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The Influence of the Geometry of Coaxial Adhesively-Bonded Joints on the Transmitted Load

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Abstract

For the assembling of dissimilar material or of composite materials, the use of adhesive for the design of assemblies can reduce the cost and the weight of structures. Adhesive joining techniques do not require holes, such as for riveted or bolted joints, which can lead to stress concentrations, but adhesively bonded joints are often characterized with large edge effects associated with geometrical and material parameters. In the case of single lap type joints, peel and cleavage forces strongly limit the transmitted load of the assembly despite various techniques proposed to limit the influence of edge effects. Cylindrical joints are associated with large strength of the substrates in the radial direction; thus, peel and cleavage forces have different effect with respect to simple lap joints. But, for such assemblies, edge effects also exist. The objective is to analyse the effect of various geometries of the different parts of the assembly in order to optimize the maximal transmitted load of such joints. In the case of axial loads, the stress distributions are analysed using axisymmetric theory of elasticity. A pressure-dependent elastic limit of the adhesive is used, in order to accurately represent the difference between tensile-shear and compression-shear loads in the mechanical response of the adhesive. Designing adhesively bonded assemblies which strongly limit stress concentrations can significantly increase the load transmitted by the assembly. The first aim is to analyse the possibility of using experimental results of single lap shear specimens to design cylindrical type joints. Secondly, the influence of the angle of conical geometries of the bonded area, which can be easily used for coaxial assemblies, is analysed with respect to the stress distributions. Examples of assemblies of tubes of the same diameters are analysed. Moreover, the influence of several geometries which strongly limit stress concentrations and which have been designed for other geometries of bonded joints, are proposed.

Keywords: coaxial joint, adhesion, stress concentration, finite element analysis, joint design.

1 Introduction

The use of adhesive for the design of assemblies can reduce the cost and the weight of structures, especially in the case of assembling dissimilar material or of composite materials, but a lack of confidence limits the current use of this technology [1-3]. Thus, the optimisation of the design of adhesively-bonded assemblies requires accurate numerical tools which have to take into account the possible stress concentrations due to edge effects [4-6]. Stress concentrations can contribute to the initiation and the propagation of cracks in the adhesive. Various simplified models have been proposed in order to describe the behavior of some bonded joints using 1D or 2D models, but often such models are not able to describe the effect of stress singularities which have often an influence on the maximum load transmitted by the bonded assemblies. Therefore, understanding the stress distribution in an adhesive joint can lead to improvements in adhesively-bonded assemblies; for instance, the design of assemblies which strongly limit the edge effects can be very interesting.

The objective of the paper is to analyse the influence of various geometries of the bonded area of a coaxial assembly in order to optimize the maximal transmitted load. In the case of axila loads, the stress distributions are analysed using axisymmetric theory of elasticity. A pressure-dependent elastic limit of the adhesive is used to accurately optimize the maximum transmitted load of coaxial joints. Such models are often used to represent the behaviour of polymers, and they allow an accurate description of the difference between tensile-shear and compression-shear loads in the mechanical response of the adhesive [7-9]. In the first part, the differences between tensile and compressive loads are analysed using cylindrical joints. Such model can be seen as a first approach in order to analyse the influence of bending loads in a perpendicular direction of the assembly axis which are, for instance, design cases for launchers. Moreover, a comparison between the mechanical behaviour of cylindrical joints and single lap joints is proposed starting from 2D refined finite element computations under elastic assumptions. The aim is to analyse the possibility of using experimental results of single lap shear specimens to design cylindrical type joints. Secondly, the influence of the angle of conical geometries of the bonded area, which can be easily used for coaxial assemblies, is analysed with respect to the stress distributions. Examples of assemblies of tubes of the same diameters are analysed. Moreover, the influences of several geometries which strongly limit stress concentrations and which have been designed for other geometries of bonded joints, are proposed.

These results underline the strong influence of various parameters on the stress state in the adhesive in an assembly. Therefore, precise numerical simulations are useful in order to optimize the design of industrial bonded assemblies taking into account complex geometries and complex loads.

The finite element simulations were made with the code CAST3M (CEA, Saclay, France) [9]. Numerical results are presented in the case of aluminium substrates of 10 mm thickness. Unless stated otherwise, the numerical results are presented in the case of an adhesive thickness of 0.2 mm.

2 Parameters of the numerical simulations

Computations were made in 2D (under plane stress, plane strain or axisymmetrical assumptions). Results are presented for aluminium substrates (Young's modulus: E_a = 75 GPa, Poisson's ratio: ν_a = 0.3) and the elastic material parameters for the adhesive are: E_j = 2.0 GPa, ν_j = 0.4.

In order to correctly determine the way the stress evolves throughout the thickness of the adhesive, precise finite element analyses have to be performed, even assuming a linear elastic behaviour of the components, especially for large material heterogeneity of the assemblies [4-5]. Moreover refined meshes are also needed near the substrate-adhesive interfaces in order to obtain good numerical results. Various simulations have shown that good numerical results are obtained using meshes with 20 linear rectangular elements for a 0.1 mm thickness of adhesive, especially when stress concentrations are not too large [5]. Moreover, it is important to note that the aim of this study is to analyse specific geometries which significantly limit the influence of edge effects.

An optimization of the transmitted load of adhesively-bonded joints requires taking into account an accurate limit criterion, in order to represent the real behaviour of the adhesive in an assembly. Various studies underline that an accurate representation of the elastic yield surface of an adhesive requires the use of a pressure-dependent constitutive model, i.e. a model taking into account the two stress invariants, hydrostatic stress and von Mises equivalent stress. For this study, only elastic behaviour of the adhesive is used.

a (SI)	b	p_{t0} (MPa)
1. E-6	4.87	31.7

Table 1. Material parameters for the initial yield surface.

