# Analyse de l'effet d'entaille sur la distribution des contraintes tangentielles dans la couche d'un adhésif utilisé pour le collage de deux plaques en Aluminium 2024-T3 = Numerical analysis of the notch effect on tangential stress distribution of the adhesive layer used for bonding two aluminum plate 2024-T3

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# *Numerical analysis of the notch effect on tangential stress distribution of the adhesive layer used for bonding two aluminum plate 2024-T3*

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## Résumé

Dans cette étude, la méthode des éléments finis a été utilisé pour analyser la répartition des contraintes de cisaillement dans la couche d'adhésif utilisée pour assembler deux plaques en en aluminium avec et sans entaille, l'ensemble est soumis à une charge de traction. Les effets des propriétés de l'adhésif ont été mis en évidence. Les résultats montrent que l'effet d'entaille est plus sensible lorsque le bord libre de l'adhésif est près de l'entaille. Cet effet disparaît complètement pour un petit rayon d'entaille. La présence de l'entaille n'a pas d'effet sur la répartition des contraintes de cisaillements dans la couche d'adhésif.

## Abstract

In this study, the finite element method was used to analyze the distribution of the adhesive shear stresses between two plates with and without notch subjected to a tensile load. The effects of the adhesive properties were highlighted. The results show that the notch effect is more sensible when the free edge of the adhesive is near the notch. This effect disappears completely for small radius of the notch. The presence of the notch does not have any effect on the distribution of the adhesive shear stresses.

**Mots Clés :** contrainte de cisaillement, adhésif, entaille, aluminium 2024-T3
**Keywords :** tangential stress, adhesive, notch, aluminium 2024-T3

## 1. Introduction

Adhesively bonded joint is being increasingly used in structural applications, which is justified by its well-known advantages over mechanically fastened joints: fewer sources of stress concentrations, more uniform distribution of load, and better fatigue properties. These characteristics of adhesive joints make them attractive in industries such as aeronautics, automotive and civil engineering. However, several problems related to load transfer between structural composite components are not solved. In fact, the stress concentrations at critical regions such as interfaces adherent–adhesive can be a source of damage due to interfacial shear and transverse normal stresses.

Many models for adhesively bonded joints were developed using single lap-joint geometries. This configuration has been adopted by the American Society of Testing Materials as a standard mechanical test for the determination of shear strength [1].

Single lap joint configurations have been studied for about sixty years and numerous analytical and numerical models have been developed. Volkerson [2] proposed the first approach: in this model, only shear deformation of the adhesive and axial deformation of the adherends is considered. Goland and Reissner [3] were the first to include the effect of the eccentric load path by applying moments at the edges of the joint. Allman [4] produced a model that allows linear variation of the peel stress through the adhesive thickness and gives zero shear stress at the overlap joint edges. Chen and Cheng [5] also formulated a two-dimensional model that results in zero shear stresses at the joint edges.

Numerical methods can be used to analyze models with arbitrary geometries and load conditions. They are suitable for the analysis of structures comprised of different materials. Bigwood and Crocombe [6] used the finite difference method to solve the differential equation that represents the peel and shear stresses in an adhesive layer and developed an elasto-plastic analysis of adhesively bonded joints using the same approach.

The two-dimensional finite element analysis of adhesively bonded joints can be carried out using conventional plane strain or plane stress elements. When this approach is followed, very fine meshes are required, particularly in the adhesive layer, in order to obtain a reasonable accuracy. Wooley and Carver [7] made one of the first finite element analyses of a single lap joint. They used plane stress elements and their results were comparable with those from the Goland and Reissner [3] solution. Harris and Adams [8] developed a geometrically nonlinear finite element that included the elasto-plastic response of adhesive and adherent components. They performed the analysis of a standard single lap joint and of a single lap joint with fillets at the edges of the overlaps.

All the cited studies have in common the use of quadrilateral plane stress or plane strain elements. This type of element requires fine meshes to get adequate accuracy. The stress concentration is minimized due to the larger bonded area, and the stress distribution becomes more uniform in the overlap region. Thus, to ensure the safety of bonded structure, it is necessary to analyze the stress distribution on the adhesive layer. It is known that the presence of stress concentrators such as notch is a common cause of failure of the structure [9]. For a bonded structure, the failure occurs at the free end of the adhesive layer. Many researchers studied the stress distribution in the adhesive layer by analytical model, such as Hart-Smith [10], Du Shen et al. [11], or by numerical method such as Reddy et al. [12] and Richardson et al. [13]. However the most of these studies did not take into account of the effect of the presence of geometrical defects in the structures.

The objective of this work is to apply numerical and experimental study to obtain accurate stress distributions for adhesively bonded joint of aluminum alloys 2024-T3. Joints with similar and different materials are studied numerically and some differences in their mechanical behaviors are discussed. The analysis considers the stress variation across the adhesive length. The results highlight particular stress concentrations in the adhesive layer and the influence of the adhesive properties such as thickness, length, shear modulus and variation of the nature of adherent material in the stress distributions. In the second section, we discussed the influence of the presence of the notch in adhesively bonded joint on the stress distribution in the adhesive layer between two bonded plates using the finite element method. The effects of the adhesive properties and the notch radius on the adhesive stress distribution were highlighted.

## *2. Geometrical model and materials definitions*

Let us consider two aluminum thin plates having dimensions as: high H, width W such as W/H= 0.5. Each plate presents a semi circular lateral notch of radius. The elastic properties of Aluminium 2024-T3 material and the adhesive **epoxy bi-component** structural **ADEKIT** A140 made by AXSON company and contains modified epoxy resin are shown in Table 1. The two plates are bonded with the adhesive as shown in Figure 1. The structure is subject to uniaxial tensile load for stress level lower than □□=140MPa. It is important to notice that all the calculations have been carried out with elastic behaviour for each element (plates and adhesive).

| Plates | Mechanical properties |
| --- | --- |
| **Aluminum** | $R_{0.2}$ *(MPa)* =230MPa<br>$R_m$ *(MPa)*= 452MPa<br>*A*=2,4 %<br>E=68,8GPa<br>ν=0,33 |
| **Adhesive** | E=2690MPa.<br>$G=\frac{E}{2(1+\nu)}=\frac{2,69}{2(1+0,3)}=1GPa$<br>ν=0,3 |

Tab. 1. Mechanical properties of aluminum plates and adhesive ($R_{0.2}$ yield stress, $R_m$ ultimate stress, elongation A%, E and $E_i$ Young Modulus, ν: Poisson coefficient).

