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THREE DIMENSIONAL STRESS ANALYSIS IN ADHESIVELY BONDED JOINTS

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Abstract- Stress and strain distributions in the adhesive bonded joints subjected to distributed forces are investigated using finite element method. Two different cases are considered, the bonded materials are the same, and the bonded materials are different. The investigations are conducted on a three dimensional model. The finite element model of the joint is obtained using isoparametric three dimensional elements having eight nodes with three degrees of freedom each. The stress components and their distributions both on adhesive surface and on metallic elements are given in dimensionless form using three dimensional graphics.

1. INTRODUCTION

There is great technical interest in the analysis of adhesive joints. The main purpose is to assess the severity of the load carried by the joint. Adhesive joints have come to be widely used in the fields of aeronautics, space, automobile, semiconductor, and other industries with the recent development of tough structural adhesives and the substantial improvement in the strength of adhesive joints. Adhesive joints have come to be used in mechanical structures with the improvement in the properties of adhesive bond. They have not been used in principle load-bearing parts of structures.

Early theoretical studies of the stresses in adhesive joints were directed towards the single-lap joint. The paper by Goland *et al.* [1] is regarded as a classical work in the area of static analysis of a simple lap joint. The peel and shear stresses were assumed to be constants across the adhesive thickness in the analysis. Finite element method for the single lap joints is first used by Wooley and Carver [2]. They presented stress concentrations as functions of geometry and material parameters. The plane stress model has been used in that paper. In the paper by Erdoğan and Ratwani [3], the stress distribution in plates bonded through stepped joints is analyzed. The problem is solved under the assumption of plane stress. In reference [4] the development of linear analytical procedure for design of flat bonded joints with tapered adherends is presented. Reducing

the shear stress peaks in the adhesive by tapering the adherends from the nominal thickness often creates a secondary tensile stress concentration in the adherends. A similar study is carried out by Chang and Muki [5]. Lap joints, stepped joints and scarf joints are analyzed by Yuceoğlu and Updike [6] using analytical formulation. Chen and Cheng [7] presented a closed form solution. Special attention is given to the stress distribution in the end zones where high stress intensities occur.

In a different paper by Chen & Cheng [8] the stress distributions in a plane scarf joint is analyzed. The butt joint is treated as a special case of scarf joints. Cheng *et al.* [9] analyzed the joint for which the two adherends could have different thicknesses and lengths, and consist of different materials. Kaya [10] investigated stress distributions in adhesive bonded lap joints, single and double, under tension force by the finite element procedure and the distributions of the stresses are plotted in nondimensional coordinates. The work considers nonidentical adherends regarding and disregarding the thickness of adhesive layer to find out the effects of the parameters on the stress distribution. In the work by Lin and Lin [11] a finite element formulation for analyzing the stresses in the adhesive of a single-lap joint is presented. The element is based on the Timoshenko beam theory and an assumed variation of the transverse shear stress and transverse normal stress through the thickness of the adherends.

The three-dimensional nature of the state of deformation in a single-lap test specimen is investigated by Tsai and Morton [12] in a linear elastic finite element analysis in which the boundary conditions account for the geometrically non-linear effects. There are many analytical and numerical studies in the literature for double lap joints, butt joints, and cylindrical adhesive lap joints subject to torsion [13, 14]. There are a few works on three dimensional finite element stress analysis for adhesive bonded joints. Much of the work deal with the subject as an in-plane problem. In this study two kinds of adhesively bonding joint problem are numerically investigated. These are bonding of identical and non-identical metallic adherends.

2. ADHESIVELY BONDED SINGLE LAP JOINT FOR IDENTICAL ADHERENDS

Adhesively bonded single lap joint under axial tension force is shown in Fig. 1. Joint geometry, physical dimensions, and loading of the problem for stress analysis is also indicated in the same figure. The numerical values for the parameters are given in Table 1. This problem was investigated by some authors assuming a one or two dimensional problem. In fact distributions of the stress components are not uniform across the width [y-axis]. With these considerations, a three dimensional model is considered in the work. So, variations of the stresses along the width are investigated.

2.1. Material Properties of the Single Lap Joint

The materials are assumed as isotropic and homogenous aluminum. Adhesive is also modeled as an isotropic material. Mechanical properties of selected material are given in Table 2.