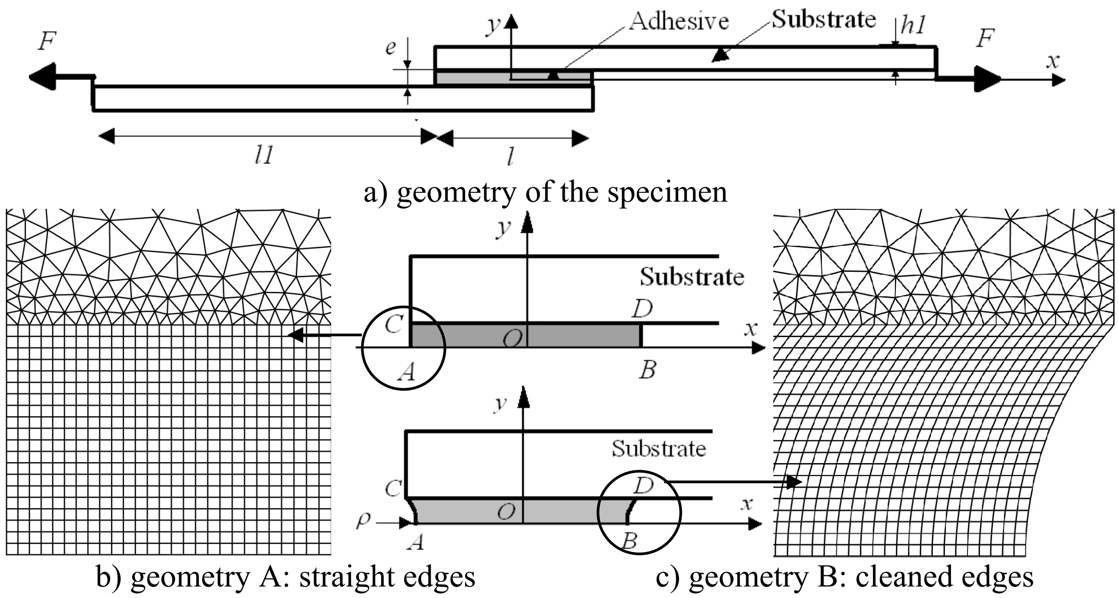
Starting from the experimental results obtained for the epoxy resin Huntsman™ Araldite® 420 A/B, with the modified Arcan device (adhesive thickness of 0.4 mm and displacement rate of the tensile machine crosshead of 0.5 mm/min) it is possible to define the initial yield function for the adhesive (neglecting viscous effects). This test allows to determine the behaviour of an adhesive in an assembly under radial loadings (tensile/compression-shear loadings) [11]. An exponential Drucker-Prager yield function allows a good representation of the experimental data for the so-called initial elastic limit [12]:

$$E = m \cdot c^2 F_0 = a\,(\sigma_{vm})^{\,b} + p_h - p_{t0} = 0 \tag{1}$$

Where σ_{vm} is the von Mises equivalent stress and p_h is the hydrostatic stress. a, b and p_{t0} are material parameters. The results of the identification are presented in Table 1.

3 Stress distribution for single lap shear specimens

Figure 1 presents the main geometrical parameters, of the single lap shear joint, which are used herein. The substrate length and the overlap length are denoted by l_1 and l. The substrate and adhesive thicknesses are denoted by h_1 and e. Numerical results are presented for a substrate length ($l_1 + l$) of 110 mm. Two geometries of the adhesive free edges are used in order to emphasize the influence of this geometrical parameter on the stress concentrations (geometry A: straight edges, $\rho = \infty$; geometry B: cleaned edges, $\rho = 0.75\ e$; e being the adhesive thickness). In the case of single lap shear specimens, various studies have been developed on the influence of spews and chamfers [13] or slots [14]. Figure 1 also shows the mesh of the adhesive close to the free edges and the mesh of the substrate close to the substrate-adhesive interface in order to respect the properties previously presented. Such types of meshes are also used for the other geometries of bonded assemblies presented in the following.