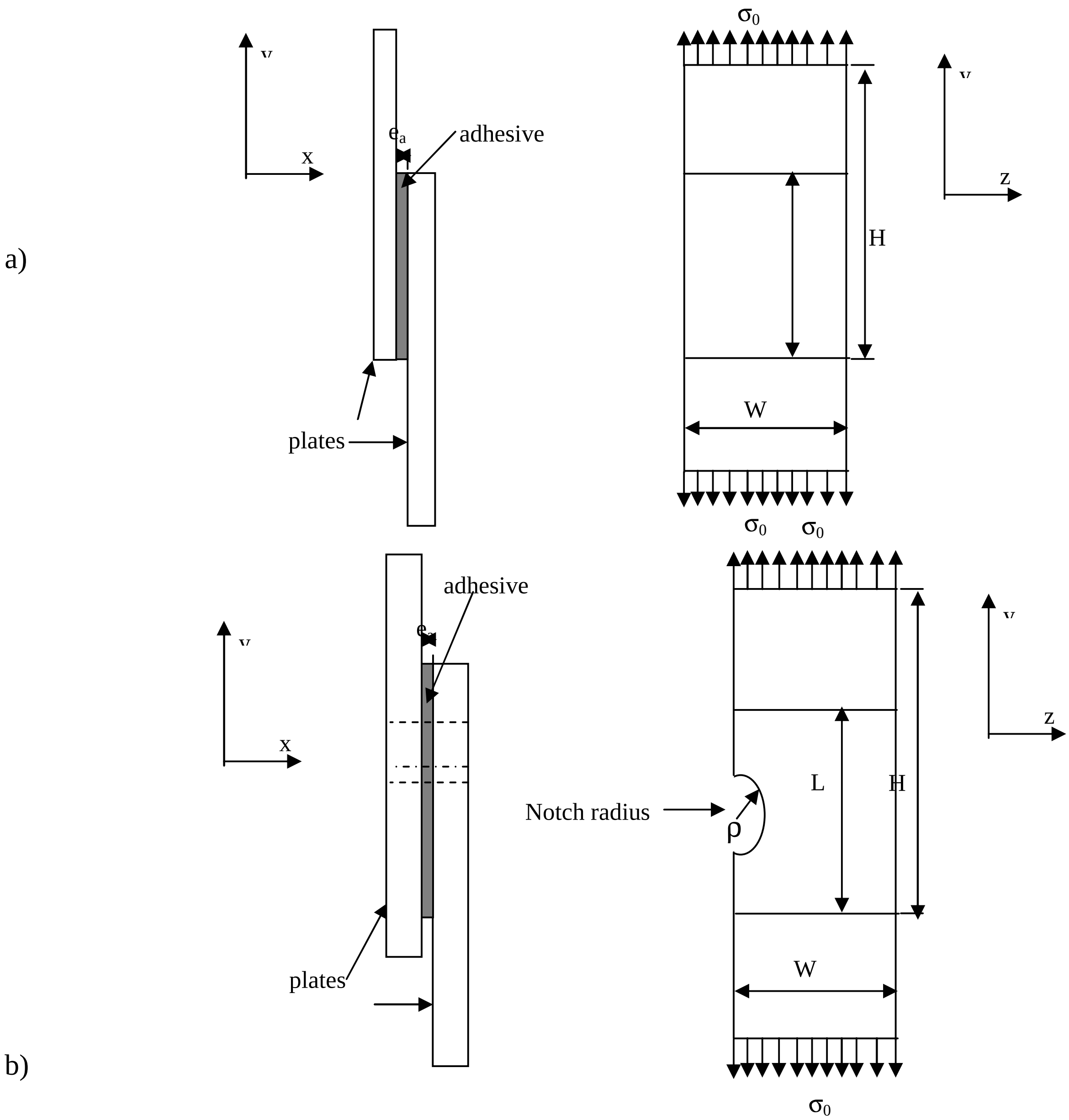



Fig.1: Geometrical model of the bonded structure, a) no notch, b) with notch.

## 3. Finite elements modeling

Finite element analysis of the configurations of Figure 2 is done using the finite element code Franc2D/L developed at Kansas University [14]. A layered structure is actually a three dimensional structure. A three-dimensional finite element or mathematical modeling of such a structure would involve several degrees of complexity. In this study, simplifying assumptions are made which still allow us to capture the essential features of the response. These assumptions include:

- each layer is considered as an individual two-dimensional structure under a state of plane-stress,
- individual layers can be connected with adhesive bonds,
- it is assumed that the adhesive layer is homogeneous, linear elastic and isotropic,
- the adhesive is assumed to deform only in shear and this deformation is uniform throughout the adhesive thickness,
- the surface shear transmitted through the adhesive is assumed to act as surface traction on the substrates.

