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STRESS ANALYSIS IN ADHESIVE-BUTT JOINTS OF CYLINDRICAL PRESSURE VESSELS

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Abstract: In this study, the stress distributions in adhesive butt joints of cylindrical pressure vessels are analyzed by the finite element method using eight-node-quadrilateral axisymmetric finite elements. The structure consists of two cylindrical adherends bonded by an adhesive and the joint is subjected to an internal pressure. Stress distributions and the effects of the adhesive thickness on the stress distributions in the joint are studied.

1. INTRODUCTION

Modern technology has placed increasing demands on the use of bonded joints in structural applications. Adhesive bonding offers significant advantages over mechanical fasteners, for example in aerospace and automobile composite structures. Adhesive joints have previously been designed empirically. Now, data are available to design adhesive joints in an optimal manner. For optimal design of adhesive joints, accurate knowledge of the stress distributions of the joints is essential.

Up to now, investigations in general have been carried out on lap, scarf and butt adhesive joints of rectangular plates subjected to tensile, bending moment, cleavage and shear loads. However, only a few investigations have been carried out on cylindrical joints in which adherends were joined by an adhesive.

The stresses in adhesive bonded tubular lap joints, subjected to axial and torsional loads, are investigated by Adams and Peppiatt [1], using the finite element technique. Shi and Cheng [2], proposed a closed-form solution based on the principle of minimum complementary energy for the adhesive bonded cylindrical lap joint, subjected to axial load. Cylindrical butt joints subjected to axial load are investigated by Günay at al [5]. The stress analysis of adhesive bonded cylindrical lap joints, subjected to axial load, is investigated by Günay and Aydemir [3] using finite element method. Adhesive-butt joints of cylindrical pressure vessels are also investigated by Günay at al [4] using finite element method.

2. DEFINITION OF THE PROBLEM

The axisymmetric cross-section of the studied joint is shown in the Fig. 1. In the joint, there are two identical adherends. The inner and outer radius of the adherends are a and b, respectively. The adhesive thickness is t. The length of non-slanted portion of the adherends is $3a$.

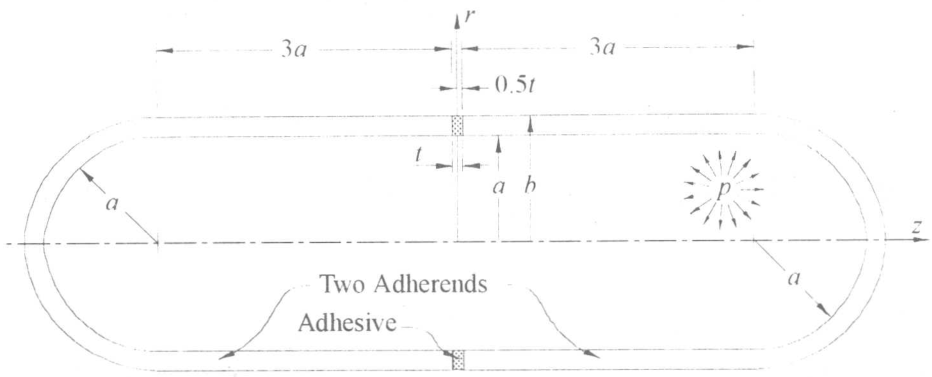


Figure 1. The adhesive-butt joint of cylindrical pressure vessel.

3. FINITE ELEMENT MODEL

Because of the symmetry of the joint, half part of the joint is taken for the finite element model, this means quarter of cross-section of the structure, as shown in the Fig. 2, will be subdivided into finite elements because of axisymmetric finite elements used. Because there exists an axisymmetry with respect to the joint's geometry and material properties, in the finite element solution eight nodes-quadrilateral axisymmetric finite elements are used. The effect of the adhesive layer thickness on the stress distribution is investigated.

3.1 Geometry of the Joint

Inner and outer radius of the adherends are a = 20 mm and b = 25 mm, respectively. The radiusses are taken as constant in all solutions and t is changed from t = 0.50 mm to t = 4.00 mm incrementing by 0.50 mm. Radial axes r_m and axis r_a originated at A and D, respectively. Longitudional axes z_i, z_m and z_o originated at A, B and C, respectively.