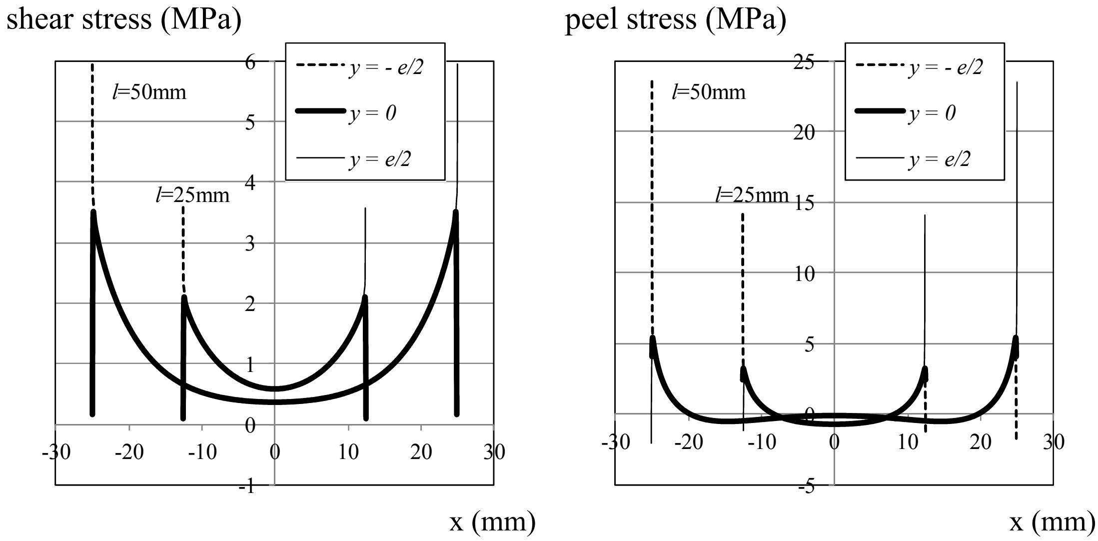
a) geometry of the specimen

b) geometry A: straight edges

c) geometry B: cleaned edges

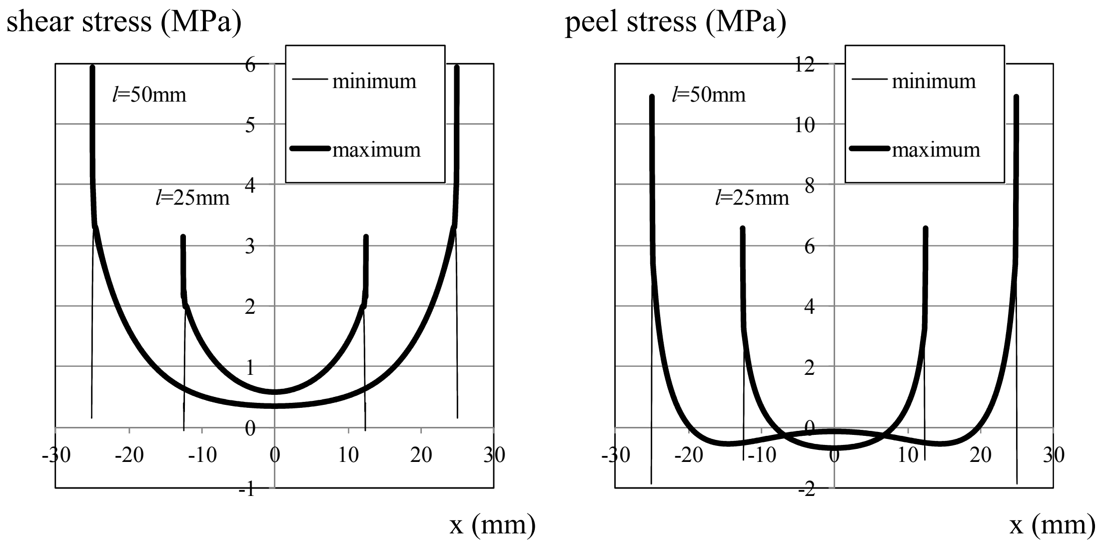
Figure 1: Presentation of geometries A and B of the single lap-shear joint and meshes for an adhesive thickness of e = 0.2 mm (close-up view).

Figure 2 presents, for the geometry A, the stresses in the middle of the adhesive and at the interfaces ($y = -e/2$ is the lower adhesive-substrate interface, $y = 0$ is the middle line of the adhesive and $y = e/2$ is the upper adhesive-substrate interface). It should be noted that there is a large variation in the stress close to the adhesive free edges, close to points C and D (Figure 1). Stress concentrations are mainly observed for the peel stress (*yy* stress component). For such geometry the maximum values of the stresses are reached at the adhesive-substrate interface close to the adhesive free edges, which is often the weakest part of the assembly.



a) shear stress b) peel stress

Figure 2: Shear and peel stresses in the adhesive ($y = -e/2$, 0, $e/2$) with respect to the overlap length for geometry A , for an average shear stress of 1 MPa.



a) shear stress b) peel stress

Figure 3: Minimum and maximum values, in the thickness of the adhesive, of shear and peel stresses with respect to the overlap length for geometry B, for an average shear stress of 1 MPa.

Figure 3 presents the influence of a slight modification of the free edges of the adhesive, using the so-called “cleaned edges”, geometry B. It can be seen that stress concentrations are lower than for geometry A, especially for the peel stress. Moreover, the maximum stress state is not obtained close to the adhesive-substrate interfaces but within the adhesive thickness [5, 15]. Therefore, only the minimum and maximum values of the stress components in the thickness of the adhesive are represented.

Moreover, for the two geometries A and B an increase in the overlap length (l = 25 mm and 50 mm) increases the peak stress close to the adhesive free edges [15].

4 Stress distribution for tubular bonded assemblies

For tubular bonded assemblies, presented in Figure 4, a tube thickness of dr =10 mm was used and two overlap lengths l = 25 mm and 50 mm were studied.

Figure 5 presents, for geometry C, the maximum and minimum values of the shear and peel stresses throughout the adhesive thickness under an axial tensile load with respect to overlap length and for an average radius r = 50 mm. The influence of the geometry of the adhesive free edges on the stress state is similar to that of single lap shear specimens. However, under tensile loads, tubular assemblies are associated with compressive peel stresses, which is an important point for the bonded joint strength. Moreover, the stress state is not symmetrical with respect to the centre of the adhesive, as the radiuses of the two tubes are not equal. Other results can be found in [6].