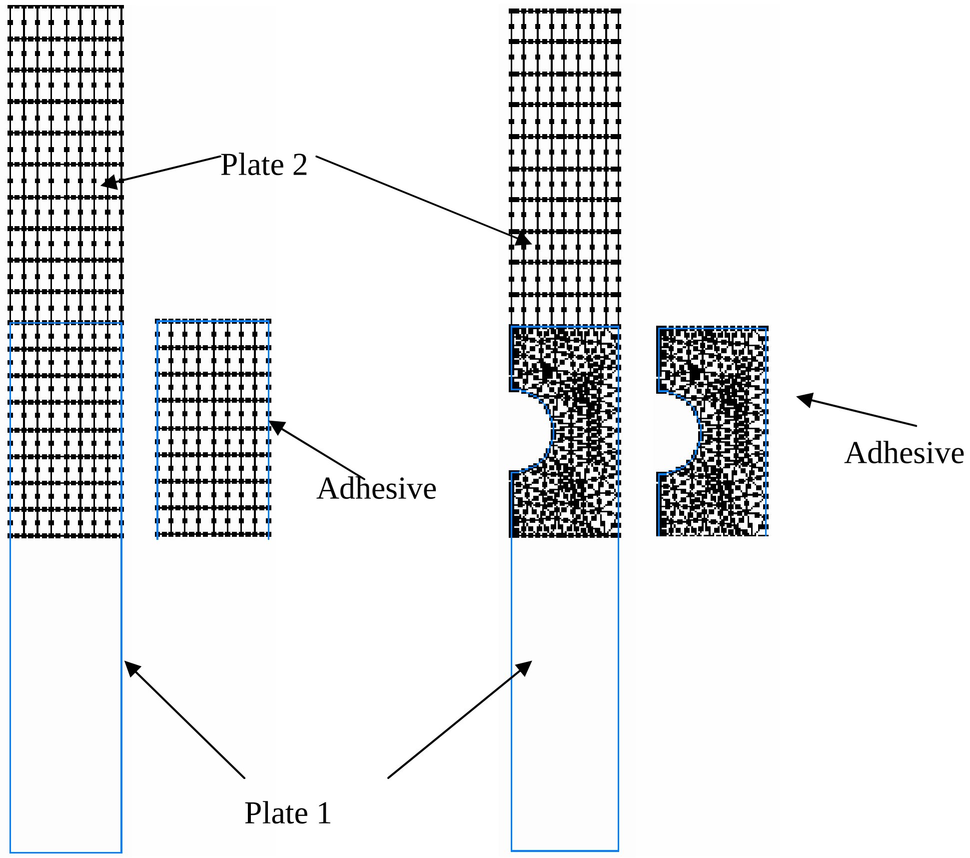



Fig. 2: Typical mesh model of one aluminum plate, a) no notch, b) with notch.

The shear stress in the adhesive is given by:

$$\tau = \frac{G_a}{e_a}(u_1 - u_2) \qquad (eq.\ 1)$$

where $u_1$ and $u_2$ are the displacements in the plates 1 and 2 respectively, $G_a$ and $e_a$ are respectively the shear modulus and the thickness of the adhesive.

The adhesive forces are obtained by using the adhesive shear stresses as surface tensions on the layer and integrating. Since the surface tensions are proportional to the relative displacement of the two layers, the adhesive force can be expressed in term of nodal displacements of the top and bottom layer. This gives a stiffness matrix for the adhesive elements. The total structure is meshed using standard eight nodded serendipity elements with quadratic shape functions. Figure 2 shows typical mesh model of one bonded layer.

## 4. Analysis and results

In this section we examine the effect of the adhesive thickness and length first; then we discuss the consequence of a modification of the adherent materials.

### *4-1. Influence of the adhesive thickness*

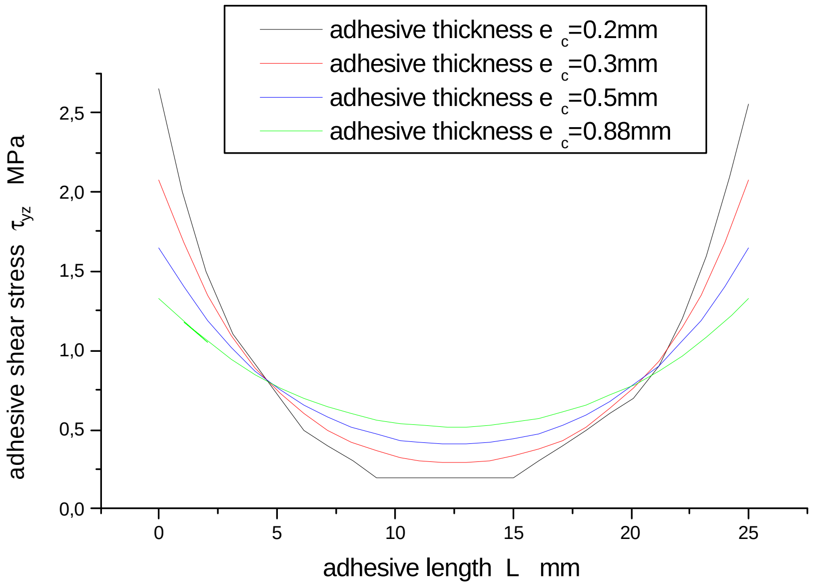



Fig. 3: Distribution of adhesive shear stress according to the length of adhesive for different adhesive thicknesses (without notch and with aluminum plates, applied stress =35 MPa).

Figure 3 shows the variation of the adhesive shear stress in the yz plan for different values of the adhesive thickness. It can be noted that an increase of the adhesive thickness involves a reduction in the shear stress $\tau_{yz}$, i.e. for a rather significant thickness, the adhesive becomes very resistant and behaves like a third material. In addition, the rupture becomes increasingly adhesive when the adhesive thickness increases, as already reported [15]. This result leads to the conclusion that the choice of the adhesive thickness must be optimized. Many researches showed that the influence of the adhesive thickness varies according to assembled materials and the passage by an optimum cannot always be highlighted. Generally, one locates this optimal thickness between 0.1 and 0.2 mm.
These phenomena were explained by Giraud [16] who, using a modeling by finite element, showed that there is an increase in the excess stresses and in particular in normal constraints ($\sigma_{zz}$) when the film thickness increases. It also highlights the existence of an optimal thickness (around 0.2mm) where the excess stresses are lower.

