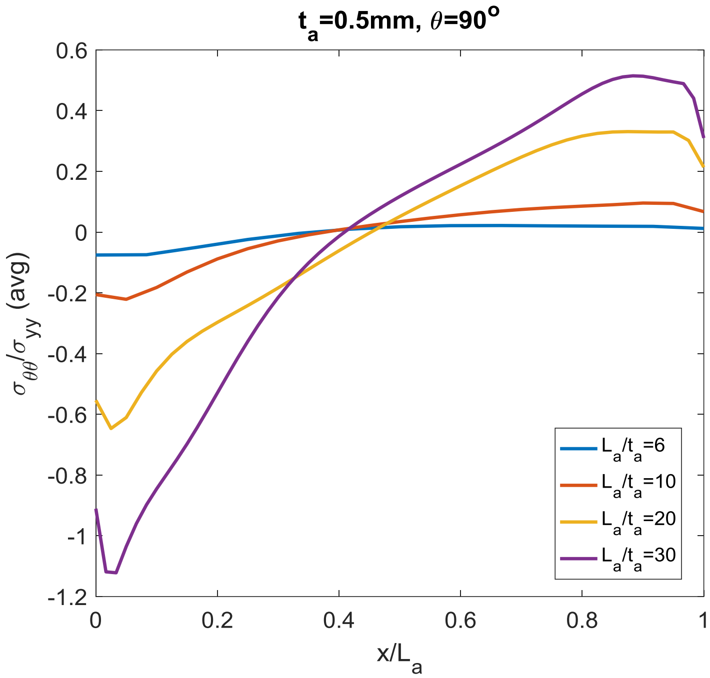


Figure 5. Effect of overlap length on circumferential stress versus normalized distance from the free edge of the outer adherend.

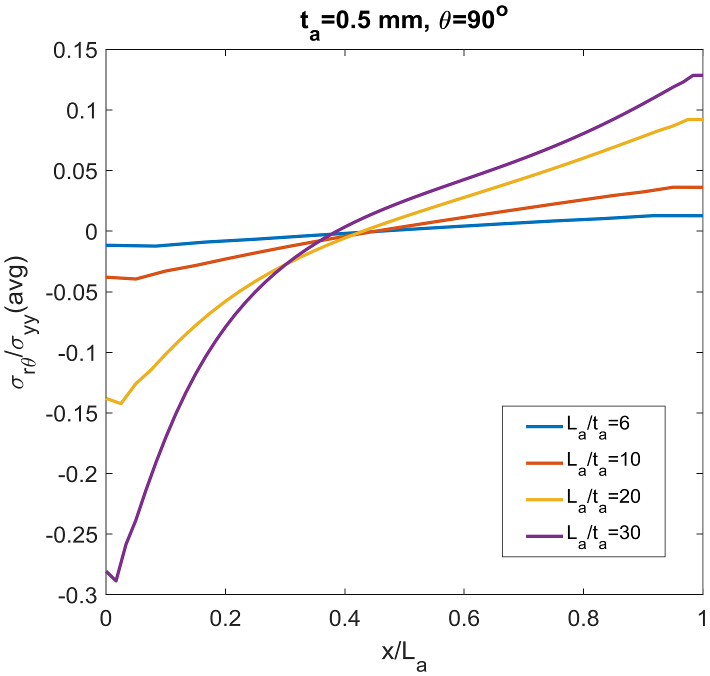


Figure 6. Effect of overlap length on shear stress versus normalized distance from the free edge of the outer adherend.