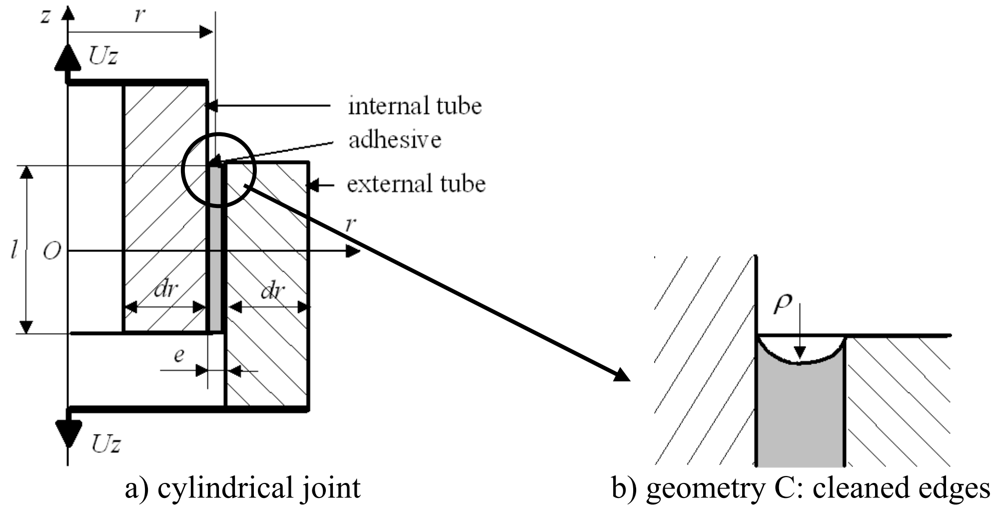
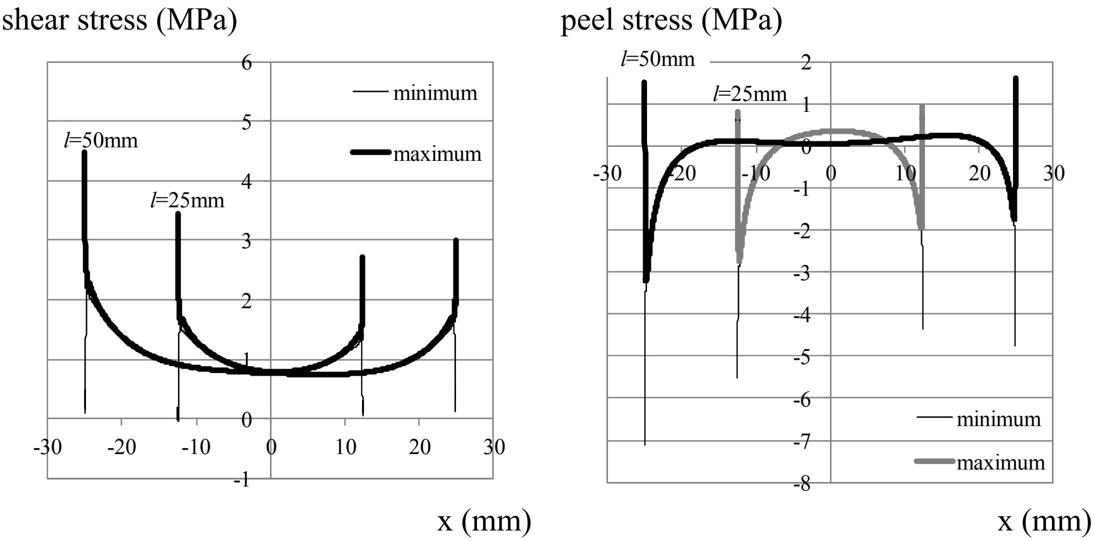


Figure 4: Presentation of the cylindrical joint studied with cleaned edges of the adhesive (close-up view).

5 Comparison between cylindrical and single lap joints

In this section, a comparison between the mechanical behaviour of cylindrical joints and single lap joints is proposed starting from 2D refined finite element computations under elastic assumptions.

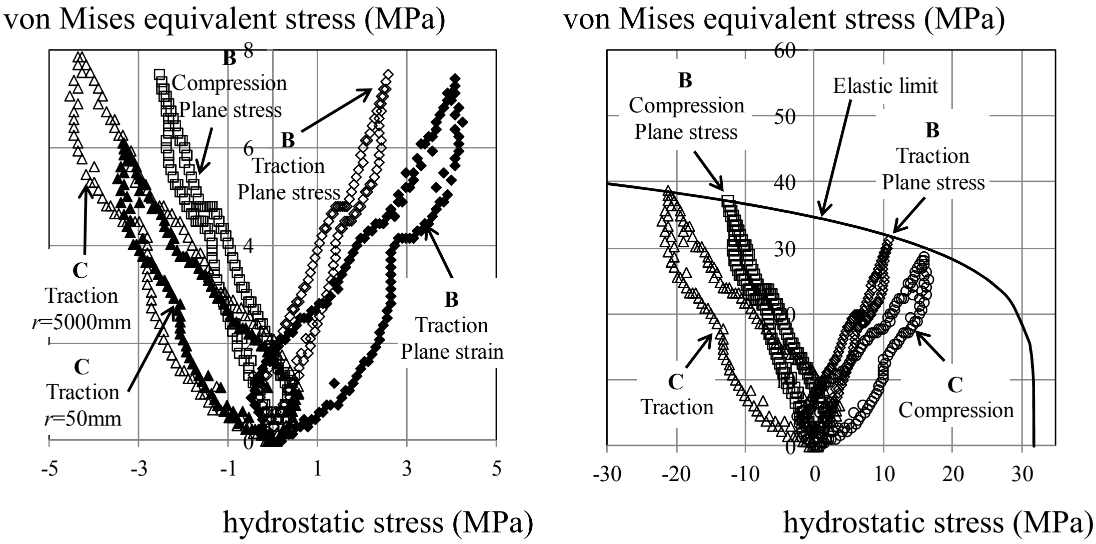
Figure 6 presents, in the hydrostatic stress - von Mises equivalent stress diagram, envelopes of the stresses in the adhesive, for geometries C and D, under traction and compression loads, for an overlap length of 25 mm. The hydrostatic stress and the von Mises equivalent stress are computed for each elements of the finite element mesh, and, only, the envelope of those points is drawn in order to simplify the presentation. For single lap shear specimens the differences between results under plane stress assumption and plane strain assumption can be seen. In order to simplify the comparison, the single lap shear specimen is also loaded under compression load, even if the realisation of such tests is difficult because of the possible buckling of the specimen. The envelopes of the stress states under tensile and compressive loads are symmetrical with respect to the y axis (Figure 6). For tubular assemblies, results are presented for an average radius: r = 50 mm and r = 5000 mm; a slight influence of the average radius r can be noted. Figure 6 presents the stress state in the adhesive for an average shear stress of 1 MPa and for the maximum transmitted load. It is possible to note the beneficial effect of the compressive peeling stress on the maximum value of transmissible load by the two types of tests in association with the elastic limit of the adhesive.



