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EFFECT OF ADHEREND SHAPE ON STRESS CONCENTRATION REDUCTION OF ADHESIVELY BONDED SINGLE LAP JOINT

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Abstract:

In this work, the effect of adherend shape on the tensile strength of adhesively bonded single lap aluminum structures joint was numerically studied using three dimensional finite element models. Six joint models were investigated. In this paper, a static finite element analysis was performed in ANSYS considering geometric nonlinearities. The results show that the adherend geometry has the highest effect on peel and shear stresses. Similarly, for rounded and/or tapered geometries, adhesive material properties also cause a higher percent reduction in stress concentration.

1 Introduction

In recent years, the use of adhesively bonded joints has increased rapidly in industry application especially maritime, automotive and aerospace applications [1-3].

Adhesive joints have a lot of advantages with their low stress concentration, effective weight reduction [4], high fatigue resistance and low cost. Moreover, they typically provide higher structure integrity than other conventional joints and can be applied to connect dissimilar materials [5]. In addition, adhesively bonded joints have uniform stress and load distribution as well as better fatigue performance compared to conventional methods such as bolted and riveted joints [1, 3]. They enable structures with easy and simple joints [6].

Accurate failure predictions are required for efficient joint designs [3]. Over the years, single lap joints have been the most widely used adhesive joints and the subject of many research studies [1, 7, 8].

Erdogan [7] developed a mathematical model for the calculation of stresses in bonded overlapped joints in plates and tubes. Adams and Peppiatt [8] analyzed a bonded joint using a two dimensional linear elastic finite element method with plane strain assumption. Shi and Cheng [9] considered the stress distribution in adhesive-bonded cylindrical lap joints for which the two adherends subjected to axial loads may have arbitrary thicknesses and consist of different materials and the adhesive layer may be flexible or inflexible. Her [10] presented a simplified one dimensional model based on the basic elasticity theory and obtained analytical solutions of shear stress in the adhesive and longitudinal stress in the adherend. Silva and Adams [11] investigated how to decrease the transverse stresses in the composite and so to increase the joint strength, particularly at low temperatures, thus making the use of a mixed adhesive joint more justifiable. Ozel et al. [12] investigated mechanical properties of adhesively bonded single-lap joint geometry with different configurations of lower and upper adherends under tensile loading.

Kaye and Heller [13] developed an optimal design of free-form bonded and double lap joints, with the

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aim of achieving reduced peel stresses on the bondline region.

The effect of the length and depth of a parallel slot as well as the elastic modulus of the adhesive on the stress distribution at the mid-bondline and in the adherend was investigated by Yan et al. [14] using the elastic finite element method. In the study of Gültekin et al. [15], mechanical properties of different single lap joint configurations with different adherent width values subjected to tensile loading were investigated experimentally and numerically. In the work of Pinto et al. [16], the effect of adherend recessing at the overlap edges on the tensile strength of single lap joint, bonded with a brittle adhesive, was experimentally and numerically studied.

The main objective of this paper is to investigate strength of different end part of the adherend geometries in single lap joints subjected by tensile load. For this purpose, stress analyses of the adhesively bond models with six different end part of adherend were performed by using finite element method. In addition, the effect of the change of the adhesive Young's modulus in lap joints with six different models on the maximum shear and normal stress in the adhesive were investigated. The results obtained from each models were compared with each other.

2 Finite Element Model

The aluminium alloy 7075 was selected for the upper and bottom adherend in all model analyses, which has Young's modulus (E) of 71.7 GPa and a Poisson's ratio of 0.33. A wide range of adhesive Young's modulus, including 2.5, 4, 5.5, and 7 GPa were used. In addition, adhesive Poisson's ratio of 0.3 was used in all finite element analyses. The adhesive thickness (ta) is 0.2 mm and the length of bondline (2c) is 12.5 mm in all joint models as shown in the Fig. 1. Bonded lap joint with two thin aluminum adherends of various materials having the following dimensions: length (l) of 100 mm, width (w) of 25 mm and a thickness (t) of 2 mm were considered. A finite element analysis was performed considering an applied static force of 3 kN (Fig. 2).

The geometry and boundary conditions of the model are shown in Fig. 2. The models consist of approximately 96 138 nodes, and 21 112 hexahedral elements (see Fig. 3). The element was composed of eight different nodes with three degrees of freedom.

The choice of the mesh type was an important factor on numerical calculation efficiency [17]. Therefore element numbers are determined according to the prior works [4, 14]. Geometric nonlinearity was considered in all analyses.