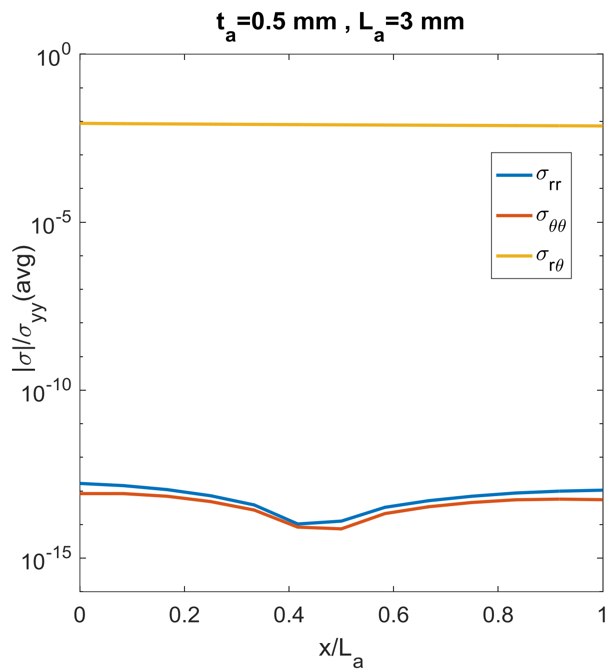


Figure 7. Stress data for $\theta$ = 0 versus normalized distance from the free edge of the outer adherend.

### *3.2 Effect of Adhesive Thickness*

Similar to the approach undertaken to investigate the effect of overlap length on the stress distribution of the tubular adhesively bonded joint, the developed finite element model was employed to investigate the effect of adhesive thickness. Stress components in cylindrical coordinates were evaluated for four different values of thicknesses, i.e. 0.5mm, 0.7mm, 1.0mm and 1.5mm. Results are depicted in Figures 8 to 10. The results obtained from the analyses indicate that the effects of adhesive thickness on the stress distribution follow a trend similar to that of overlap length, that is, lower $L_a/t_a$ ratios mitigate stress risers. Consequently, it can be inferred from the analyses that a joint design benefits from a larger adhesive layer thickness in conjunction with a shorter overlap length. As such, the $L_a/t_a$ ratio as a governing parameter plays an important role in the mitigation of stress concentrations in tubular adhesively bonded joints subjected to bending loadings.

## 4 Conclusions

In the presented research work the stress distribution in the tubular adhesively bonded joints subjected to bending was studied. The effects of the design parameters adherend overlap length and adhesive thickness were investigated employing a three-dimensional finite element model approach. The model simulated the stress distribution of a composite shaft bonded to a quasi-rigid steel component. The analysis results indicate that a low ratio of adherend overlap length to adhesive thickness is beneficial for reducing stress risers.

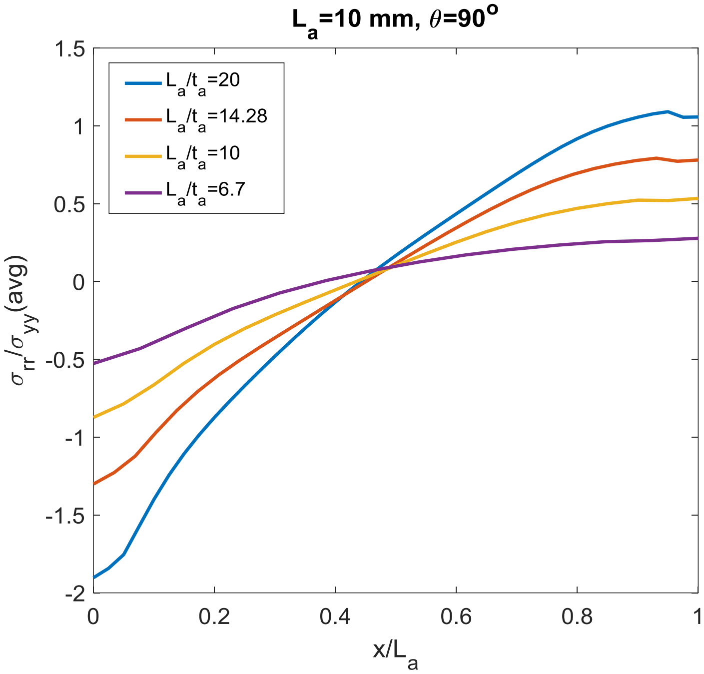


Figure 8. Effect of adhesive thickness on radial stress versus normalized distance from the free edge of the outer adherend.