a) shear stress b) peel stress

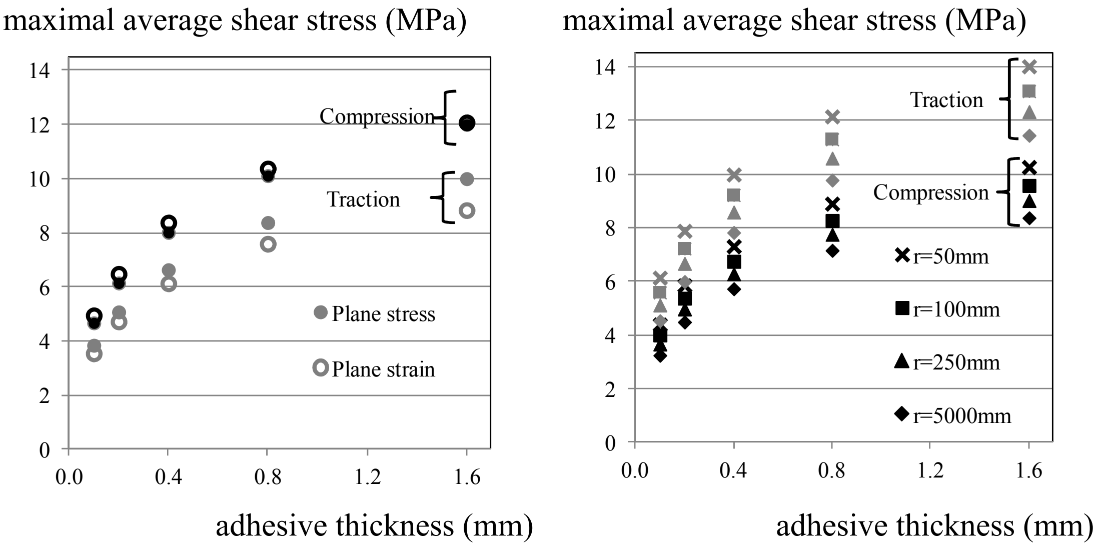
Figure 5: Minimum and maximum values, in the thickness of the adhesive, of shear and peel (radial) stresses with respect to the overlap length for geometry C, for an average shear stress of 1 MPa.

Figure 7 underlines the influence of the average radius of the cylindrical joint on the maximal average shear stress under elastic behaviour. The maximal average shear stress is associated with the maximal transmitted load by the structure and it is numerically determined, for an increasing external load, when a point of the stress envelop reaches the elastic limit defined by equation (1). For a value of the radius r higher than 5000 mm the solution stays almost constant. To carry out the comparisons, results are also presented for single lap joint specimens under the assumptions of plane stress and the plane strain. Moreover, Figure 7 presents the influence of the adhesive thickness on transmitted load (under elastic assumption); for both specimens with cleaned free edges of the adhesive, an increase in the adhesive thickness leads to an increase in the transmitted load for both geometries.



a) average shear stress of 1 MPa b) maximum transmitted load

Figure 6: Stress state in the adhesive with cleaned edges for geometries B and D in the hydrostatic stress - von Mises equivalent stress diagram, for an overlap length of $l = 25$ mm.



a) single lap-shear joint (geometry B) b) cylindrical joint (geometry C)

Figure 7: Influence of the adhesive thickness on the maximal average shear stress for an overlap length of $l = 25$ mm.

These results show that 2D simulations of single lap joint specimens give results which are only close to the results of the cylindrical assemblies with same substrate thicknesses, even for large radius *r*. However, the influence of various possible defects on the experimental mechanical behaviour for single lap shear specimens, or the scatter in the results, is not easy to analyse. Therefore, Figure 7 underlines that experimental results under tensile loads for single lap joint specimens can be used to estimate the maximal transmitted load by a tubular joint under compression loads. But, these values underestimate the maximal transmitted load by a tubular joint under tensile loads, as experimental results under compression loads for single lap joint specimens are generally not available (possible buckling of the specimen). Thus, the use of accurate numerical models can be useful in order to optimize the design of complex bonded assemblies.

6 Numerical results for conical assemblies

The aim of the parametric study presented in this section was to analyse the influence of the geometry of adhesive joints, for cylindrical assemblies loaded in traction and compression.