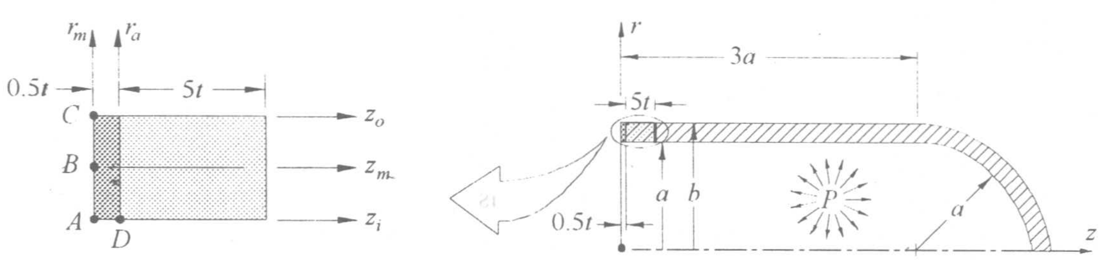


Figure 2. The axisymmetric half part of the joint used for finite element model, and coordinate systems.

Joint dimensions are given in Table 1. All dimensions given in Table 1 are in millimeters. In Table 1 the superscript (*) means given dimension is taken as from first number **T**o second number **I**ncrementing by third number. So, 0.5**T**4**I**0.5 means from 0.5 mm to 4 mm incrementing by 0.5 mm.

Table 1. Joint dimensions.

Notation	Case 1
a	20
b	25
t	0.5**T**4**I**0.5*

Table 2. Material properties.

Adherends	Adhesive
E = 70 GPa	E = 3.5 GPa
$\nu = 0.33$	$\nu = 0.41$

3.2 Material Properties, Boundary Conditions and Applied Load

The material properties used in the all cases are given in Table 2. Material properties of adherends and adhesive are assumed elastic and isotropic.

In all cases, displacements in r-direction at $r = 0$ and displacements in z-direction at $z = 0$ are set to zero for boundary conditions, Fig. 2. The load is applied as a pressure of $p = 1$ MPa to the vessel for all cases.

4. RESULTS

The term "stress analysis" of an adhesively bonded joint is used for investigation of the stress distributions in the adhesive and the region neighbor of adhesive as shown in the Fig. 2. Axsymmetric stress components are σ_r, σ_θ, σ_z and τ_{rz}. From these components only normal stresses are presented. Shear stress τ_{rz} in the middle surface of adhesive is approximately zero. But in the interface of adhesive the shear stress τ_{rz}/p changes from −1.5 at the inner radius to 1 at the outer radius. So, the distributions of the radial stress σ_r, longituditional stress σ_z, hoop stress σ_θ and von Mises equivalent stress σ_e have been presented on the middle surface of the adhesive, i.e. along the axis r_m, and on the adjacent surfaces of the adhesive and adherends, i.e. along the axis r_a.

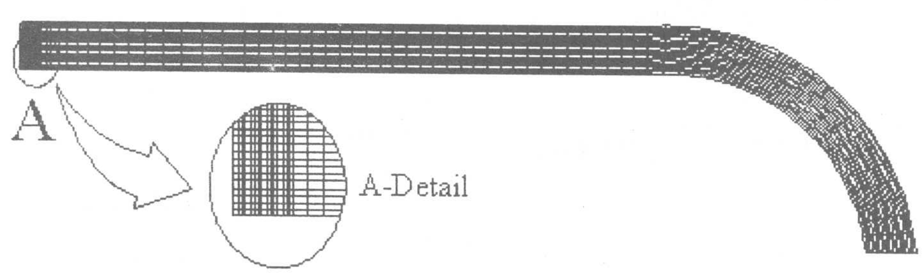


Figure 3. The finite element mesh and A-detail.