### 4-2. Influence of the adhesive length

Figure 4 shows the distribution of the adhesive shear stresses according to the adhesive length (according y axis) for different values of the adhesive length. It can be noticed that the shear stresses decrease if the length of the adhesive increases. This reduction is observed only on the edges of the adhesive. On the other hand, in the center of the adhesive, the reduction of the shear stress value is valid only for length which is close to the width of the plate. If the length becomes significant, the value of the shear stresses tends towards zero. This phenomenon is well explained in the Figure 5, where the distribution of the adhesive

shear stresses is represented *versus* the length of the adhesive for different applied loads, for two adhesive lengths a) $L_c$ = 25mm (figure 5a), b) $L_c$ = 50mm (figure 5b).
For $L_c$ = 25mm, the increase of the applied load increased the shear stresses according to all the lengths of adhesive with varied level. The maximum is observed on the edge of adhesive and the center of adhesive undergoes an increase in the shear stresses but with a light variation. On the other hand, for a length $L_c$ = 50 mm, the edges of the adhesive undergo an increase in the shear stresses but the center of adhesive do not and the value remains null for different values of the applied load.

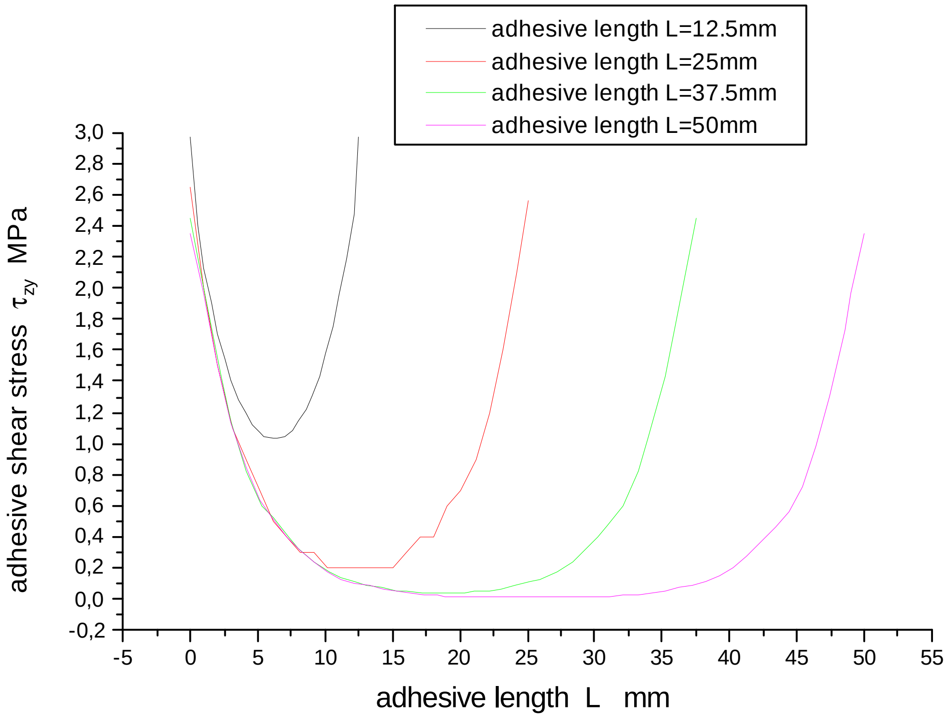
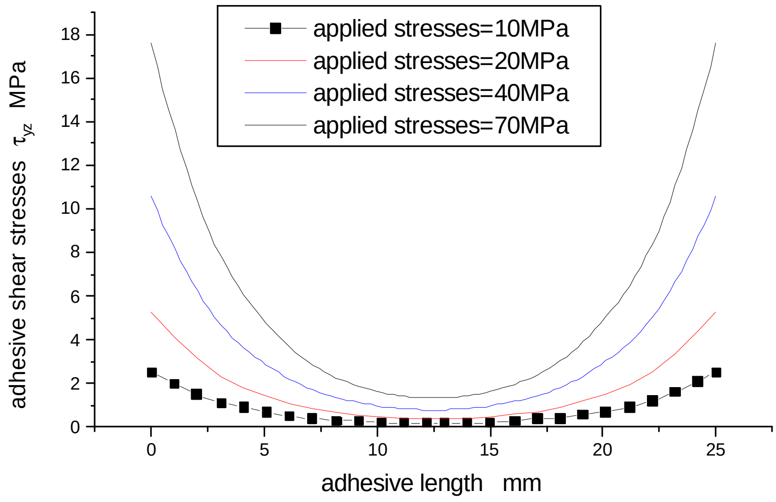
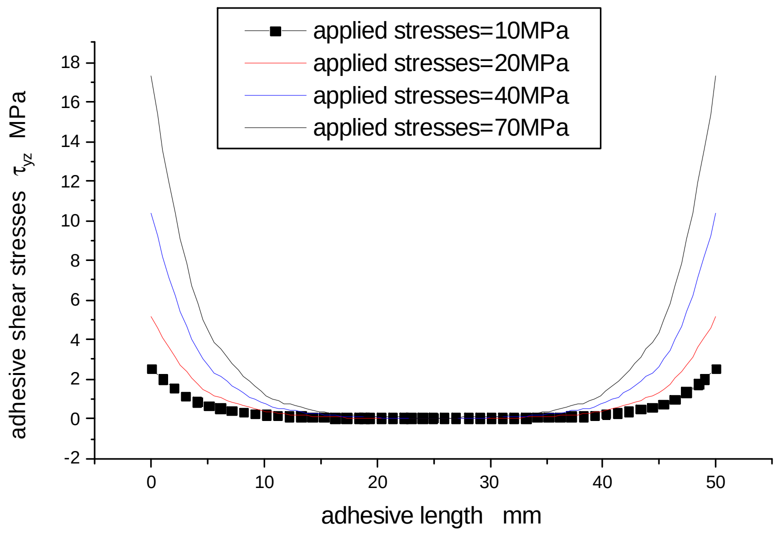



Fig. 4: Adhesive shear stress distribution according to the different adhesive lengths for aluminum plates without notch (adhesive thickness=0.2mm, applied stress =35 MPa).




a)




b)

Fig. 5: Variation of the adhesive shear stresses *versus* the length of the adhesive for different values of the applied stresses. a) Adhesive length=25mm. b) Adhesive length =50mm. (aluminum plates without notch)

### *4-3. Bonded aluminum plates with notch*

In this section, one tried to see the influence of the presence of a semi circular notch on the behavior of a single lap bonded aluminum/aluminum .