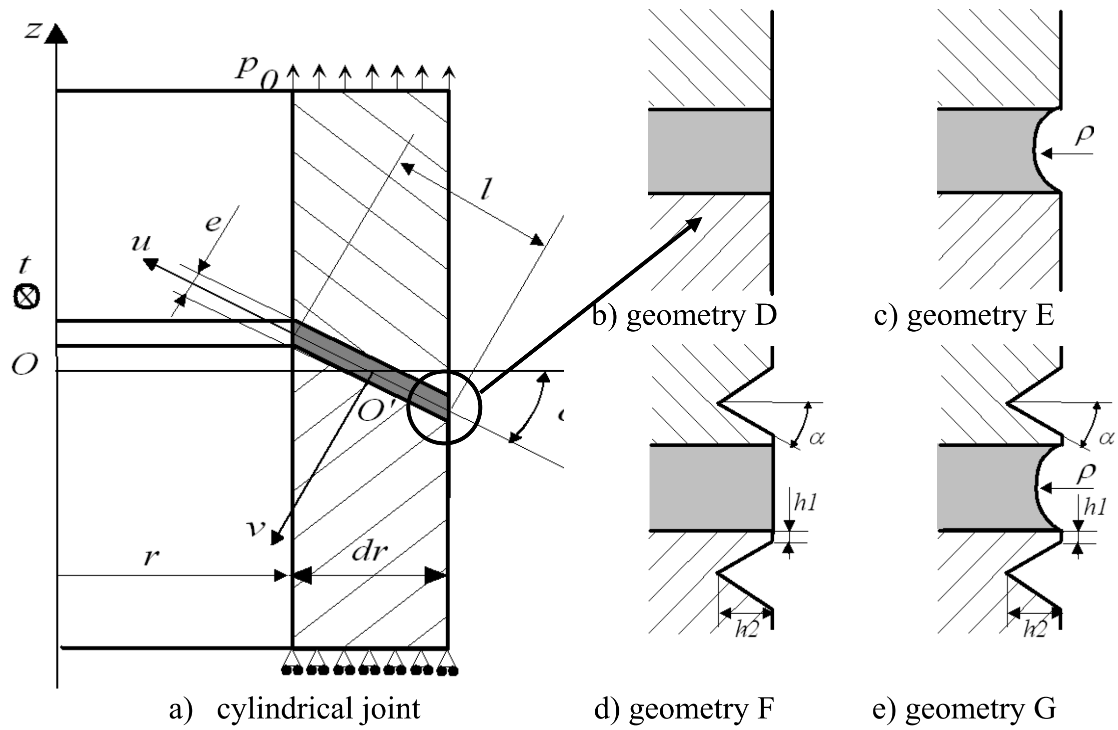


Figure 8: Presentation of the cylindrical joint studied with four local geometries (D, E, F and G) of the adhesive (close-up view presented for $\delta = 0°$).

The use of a conical geometry, for the adhesive, makes it possible to modify the relationship between the traction and shear stresses in the adhesive, according to the geometrical parameters. These types of geometry of bonded joints can be seen as representative of various industrial applications, for instance such types of geometries are used in the design of launchers in a more complex form. The case of conical bonding surfaces for the assembly of tubes with the same diameter is studied in the following.

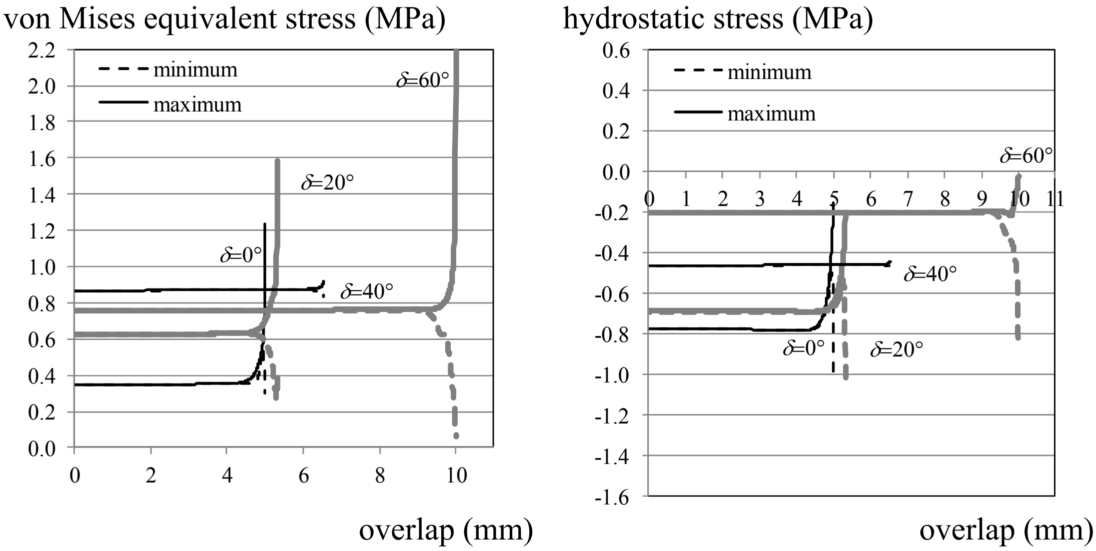
Figure 8 presents the axisymmetrical model studied under axial traction and compression loads. Four geometries of the assembly, near the free edges are proposed; they are presented in Figure 8 for the angle $\delta = 0°$. The geometrical parameters are: $r = 50$ mm, $dr = 10$ mm, $e = 0.2$ mm, $h1 = 0.1$ mm, $h2 = 1$ mm, $\alpha = 30°$, and $\rho = 0.75e$. One of the aims of this study was to analyse the influence of the angle δ on the bonded assembly strength.

For an axisymmetrical model, computations were made using the base (r, t, z) (Figure 8). In order to facilitate the analysis of the loading of the adhesive, the use of the base (u, t, v) is interesting. In this base, the components of the stress tensor are denoted by: $\boldsymbol{\sigma_u}$, $\boldsymbol{\sigma_t}$, $\boldsymbol{\sigma_v}$ and τ_{uv}; where $\boldsymbol{\sigma_v}$ is the peel stress and τ_{uv} is the shear stress. The axial load transmitted by the assembly is such that:

$$F_Z = \int_S \{ (\boldsymbol{\sigma_v} \cos(\delta)) + (\tau_{uv} \sin(\delta)) \} ds \quad (2)$$

where S is, for instance, the average area of the adhesive.