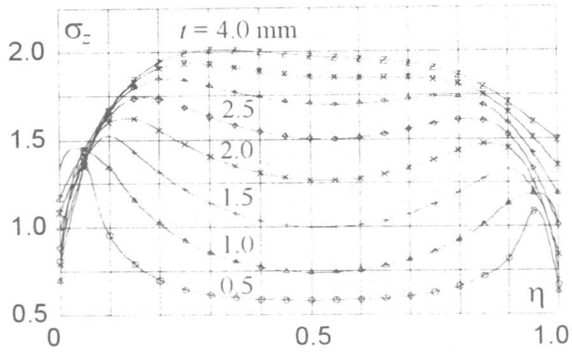


Figure 4. Longituditional stress distribution along r_a.

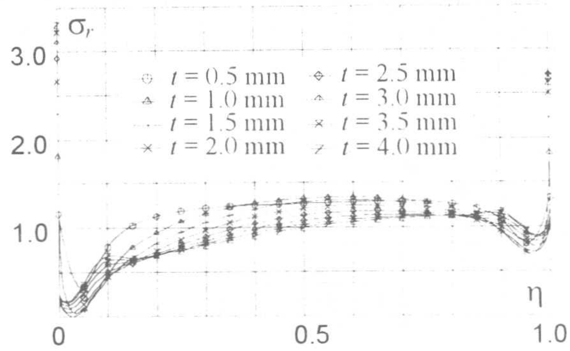


Figure 5. Radial stress distribution along r_a.

Figs. 4-7. All plotted stresses are non-dimensionalized by dividing by the applied internal pressure p, and are placed in vertical direction of all the graphics. In the Figs. 4-7, horizontal axes of the graphics are adherends' thickness and are non-dimensionalized by dividing by the 5 mm. In the Figs. 4-7, hence, the vertical direction is (stress) $/p$ and the horizontal direction is $\eta = r_a/(b-a)$, and $(b-a) = 5$ mm.

There is a discontinuity at the interface of the adhesive and adherends. Stresses at the interface are different on both sides. For the stress distribution, stresses in the adhesive side are plotted, Figs. 4-7.

The adhesive thickness has much effects on the stresses. However, only the stress concentrations at the inner and outer surfaces significantly differ, Figs. 5-7. The stress distributions between the inner and outer surfaces does not nearly change. Only the distribution of σ_z changes with the adhesive thickness between the surfaces, Fig. 4. At the inner surface of the adhesive radial stress σ_r has a positive value, Fig. 5. This means adhesive at the interface subjected to a positive radial stress. This is because of different material properties of the bonded and bonding materials.

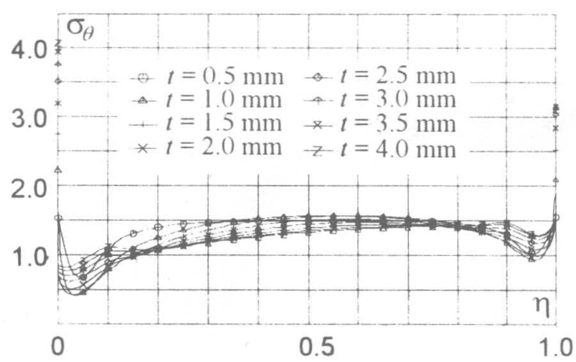


Figure 6. Hoop stress distribution along r_a.

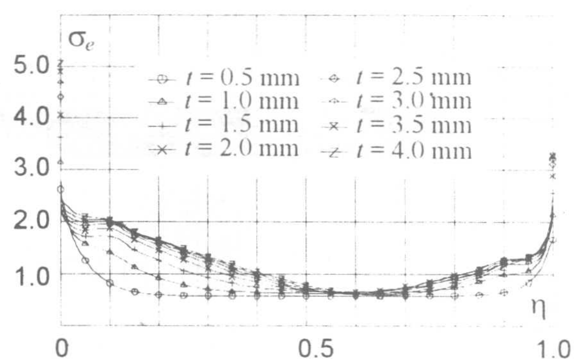


Figure 7. Von Mises stress distribution along r_a.