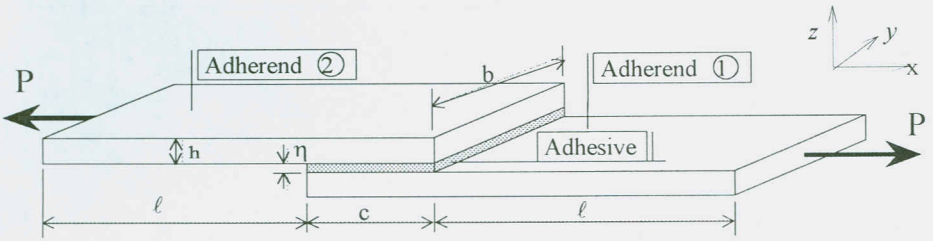


Figure 1. Three dimensional adhesive-bonded single lap joint system

Table 1. Parameters of the single lap joint

P = 3.2 kN	c = 80 mm	b = 40 mm	η = 0.9 mm	h = 16 mm	ℓ = 200 mm

Table 2. Material properties of the single lap joint

Elasticity modulus for adherend (Aluminum)	E_1 = 70 GPa
Elasticity modulus for adhesive	E_A = 3.5 GPa
Poisson's ratio for adherend (Aluminum)	υ = 0.33
Poisson's ratio for adhesive	υ = 0.34

2.2. Finite Element Model and Boundary Conditions

Finite element model of the problem is shown in Fig. 2. Hexahedral elements with eight nodes having three degrees of freedom each are used in the analysis. The degrees of freedom are the displacements u, v, and w in the direction of x, y, and z axis respectively. The model consists of 4048 elements, and 5148 nodes. The boundary conditions are also given in Fig. 2. Six elements were used through the thickness of the adherend, and two elements through the thickness of the adhesive layer. The meshes in the adherends and adhesives were relatively fine. It was seen that refining of mesh for present problem didn't give rise to any considerable changes for stress components. The computations are performed using ANSYS software. The derivation of the stiffness matrix and other issues can be found in reference [15].

The variation of stress components are plotted along the middle-plane of adhesive layer. In the first solution, stress components, (normal stresses σ_x, σ_y, σ_z, shear stresses τ_{xy}, τ_{yz} τ_{xz}, principal stress σ_1 and equivalent stress σ_{eqv}), are obtained. The variations of stress components are plotted in Figs. (3-10). Values of the normal, principal, equivalent and shear stresses are divided by nominal stress $[\sigma_0]$ and $[\tau_0]$ respectively, and distributions are shown as a function of the non dimensional coordinates, $[\sigma/\sigma_0]$ or $[\tau/\tau_0]$ in the z direction, [x/c] in the x direction, [y/b] in the y direction. There is another important point which should not be ignored that the origin of the plots in Figs. 3 to 10 is not on the left side, but on the right side. As it should be, the maximum normal stress values occur as σ_z and maximum shear stress values as τ_{xz}. σ_x and σ_y are smaller than σ_z, and τ_{xy} and τ_{yx} than τ_{xz}.