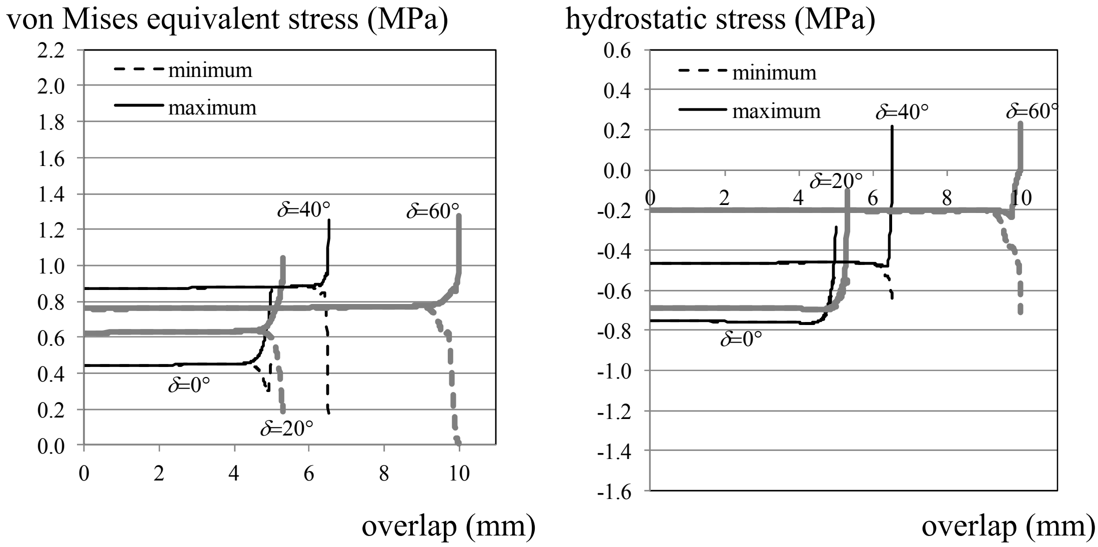
It is important to note that the shear stress (τ_{uv}) and the peel stress ($\boldsymbol{\sigma_v}$) contribute in the definition of the transmitted load. Moreover, in the radial plane (0, r, z), the overlap length also depends on the value of angle δ.



a) von Mises equivalent stress b) hydrostatic stress

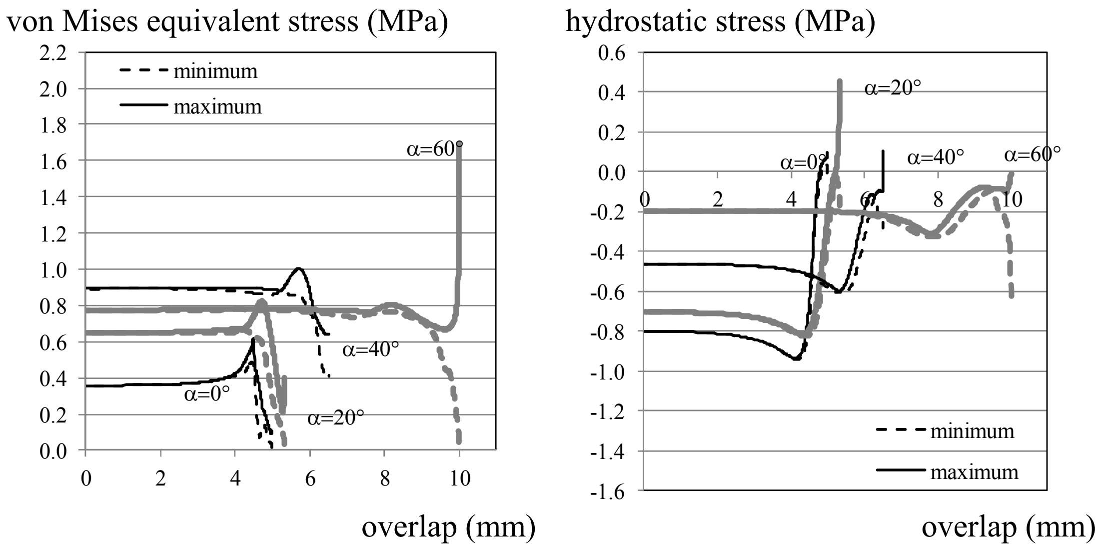
Figure 9: Influence of the angle δ on the minimum and maximum values of the stresses in the thickness of the adhesive for geometry D and for a pressure $p_0 = 1$ MPa.

Figures 9-12 present the minimum and maximum values of the stress components in the thickness of the adhesive for the geometries D, E, F and G for a pressure $p_0 = 1$ MPa. The minimum and maximum of the shear stress, the peel stress, the von Mises equivalent stress and the hydrostatic stress are plotted as a function of the position along the overlap length.



a) von Mises equivalent stress b) hydrostatic stress

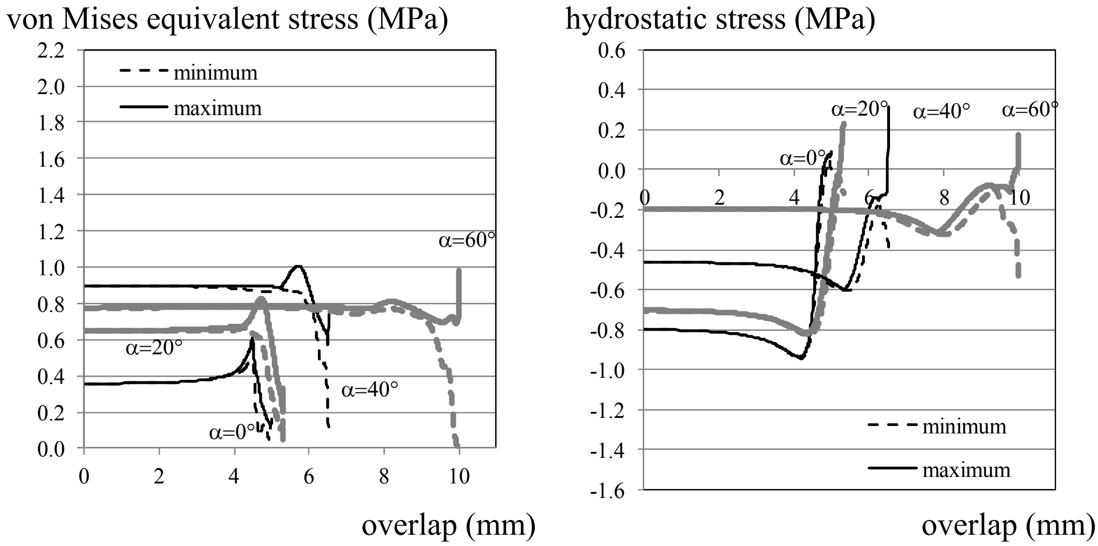
Figure 10: Influence of the angle δ on the minimum and maximum values of the stresses in the thickness of the adhesive for geometry E and for a pressure p_0 = 1 MPa.