### *4-4-1. Adhesive Shear stress level*

The figures 6 and 7 illustrates the levels of the shear stresses in the adhesive layer for a various radius of notch $\rho/2$=0.L, 0.1L, 0.3L and 0.4L. One notices that the distribution of the shear stresses is the same one for the various cases (the maximum stresses are located on the free edge of the adhesive layer). The central zone of adhesive remains inactive for adhesive lengths larger or equal to the width of the plate (figure 6.b and c). However, if the covering length is smaller than the width of the plate, the central zone of adhesive becomes active but always with values of adhesive shear stresses less than those located on the free edges (figure 6a). The presence of the notch leads to a distribution of the shear stress values and the central zone remains always inactive for small radius of notch. However, this zone becomes increasingly active if the diameter of the notch becomes rather significant compared with the adhesive length (figure 7). These behaviors are mainly due to the fact that on the levels of the free edges, displacements on the two sides of the adhesive layer (in the substrates) are maximum in absolute values. The effort in the adhesive layer is

proportional to the variation of displacement between plate 1 and plate 2 ($u_1$-$u_2$). The rupture of the adhesive occurs mainly on the levels of the free edges. It should also be noted that there is no stress concentration at the level of notch in the adhesive layer, contrary to the normal stresses in the plate. It can then be stated that the presence of the notch does not have any effect on the distribution of the adhesive shear stresses. However, it is noted that the variation of the notch radius influences the value of the shear stresses in the adhesive layer.
The increase of the adhesive length increases the adhesive shear stresses when the notch radius is rather significant (figure 8).

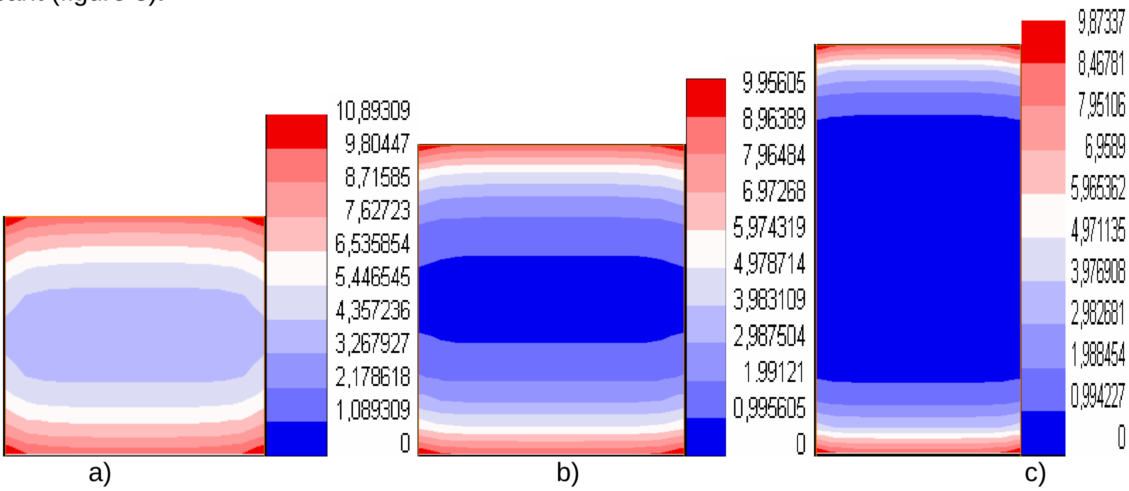



Fig. 6: Contour of the levels of shear adhesive stresses for different adhesive length
a) L=12.5mm, b) L=25mm and c) L=50mm (applied load is 35MPa, adhesive thickness=0.2mm, width of plate is 25mm)

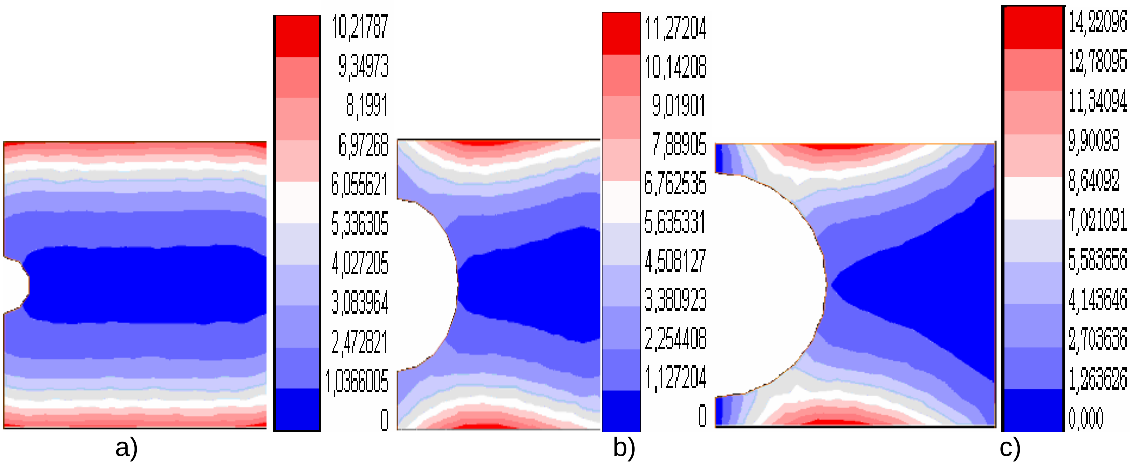



Fig. 7. Contour of the levels of shear adhesive stresses for different notch radius
a) ρ=5mm, b)ρ =15mm and c)ρ =20mm ( applied load is 35MPa, adhesive thickness=0.2mm, adhesive length L=25mm width of plate is 25mm)