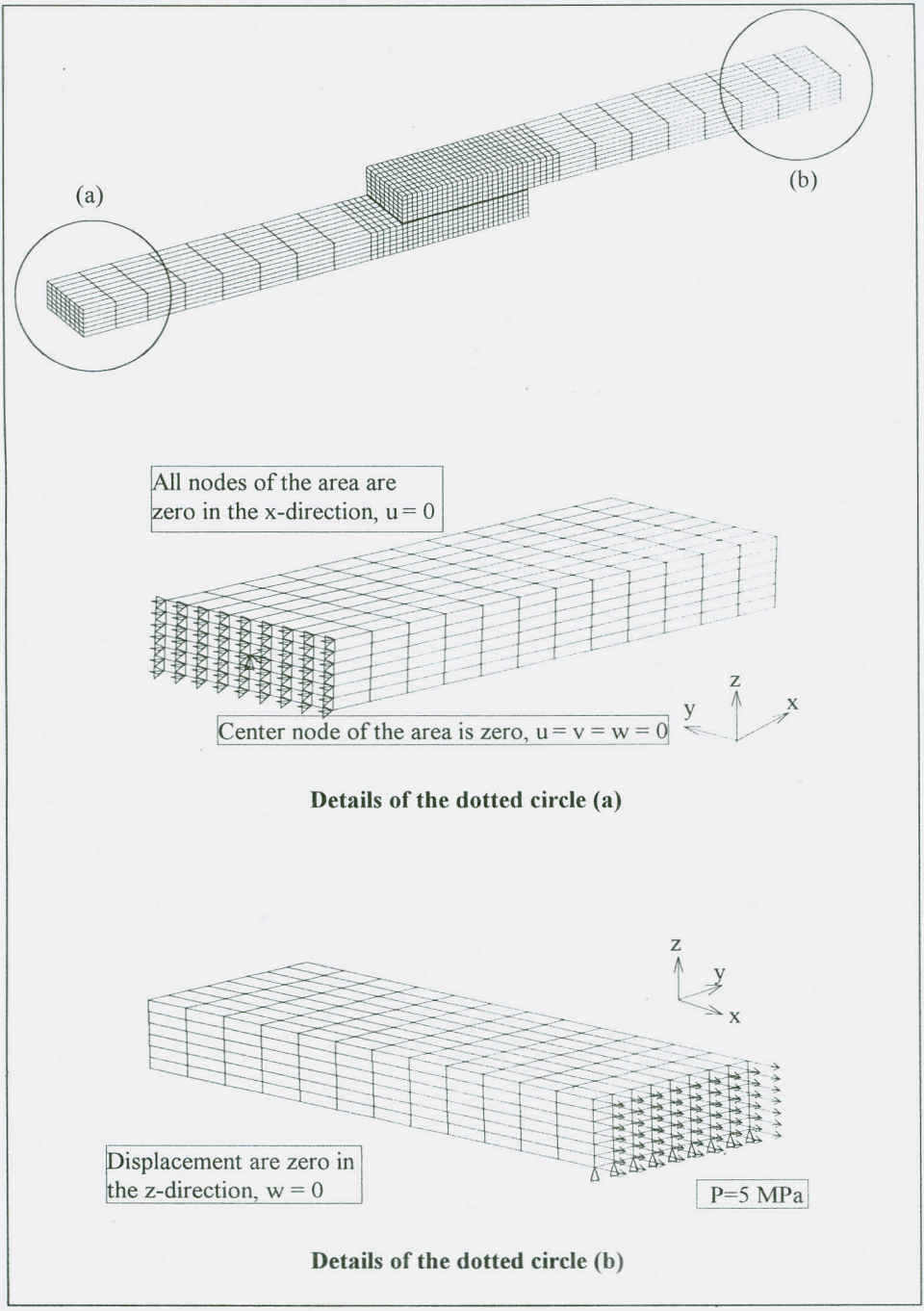


Figure 2. Boundary conditions and finite element model of the system

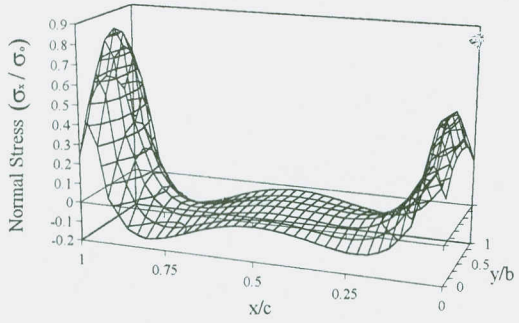


Figure 3. Variation of the normal stress for identical adherends

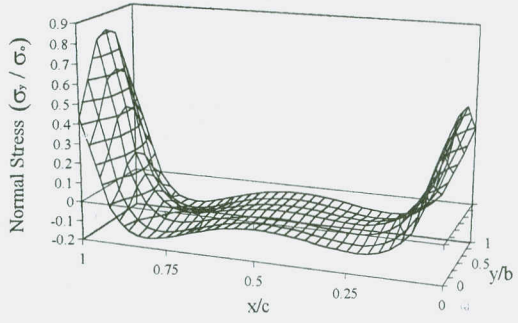


Figure 4. Variation of the normal stress for identical adherends

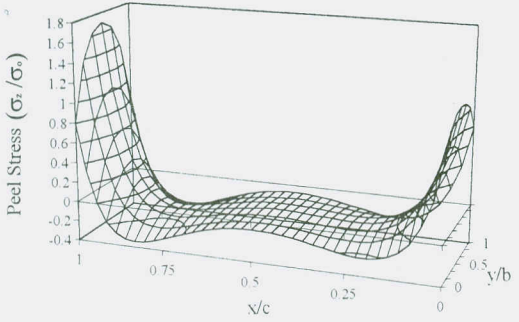


Figure 5. Variation of the peel stress for identical adherends

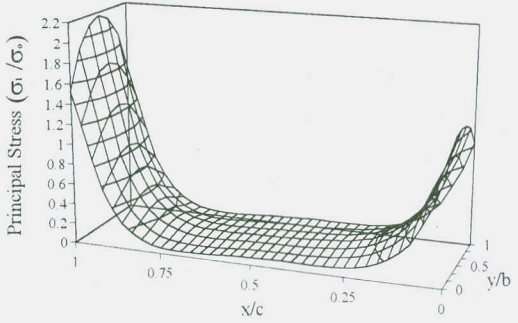


Figure 6. Variation of the principal stress for identical adherends

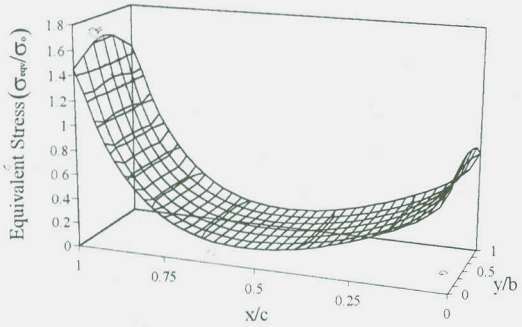


Figure 7. Variation of the equivalent stress for identical adherends

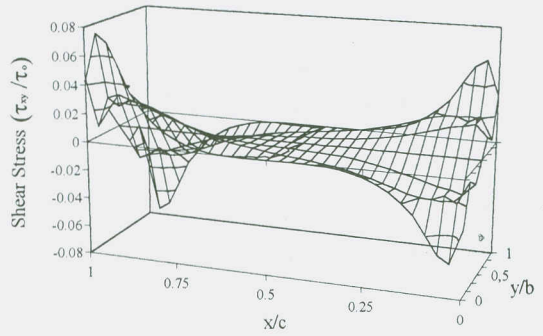


Figure 8. Variation of the shear stress for identical adherends

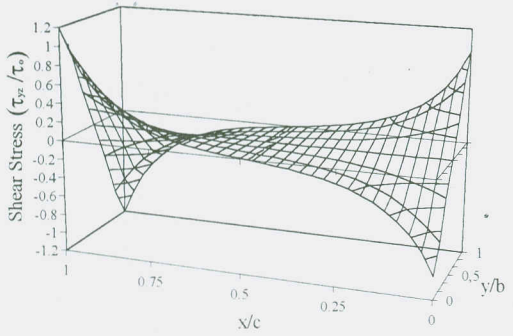


Figure 9. Variation of the shear stress for identical adherends

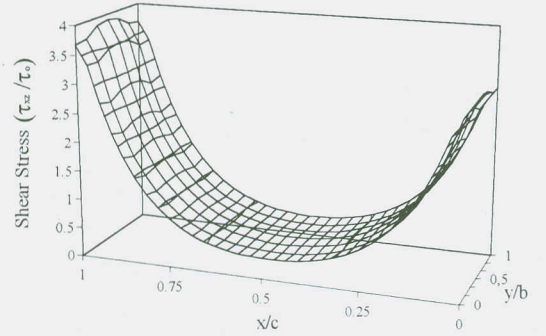


Figure 10. Variation of the shear stress for identical adherends

3. ADHESIVELY BONDED LAP JOINT FOR NON-IDENTICAL ADHERENDS

The bonding of different materials for various applications can be required. As it is well known the welding of different metallic materials needs very special methods. In addition the adhesive bonding has several advantages if sealing is important. Aluminum and steel are selected as different materials for the present study.

The details of the joint are the same as in Fig. 1. The dimensions of the joint are also the same as in Table 1. The material properties are given in Table 3.

Table 3. Material properties of the single lap joint

Elasticity modulus for adherend (Aluminum)	$E_1 = 70$ GPa
Elasticity modulus for adherend (Steel)	$E_2 = 200$ GPa
Elasticity modulus for adhesive	$E_A = 3.5$ GPa
Poisson's ratio for adherend (Aluminum)	$\upsilon = 0.33$
Poisson's ratio for adherend (Steel)	$\upsilon = 0.30$
Poisson's ratio for adhesive	$\upsilon = 0.34$

The mesh used for this study is the same as for identical materials. The results obtained for this case are given in the form of previous case. Figs. 11 to 18 show stress components for the middle plane of adhesive layer. The examination of plots reveals that there aren't much important differences in stress values for the identical and non-identical cases. The reason for this situation is that the given stress values are for the middle section of the adhesive layer [15]. If the stresses were given for the adherends, the differences might then occur.