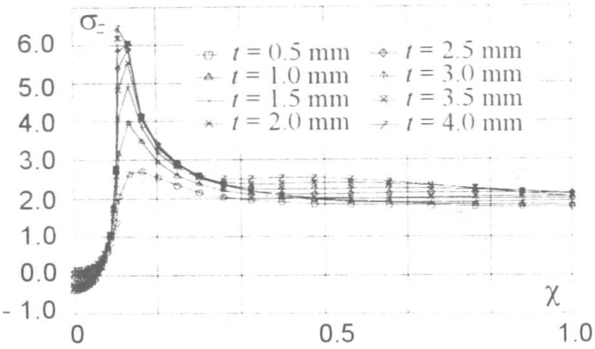


Figure 8. Longituditional stress distribution along z_i.

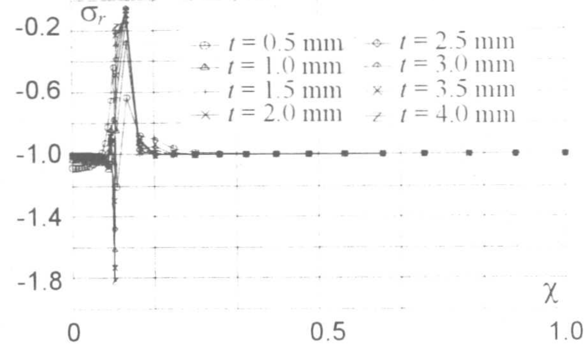


Figure 9. Radial stress distribution along z_i.

Figs. 8-11. In the Figs. 8-11, horizontal axes are z_i and are non-dimensionalized by dividing by the $5.5t$. So, in the Figs. 8-11, the vertical direction is (stress)/p and the horizontal direction is $\chi = z_i/(5.5t)$.

For the stress distribution, averaged stresses at the interface are plotted, Figs. 8-11. With the increase of the adhesive thickness stress concentrations in all plotted stresses increase, Figs. 8-11. One of the important results is that the maximum values of the stresses not exactly occur at the interfaces between adhesive and adherends. They occur in a place near the interfaces in the adherends.

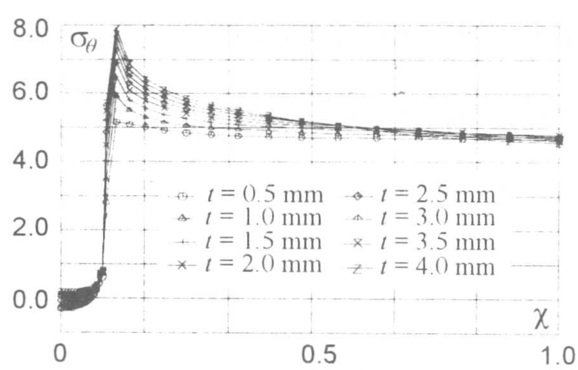


Figure 10. Hoop stress distribution along z_i.

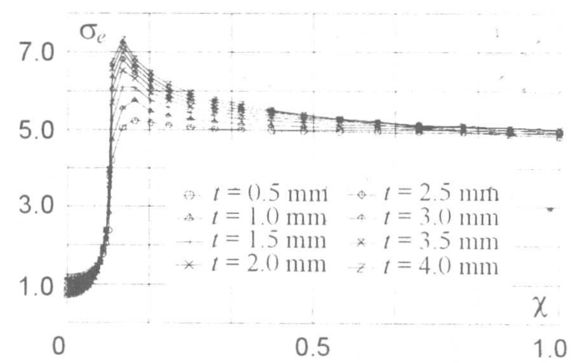
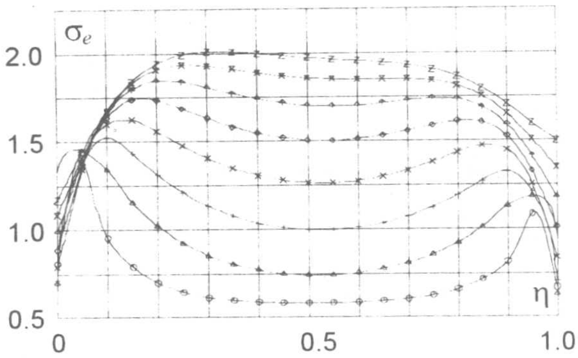


Figure 11. Von Mises stress distribution along z_i.