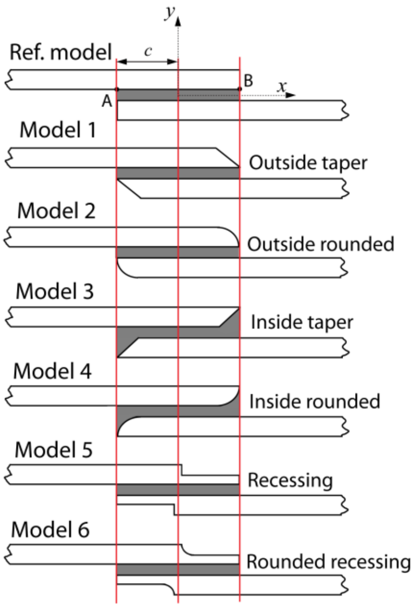

Figure 1. Various shape of the adhesively single lap joints.

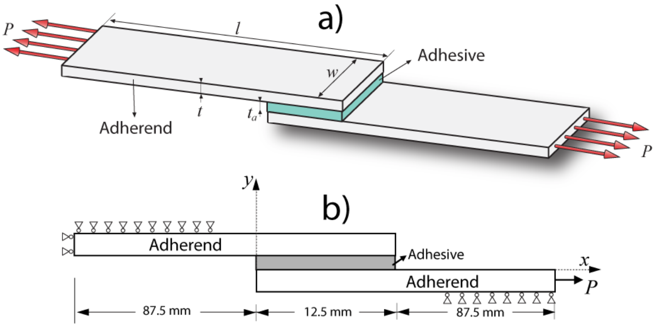

Figure 2. a) Three dimensional single lap joint, b) Boundary conditions and geometric dimensions of bond joint.

3 Result and discussion

The adhesively single lap joint 3D models shown in Figure 1 and 2 were analyzed by using finite element package software ANSYS (Swanson Inc., Houston, PA).

The failure begins at the ends of overlap length of adhesively bonded joints, since maximum stress concentrations occur at the ends of overlap length of adhesively bonded joints subjected to tension loading. Therefore, critical regions of stress distributions at the bonding area were divided into smaller elements. Also, the meshing in adhesive and adherend in bondline region was performed in a more sensitive manner by dividing it into small pieces as shown in Fig. 3.

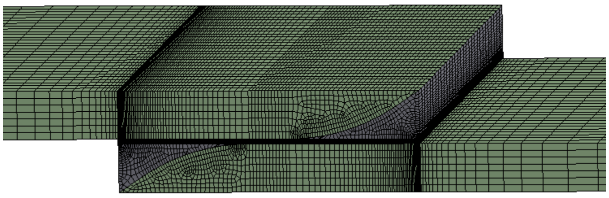

Figure 3. Finite element mesh of Model 4.

The stresses that cause the failure of joints include the shear stress and transverse normal (peeling) stress in the adhesive layer [10]. Therefore, the peel and maximum shear stress in adhesively joint models were taken into account.

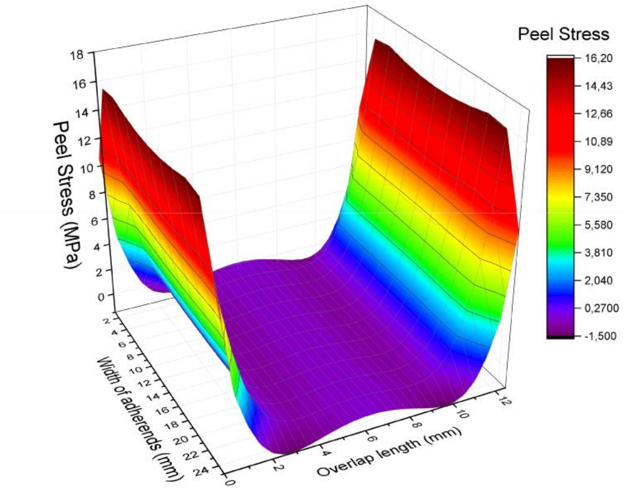

Figure 4. 3D stress distributions on the adhesive.

3D peel stress distributions for reference model have been presented in Fig. 4. In the 3D finite element analysis, stress distributions were plotted at the mid-width along the overlap length (see Fig. 5).

The peel and shear stress distributions for the adhesively single lap joints have different geometric shapes, as shown in Fig. 5. Fig. 5 represents the results in which, Young's modulus of the adhesives assumed 2.5 GPa.

The peek values of peel stresses at the interface between adhesive and adherend along bondline in model 1, 2, 3, 4, 5 and 6 are less than 13.9, 37.3, 35.2, 54.4, 72.1 and 72.3%, respectively, to the reference model. The changes in the peel stresses can be seen in Fig. 5.

The peak values of maximum shear stresses along the overlap region at the interface between adherend and adhesive in model 1, 2, 3, 4, 5 and 6 are less than 10.7, 28.9, 32.4, 35.1, 20.5 and 21.4%, respectively, to the reference model, as shown in Fig. 5b. This behavior is in qualitative agreement with experimental results [16].