4. DISCUSSION AND CONCLUSIONS

In this work, adhesive bonded joints are modeled using three dimensional finite elements. It is assumed that the joints are subjected to in-plane forces. Two cases are considered in the analysis, that is, the bonded materials are identical, and non-identical.

Stress distributions in the adhesive region indicate that the stresses in critical regions, namely at the end points of lap area, are maximum. It is observed that the stress distribution along width (y-axis) is non-uniform. The stress components in the middle plane (along $y = 20$ mm, Fig. 1) is greater than the components in the end plane. Normal stresses σ_x and σ_y are nearly the same for the identical and non-identical joints. Indeed the identical case gives slightly greater values.

If plane stress or plane strains were assumed, then there were no normal stresses along the third dimension. But the present study indicates that the normal stresses are important like the longitudinal stresses. Figs. 4 and 12 clearly show this fact.

When the plots for σ_x given in Figs. 3 and 11 are investigated, it is seen that the stress values in free edges (that is in $y = 0$ and in $y = 40$ mm) are in between 0.2 and 0.45 while

they are nearly about 0.85 in middle plane (that is in y = 20 mm). Maximum value for σ_x occurs inside of the end region of lap area but not just on the edge.

It is observed that the peel stresses (σ_z) given in Figs. 5 and 13 are more important than the other normal stresses (σ_x and σ_y). The plots show in addition that the stresses along with y = 0 and y = 40 mm decrease about 0.6 while they become nearly three times, that is 1.7 to 1.8, in the middle plane (y = 20 mm). These values are smaller than that of the one or two dimensional solutions [1, 2, 10,16, 17].

A designer needs equivalent stress values. These are given in Figs. 7 and 15 based on Von-Mises failure criteria.

Shearing stress τ_{xz} is the greatest but the shearing stress components τ_{xy} and τ_{yz} can not be ignored. If the assumptions of plane stress or plane strain were used, only one shearing stress component might be calculated. But the present study clearly shows that the other shearing stress components can not be ignored for all cases. The plots for shearing stresses are given in Figs. 8 to 10 and 16 to 18.

The shearing stress components τ_{yz} and τ_{xy} become zero both in the middle plane (y = 20 mm) and near the middle plane. These components increase towards the corners of the lap region and they have maximum values at the corners. The numerical values can be determined from Figs. 8, 9, 16, and 17.

One of the most important stress components in an adhesively bonded joint is the shear stress τ_{xz}. This component is greater than the others. The results can be extracted from Figs. 10 and 18. The maximum values occur in the middle plane (y = 20 mm). The equilibrium conditions require that these values should be zero in the edge of the lap region. The shear stress τ_{xz} decrease from the middle plane to the free surface.

The two most important stress components in an adhesively bonded joint are the peel stress σ_z and the shear stress τ_{xz}. These components have maximum values near the end of the bonded region. The maximum value of the shearing stress τ_{xz} is given by some authors as approximately 3 in dimensionless form (τ_{max}/τ_{avr}), [1, 2, 10, 12, 16, 17]. In the present study the results are obtained as 3.954 for identical adherends, and 3.902 for non-identical adherends.

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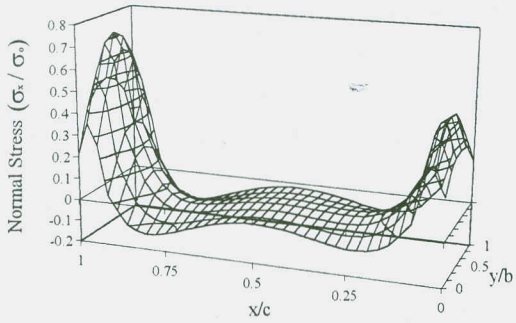