Figs. 12-14. In this group of graphics only von Mises design stress distributions are plotted. In the Fig. 12, horizontal axis is r_m and is non-dimensionalized by dividing by the adherends' thickness, i.e. $\eta = r_m/(b-a)$, and $(b-a) = 5$ mm. In the Fig. 13 horizontal axis is z_o, in the Fig. 14 horizontal axis is z_m and both are non-dimensionalized by dividing by the $5.5t$. So, in the Figs. 12-14, the vertical direction is (stress)/p and the horizontal direction is χ.



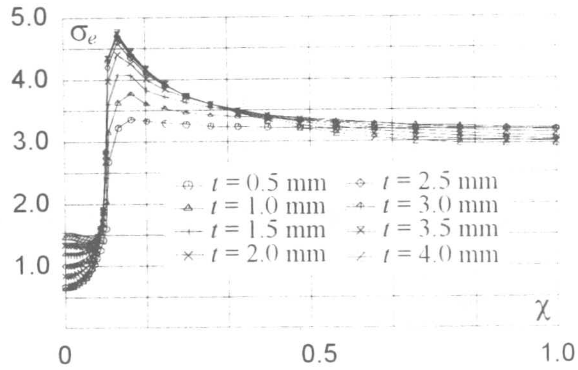


Figure 12. Von Mises stress distribution along r_m. **Figure 13**. Von Mises stress distribution along z_o.

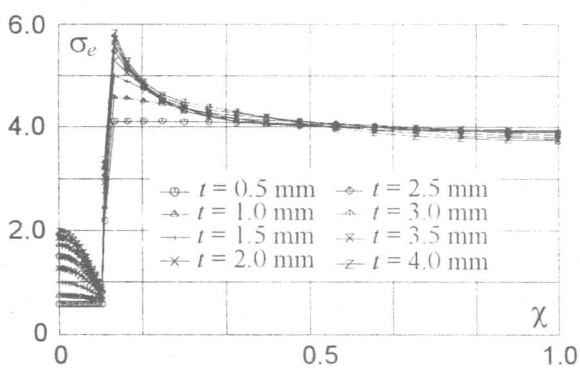
Along the adhesive middle axis r_m, stress concentration increases with the adhesive thickness, which is $(b - a)$, Fig. 12. In the Figs. 13-14, at the interface nodes, averaged stresses are plotted. It is the same that with the increase of the adhesive thickness stress concentrations in all stresses increase, Figs. 12-14, and the maximum values of the stresses not exactly occur at the interface but the place near the interface in the adherends, Figs. 13-14.

Fig. 15. In the Fig. 15, maximum values of von Mises stresses along the axes z_i, z_m and z_o are plotted in vertical direction. Horizontal axis is the adhesive thickness. From that graphic, these observations can be made:

i) Maximum value of the von Mises stress increases with the adhesive thickness,

ii) for the same adhesive thickness maximum value of the stress changes along the axis r, and

iii) for the same adhesive thickness maximum value of the stress occurs at the inner surfaces.



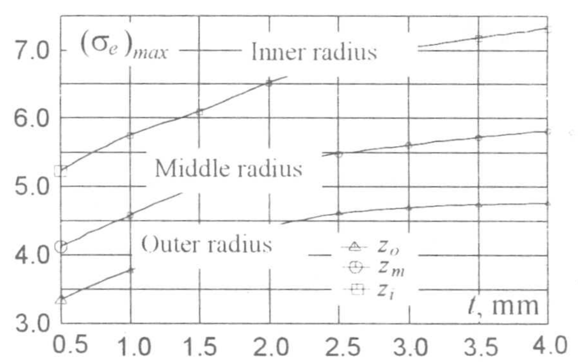


Figure 14. Von Mises stress distribution along z_m. **Figure 15**. Maximum values of von Mises stress.

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