a) von Mises equivalent stress b) hydrostatic stress

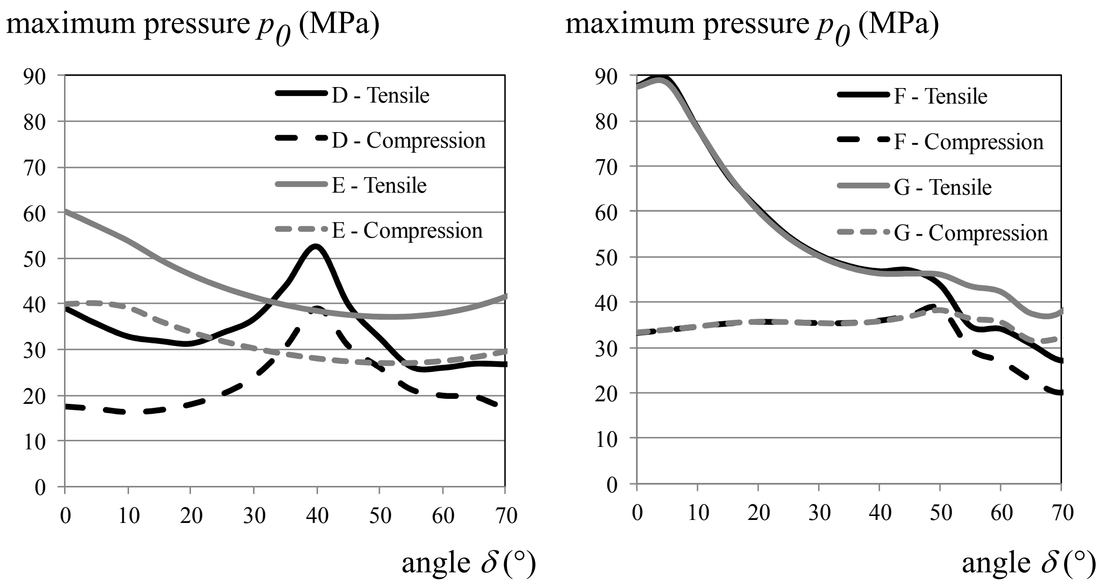
Figure 11: Influence of the angle δ on the minimum and maximum values of the stresses in the thickness of the adhesive for geometry F and for a pressure p_0 = 1 MPa.

In order to facilitate the analysis, the results are presented for half of the overlap length (segment ($0'$, u), Figure 8) knowing that the solution is symmetrical with respect to the point $0'$.



a) von Mises equivalent stress
b) hydrostatic stress

Figure 12: Influence of the angle δ on the minimum and maximum values of the stresses in the thickness of the adhesive for geometry G and for a pressure p_0 = 1 MPa.



a) geometries D and E
b) geometries F and G

Figure 13: Influence of the angle δ on the maximum transmitted load for geometries D, E, F and G.

The levels of the peel and shear stresses depend on the angle δ. If one does not take into account the influence of the edge effects, for $\delta = 0°$ the shear stress is equal to 0 and the tensile stress is equal to 1 in the joint.

For geometry D, an angle δ = 40° makes it possible to strongly limit stress concentrations. For such geometry, associated with quite large stress concentrations more refined meshes can be required in order to obtain a good representation of the stress state [16]. It can be noted that the cleaning of the adhesive free edges reduces stress concentrations as for single lap shear or bonded tubular specimens, except for an angle δ of 40° (geometry E, Figure 10). The use of beaks (geometries F and G) allows to a reduction of the stress concentration, especially for an angle δ lower than about 30°. Figure 13 presents the influence of the angle δ on the maximum transmitted load for geometries D, E, F and G, computed with the proposed model with 40 elements in the adhesive thickness of 0.2 mm. These results underline the importance of the geometry of the assembly close to the adhesive free edge on stress concentrations and the importance of the loading (traction or compression).

7 Conclusions

The objective of this ongoing project is to optimize adhesively bonded assemblies. Those assemblies offer many advantages but analyzing the mechanical behaviour is made difficult, in particular, by the stress singularities due to edge effects, especially close to the free edges of the adhesive. Therefore, it is difficult to obtain a precise dimensioning of adhesively-bonded assemblies, using simplified methods which generally are not able to analyse large stress concentrations [17]. Under elastic assumption of the materials, the use of pressure-dependent elastic limit of the adhesive can provide an accurate analyse of the transmitted load by a bonded assembly. Such models, defined from the hydrostatic and von Mises equivalent stresses, accurately represent the behaviour of various polymers.

This paper has analysed the influence of some geometrical parameters in the case of coaxial assemblies, using 2D axisymmetric finite element results in order to obtain accurate results while limiting the numerical cost with respect to 3D simulations. However, this approach can be seen as a simplified one in order to analyse the influence of more complex loads such as bending loads in a perpendicular direction of the assembly axis. A precise analysis of the stress state in the adhesive, in the case of conical geometries of the bonded area, highlights the fact that it is possible to find geometric parameters which strongly limit stress concentrations, and that the use of specific geometries, close to the adhesive free edges, can also increase the maximum load transmitted by such bonded assemblies.

Those numerical results provide some rules for manufacturing conditions in order to optimise bonded cylindrical joints under axial loadings. Under elastic assumption, the key point is to strongly limit the influence of edge effects in order to increase the transmitted load. It is necessary to analyse the influence of more complex loadings (torsion-tensile, bending), and the influence of the non-linear behaviour of the adhesive.

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