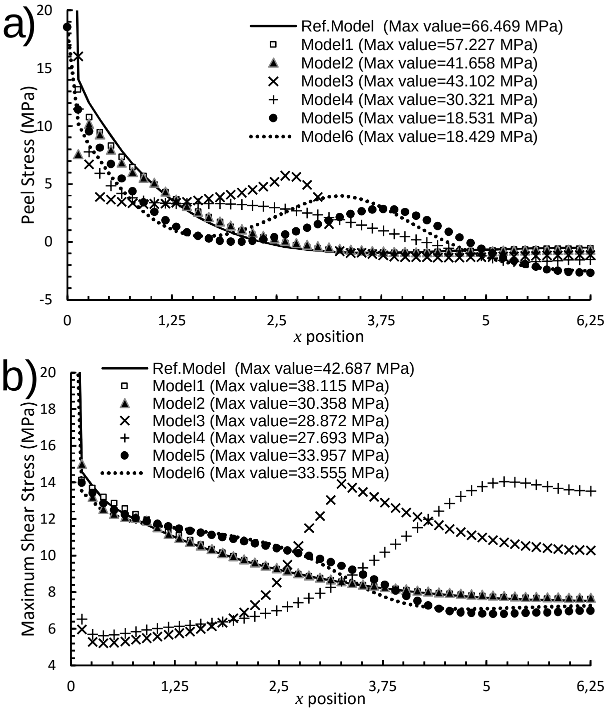

Figure 5. Stress distributions obtained from the adhesive surface-layer throughout c line at overlap length: a) Peel stress distributions b) Maximum shear stress distributions.