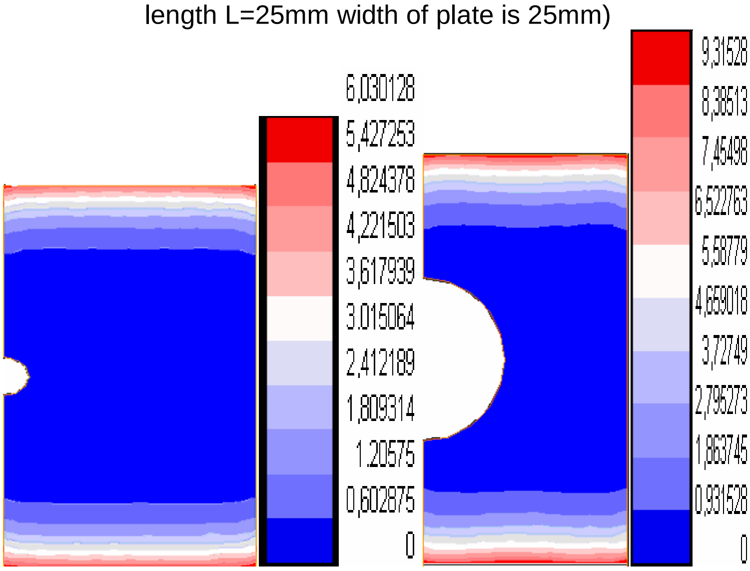
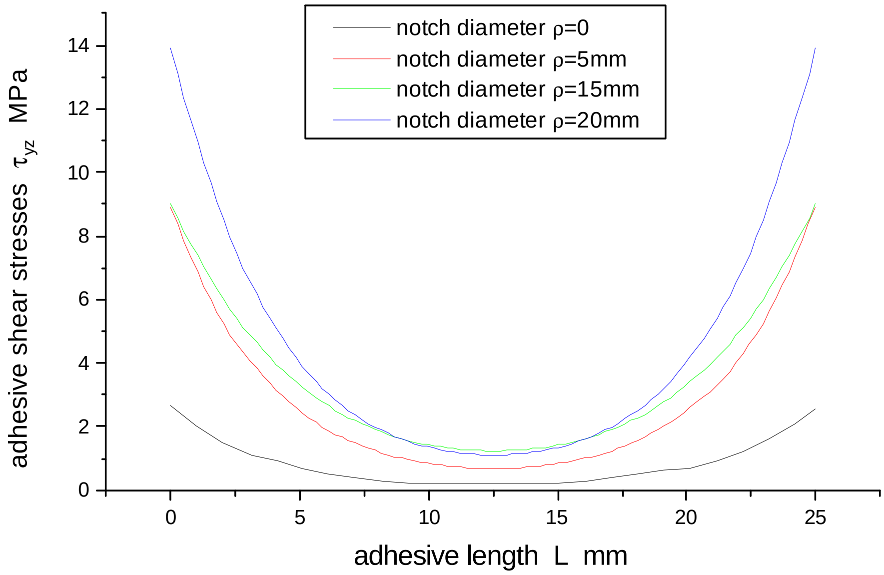


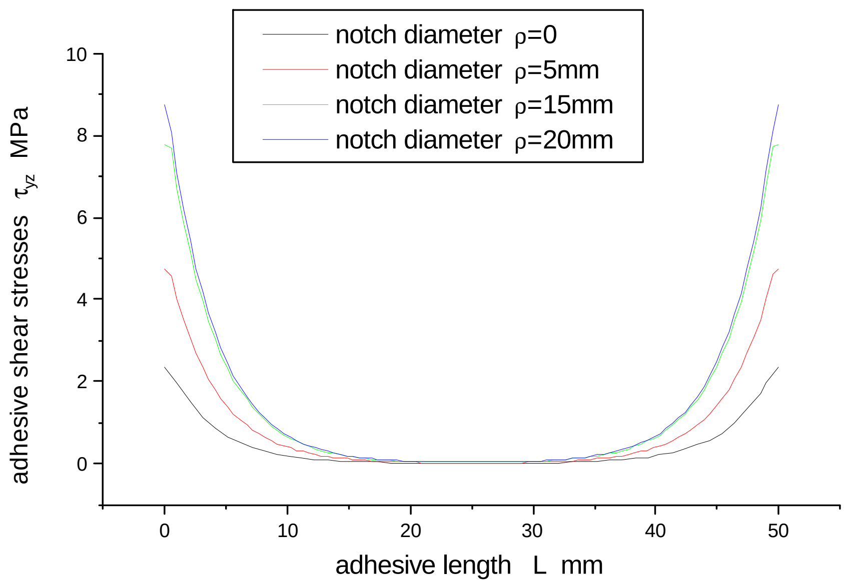
Fig. 8. Contour of the levels of shear adhesive stresses for different notch radius
a) ρ=5mm, and b)ρ =20mm ( applied load is 35MPa, adhesive thickness=0.2mm, adhesive length L=50mm width of plate is 25mm)

### *4-4-2. Adhesive shear stress distribution*

Figure 9 shows the distribution of the adhesive shear stresses along the adhesive length for different notch diameters, for two cases of adhesive length a)L=25mm and b)L=50mm. The distribution of the adhesive length is the same for both different adhesive lengths. The maximum of the adhesive shear stresses is observed on the levels of the free edges and the center of adhesive is inactive in the case where the length of adhesive is rather significant (figure 9b). The presence of a notch increased the values of the tangential shear stresses and the increase in the diameter of the notch increased the value of the tangential stresses. Moreover, the increase of the adhesive length decreases the shear stresses in the layer of the adhesive.




a)