Figure 11. Variation of the normal stress for non-identical adherends

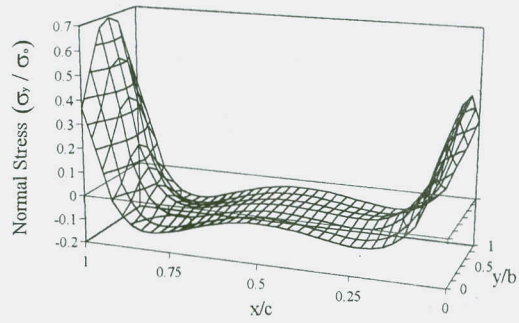


Figure 12. Variation of the normal stress for non-identical adherends

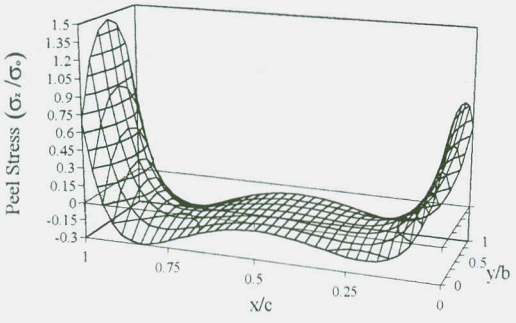


Figure 13. Variation of the peel stress for non-identical adherends

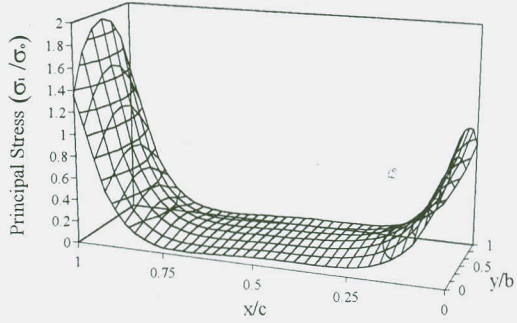


Figure 14. Variation of the principal stress for non-identical adherends

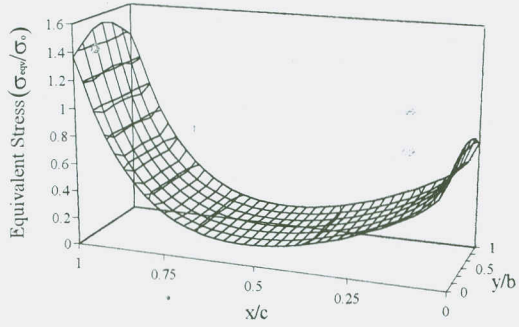


Figure 15. Variation of the equivalent stress for non-identical adherend

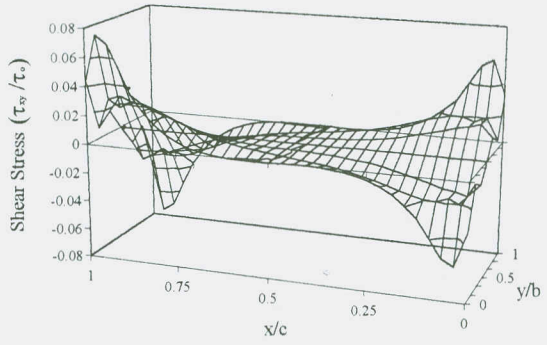


Figure 16. Variation of the shear stress for non-identical adherends

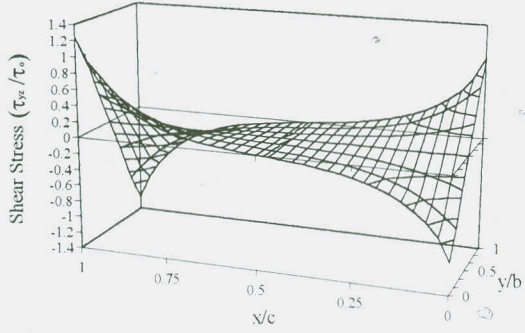


Figure 17 Variation of the shear stress for non-identical adherends

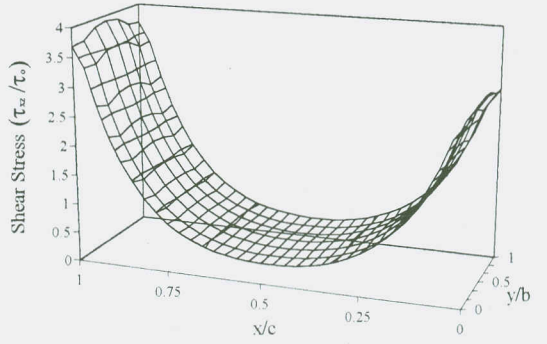


Figure 18. Variation of the shear stress for non-identical adherends