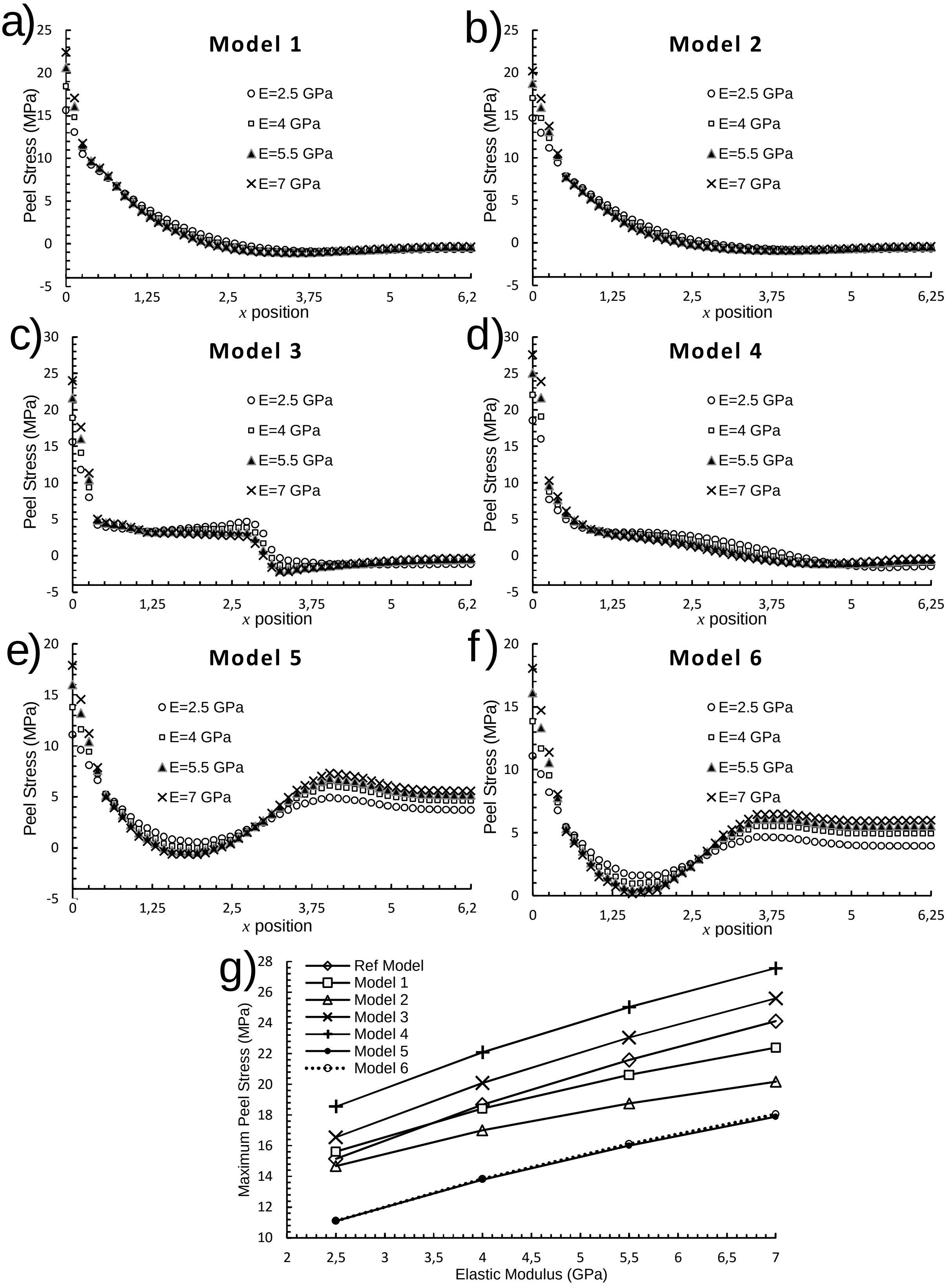

Figure 6. Peel stress distributions along the mid-bondline on the adhesive for different material properties of adhesive.

The peak values of maximum shear stresses of Model 3 and 4 occur in the zone of $2.5 < x < 5.5$ mm. On the other side, the peak values of maximum shear stresses of Model 5 and 6 are located at the point $x = 0$. By contrast, the peak values of maximum shear stresses of Model 1, 2 and reference model are located near $x = 0.3$ points.

With an outside rounded (Model 2), the peel stress is reduced by 27.2% from that of an outside taper (Model 1) joint. Similarly, with a rounded recessing (Model 6), the peel stress is reduced by 0.6% from that of a recessing joint (Model 5).

With an outside rounded, the maximum shear stress is reduced by 20.4% from that of an outside taper joint. Similarly, with a rounded recessing, the maximum shear stress is reduced by 1.2% from that of a recessing joint. Results show that rounded adherend corners reduce the magnitudes of the maximum stresses.

When taking the elastic modulus of the adhesive as a variable, the results of the peel distribution stress in the mid-bondline obtained from finite element method simulation for single lap joints having various shapes are shown in Fig. 6.

The effect of Young's modulus of the adhesives on the stress distribution of joints in the midline was investigated and the results from the finite element analysis are given in Fig. 5. In Model 3 and 4, as the adhesive Young's modulus was increased, the peel stress distributions in the mid-bondline were markedly increased. As the adhesive Young's modulus in Model 1, 2, 5 and 6 was increased, peel stress in the mid-bondline was gradually increased. As the adhesive Young's modulus was increased, the least peel stress increase was obtained in Model 2.

In case of constant tensile load and utilization of adhesive with a high modulus of elasticity, these stresses have higher values compared to the flexible adhesive in the overlap corners, and the stress concentration will hence be higher in these areas.

The values of the maximum peel stress in Model 1, 2, 3, 4, 5 and 6 are increased by 18.7, 10, 6.6, 16, 12.4, 20.3 and 21%, respectively, to reference model in which adhesive Young's modulus is increased from 2.5 to 5 GPa.

The peak value of peel and maximum shear stress of the all model occur at $x = 0$ mm (Fig. 5).

The peak value of peel and maximum shear stress of all the models occur at $x = 0$ mm (Fig. 5).

The peel stresses at free edges of the overlap is very important because they cause initiation and propagation of failure in this region [3]. This situation must be considered by designers.

4 Conclusion

Computational studies were carried out using finite element analyses in order to determine the effects of adherend shape geometry on the peel and shear stress state in adhesively bonded single lap joints. A few joints with different adherend shape configurations were regarded, which include outside taper (Model 1), outside rounded (Model 2), inside taper (Model 3), inside rounded (Model 4), recessing (Model 5), and rounded recessing (Model 6). A comparison to the simple single-lap joint (reference model) was made to make an observation of the percent reduction in stresses for each adherend geometry.

The effects of Young's modulus of adhesive and adherend shape geometry on stress distribution can be concluded as follows:

1) In Model 6, peak value of the peel stress concentration occurring on the free ends of adhesively bonded region is low compared to other models. Reduction of this stress is very effective in initiating damage and this decrease played a significant role in the increase of joint strength.
2) Decrease of Young's modulus of the adhesive leads to the lower peel stress, especially in the Model 6 compared to other joint models.
3) While the effect of adherend recessing on the peel stress reduction in adhesive single lap joint is greater than inside and/or outside tapered adherend geometry, the effect of inside tapered adherend geometry on the maximum shear stress reduction in adhesive single lap joint is greater than adherend recessing and/or outside tapered adherend geometry.
4) Effect of rounding the adherend corners on the stress reduction in bond region are significant especially in tapered adherend geometry compared to adherend recessing.

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