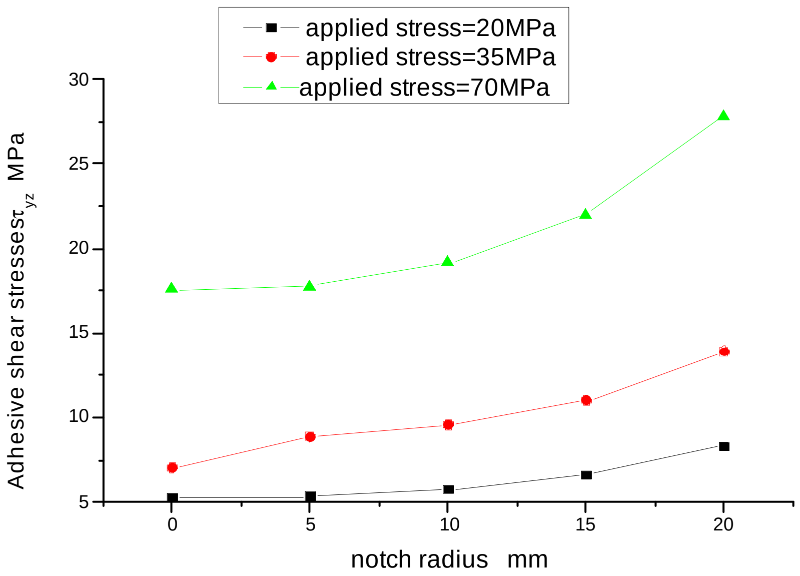



b)

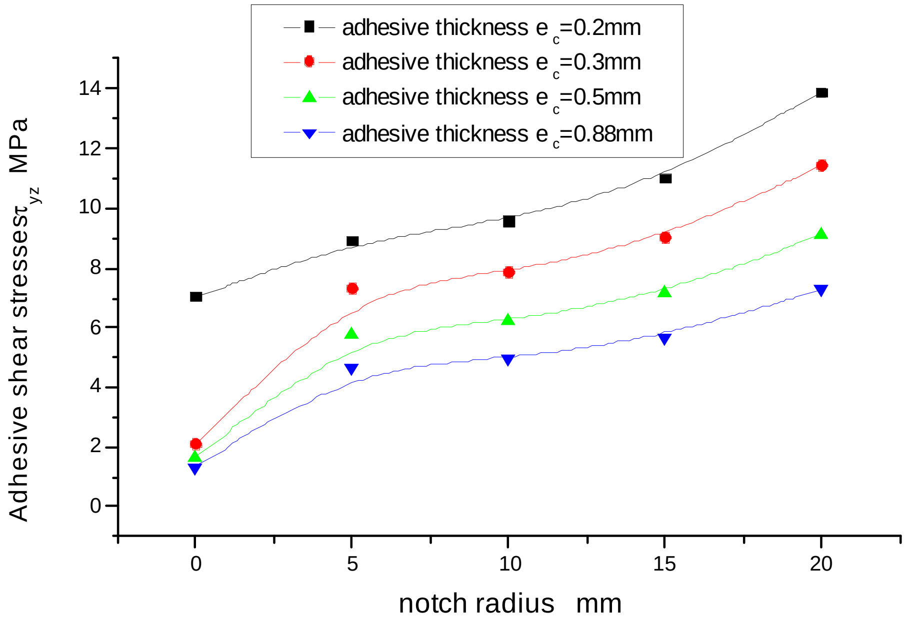
Fig. 9. Distributions of adhesive shear stresses along the adhesive length for different notch diameters; a) adhesive length L=25mm, b) adhesive length L=50mm.( applied stress =35 MPa).

***4-4-3. Variation of shear stresses versus different parameters (adhesive thickness, applied load, adhesive length).***

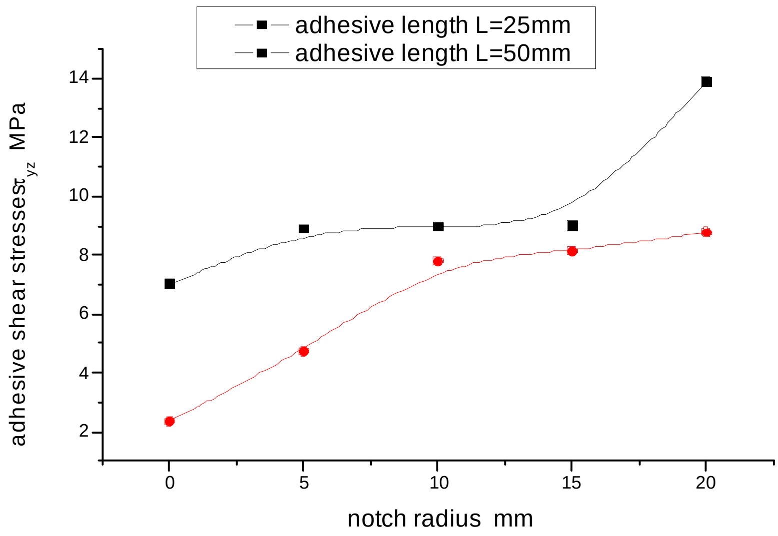
Figure 10 represents the variation of the adhesive shear stresses according to the diameter of the notch for different parameters. A higher load (figure 10a) increases the shear stresses in the adhesive layer. For rather significant applied stresses, (equivalent to 1/3 of the yield stress of aluminum), it is possible to have a value of shear stresses of rupture in the adhesive layer.
The reduction of the adhesive thickness increases the shear stresses (figure10b). The resistance of adhesion is then more brittle for low thickness. It is preferable to increase the adhesive thickness in order to improve the adhesion resistance. However, a very high thickness can be fatal for the assembly. Indeed, with a very thick layer, the adhesive behaves as a third material with very weak mechanical properties. The thickness of the adhesive must thus be optimized to reinforce adhesion and to prevent the behavior of the adhesive as a material with a weak mechanical resistance.
It is noted that the effect of the notch radius on the distribution of the adhesive shear stresses is negligible for ρ/L ratio less then 0.5. The maximum stress in the adhesive is not affected by the variation of the notch radius when the adhesive length is equal to adhesive width. This behavior is confirmed by the figure 10c (L=25mm) where the variation of the maximum adhesive stresses is plotted versus to the notch radius. For ρ/L greater than 0.5, the maximum stress increases considerably. This behavior can be explained by the fact that for great value of the notch radius, the notch is very close to the free edge of the adhesive. This interaction increases displacements in the substrate, which increases the maximum stresses in the adhesive layer. One can thus conclude that when the free edge of the adhesive is near the notch, there is an effect of interaction between the presence of the notch and the free edge of the adhesive. These shear stresses are reduced when the adhesive length increases.
The increase of the adhesive length initially decreases the average shear stress and increases the total resistance of the bonded structures. But beyond a limit value, the phenomenon of stress concentration at the ends become dominating (with a minimum in the center of very weak adhesive length) thus determining a maximum resistance and the optimal length of the joint.