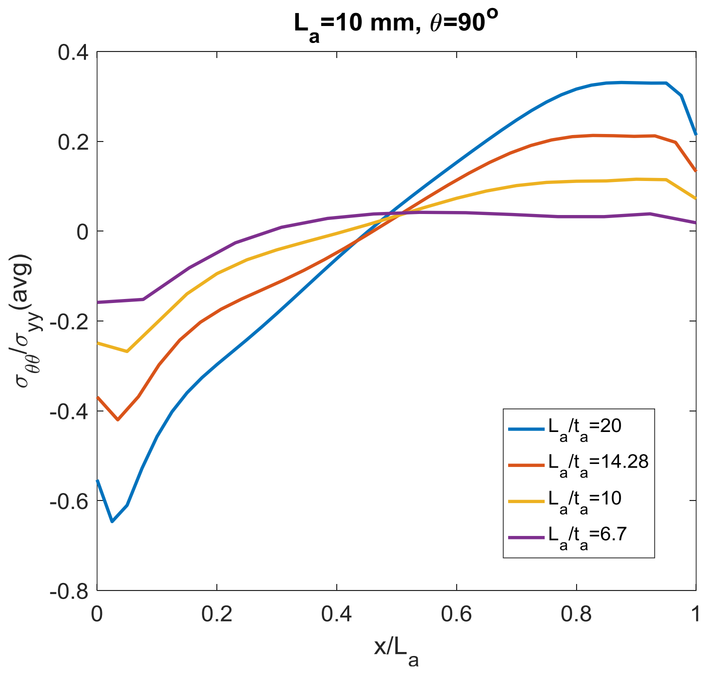


Figure 9 Effect of adhesive thickness on circumferential stress versus normalized distance from the free edge of the outer adherend.

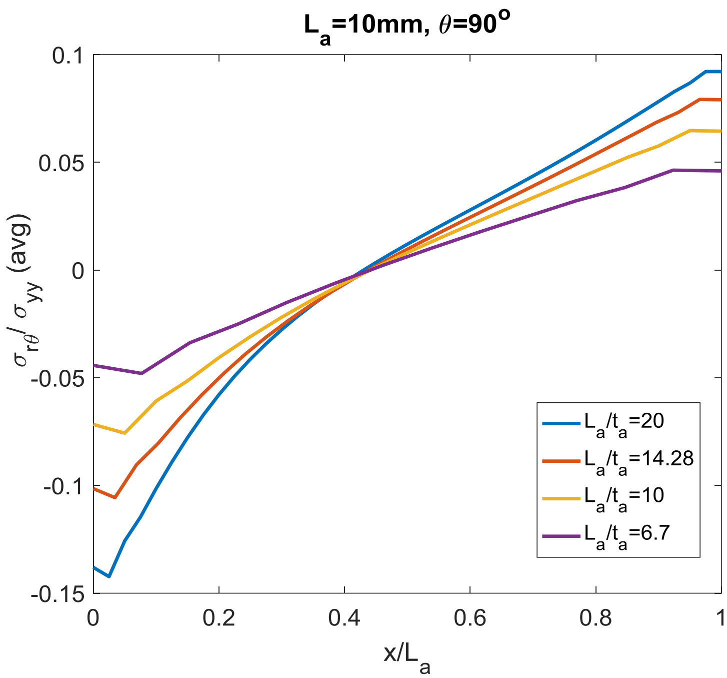


Figure 10. Effect of adhesive thickness on shear stress versus normalized distance from the free edge of the outer adherend.

# 5 REFERENCES


[1] Handbook-MIL-HDBK, Military. "17-3F: Composite Materials Handbook, Volume 3-Polymer Matrix Composites Materials Usage, Design, and Analysis." US Department of Defense (2002).

[2] C. Cao and D. Dancila, "Damage and failure analysis of co-cured fiber-reinforced composite joints with unidirectional lay-up", *Proceedings of the 44th AIAA/ASME/ASCE/AHS/ASC Structures*, pp. 7-10, 2003.

[3] J.K. Kim and D.H. Cho, "Investigation of adhesively bonded joints for composite propeller shafts", *Journal of Composite Materials*, vol. 35, no. 11, pp. 999-1021, 2001.

[4] K.S. Kim, W.T. Kim and E.J. Jun, "Optimal tubular adhesive-bonded lap joint of the carbon fiber epoxy composite shaft", *Composite Structures*, vol. 21, no. 3, pp. 163-176, 1992.

[5] P.J. Hipol, "Analysis and optimization of a tubular lap joint subjected to torsion", *Journal of Composite Materials*, vol. 18, no. 4, pp. 298-311, 1984.

[6] ANSYS® Academic Research, Release 17.1.