a)




b)




c)

Fig. 10. Variation of adhesive shear stresses according to the notch diameter with the effect of different parameter a) applied load, b) adhesive thickness (applied stress =35 MPa), c) adhesive length (applied stress =35 MPa).

***5-Conclusions***

The results obtained in this study allowed us to deduce the following conclusions:
- the increase of thickness of adhesive involves a reduction in the shear stress;
- the shear stresses decreases if the length of the adhesive increases;
- the reduction in the value of the shear stress at the center of the adhesive layer is valid only for lengths which are close to the width of the plate;
- if the length becomes significant, the value of the shear stresses tends towards zero, only the edge of the adhesive which undergoes the applied load;
- the central zone of adhesive remains inactive for adhesive lengths larger or equal to the width of the plate;
- on the other hand, if the length of covering is smaller than the width of the plate, the central zone of adhesive becomes active but always with values of adhesive shear stresses less than those located on the free edges;
- the presence of the notch does not have any effect on the distribution of the adhesive shear stresses. However, it is noted that the variation of the notch radius influences the value of the shear stresses in the adhesive layer;
- the increase of the adhesive length increases the adhesive shear stresses when the notch radius is rather significant;
- the increase in the load (figure 1a), makes well increase the shear stresses in the layer of adhesive;
- for rather significant applied stresses, (equivalent to 1/3 of the yield stress of aluminium), one will be able to have a value of shear stresses of rupture in the layer of the adhesive one. Therefore if one knows the applied stress, it is preferable to optimize the length of adhesive and its thickness to have shear stresses less raised in the layer of the adhesive;
- the reduction of the adhesive thickness increases the shear stresses;
- the thickness of the adhesive must thus be optimized to reinforce adhesion and to prevent -the behaviour of the adhesive as a material with a weak mechanical resistance. This behaviour can be explained by the fact that for great value of the notch radius, the notch is very close to the free edge of the adhesive;
- the increase of adhesive length makes lower initially the average shear stress and makes increase total resistance of the bonded structures;
- the notch effect on the distribution of the adhesive stresses is negligible if the free edge of the adhesive layer is far from the notch;
- in order to reduce the notch effect, the length of the adhesive layer must be more important than the notch radius;

In further work, the approach is extended in an elasto-plastic calculation to evaluate the consequence of non-linear adhesive plasticity on stress-strain field distribution.

## 6 - References

**[1]** Raul H. Andruet. 'Special 2-D and 3-D Geometrically Nonlinear Finite Elements for Analysis of Adhesively Bonded Joints'. Doctor of Philosophy in Engineering Science and Mechanics. Faculty of the Virginia Polytechnic Institute. 1998.
**[2]** Volkerson, O.. 'Die Nietkraft Verteilung in Zugbeanspruchten mit Konstanten Laschenquerschritten', *Luftfahrtforschung* **15**, 41-47 1938.
**[3]** Goland, M. and Reissner, E.. 'The stresses in cemented joints', *J. Applied Mechanics* **11**, A17-A27, 1944.
**[4]** Allman, D. J. 'A theory for elastic stresses in adhesive bonded lap joints', *Quaterly J. Mechanics and Applied Mathematics* **30**, 415-436, 1977.
**[5]** Chen, D. and Cheng, S. 'An Analysis of Adhesive-Bonded Single-Lap Joints', *J. of Applied Mechanics* **50**, 109-115, 1983.
**[6]** Bigwood, D. A. and Crocombe, A. D. 'Non-linear adhesive bonded joint design analysis.' *Int. J. Adhesion and Adhesives* **10**, 31-41, 1990.
**[7]** Wooley, G. R. and Carver, D. R. 'Stress concentration factors for bonded lap joint', *J. Aircraft* **8**, 817-820, 1971.
**[8]** Harris, J. A. and Adams, R. D. 'Strength prediction of bonded single lap joints by non-linear finite element methods', *Int. J. Adhesion and Adhesives*, 65-78, 1984
**[9]** Glinka G. *Engineering Fracture Mechanics* 1985; 22(3), 485-500.
**[10]** Hart-Smith L J. Adhesive-bonded single-lap joints. NASA CR-112236, 1973
**[11]** Du Shun C., *J Appl Mech*;50, 109-15, 1983.
**[12]** Reddy JN, Roy S., *Int J Non linear Mech*., 23(2), 97-112, 1998.
**[13]** Richardson G, Grocombe Ad, Smith PA., *Int J Adhesion and Adhesives,* 13(3), 193-200, 1993.
**[14]** Swenson D, James M, FRANC2D/L: A crack propagation simulator for plane layered Structures. Version 1.4 User's guide, 1998

**[15]** A. Roy. “Comportement mécanique en sollicitation monotone et cyclique d’assemblage collés composite/composite et composite/acier”, Thèse, Université de Poitiers, 1994.
**[16]** Giraud « Quelle épaisseur de colle choisir pour un assemblage collé ? », Matériaux et techniques, juin-juillet, 1980.