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A new analytical approach for optimization design of adhesively bonded single-lap joint

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Abstract. In this study the three-dimensional nonlinear finite element method was used to analyze the stresses distribution in the adhesive layer used to joint two Aluminum 2024-T3 adherends. We consider in this study the effect of different parameters witch directly affect the values of different stresses. The experimental design method is used to investigate the effects of geometrical parameters of the single lap joint in order to achieve an optimization of the assembly with simple lap joint. As a result, it can be said that both the geometrical modifications of the adhesive and adherends edge have presented a significant effect at the overlap edge thereby causing a decrease in peel and shear stresses. In addition, an analytical model is also given to predict in a simple but effective way the joint strength and its dependence on the geometrical parameters. This approach can help the designers to improve the quality and the durability of the structural adhesive joints.

Keywords: single lap joint; finite element analysis; stresses distribution; experimental design method

1. Introduction

Adhesively bonded joints are preferred due to their advantages such as formation of uniform stress distributions, ability to join different materials, high fatigue resistance and impermeability (Grant *et al.* 2009, Higgins *et al.* 2000). Different bonding configurations can be used, with differences in the stress fields and strength, but the single-lap joint is the most commonly because of the ease of fabrication.

On the other hand, the main handicap of this joining technique is the still significant concentration of stresses at the overlap edges owing to the gradual load transfer between adherends, and also the adherends rotation in the presence of asymmetric loads (Pinto *et al.* 2014). The reduction of stress concentrations along the edges of the adhesive is important to prevent premature failure of the bonded joint. However, the determination of the stress and strain field in the assemblies presents difficulties because of the complex geometry and the various properties of the materials to be assembled. One fundamental factor that affects the mechanical strength of adhesively bonded

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joints is the peel stresses that form at free edges of the overlap region. By reducing these stresses, which cause damage to the joint, the strength of the joint is increased, resulting in the greater load carrying capacity of the joint. In the literature, several different methods to eliminate these stresses are presented (Kumar *et al.* 2010, da Silva *et al.* 2007, 2009, Sun *et al.* 2001, 2008, Akpinar *et al.* 2014, Gang and Chun 2014, Madani *et al.* 2015, 2013, 2009). One of these methods exist to reduce the stress distributions in the adhesively bonded joints is the geometric modifications of SLJ, they are more widespread in the literature and, within this scope, adhesive fillets are a possibility to reduce stress concentrations at the overlap edges (Adams *et al.* 1974, Hildebrand 1994, Tsai *et al.* 1995, Campilho *et al.* 2008, 2009). Fillets act by eliminating the stress singularities that exist in the sharp corners of bonded joints and preventing premature failure, especially for brittle adhesives. Tsai and Morton (1995) addressed the influence of fillets experimentally and numerically using graphite-epoxy single-lap joints under tension. The Moiré Interferometer Method was used to extract shear strains near the fillet. It was concluded that a fillet effectively reduces shear strains, and peel and shear peak stresses near the fillet, subsequently increasing the joint strength. Adams *et al.* (1986) presented a theoretical and experimental study for the tensile strength of hybrid double-lap joints, with carbon-epoxy/steel adherends. A few geometric techniques were considered to level stresses at the overlap, such as fillets at the adherend edges or chamfers in the inner and outer adherends sides. Based on the obtained results, the best solution was the combination of an inside taper in the outer adherends with an adhesive fillet at the overlap edges. Fillet angles of 17°, 30° and 45° were tested, and the 30° fillet gave the best results. By comparing the experiments with the theoretical predictions, an accurate agreement was also found. In additional, the effect of the change in the geometry of the adherend corners on the stress distribution in SLJs and, therefore, on the joint strength has been studied numerically and experimentally by Zhao *et al.* (2011). In their study, a various degrees of rounding were studied and two different types of adhesives were used: one very brittle and another with a large plastic deformation. Experimental results on the strength of joints with different degrees of rounding were presented. For joints bonded with brittle adhesives, the effect of the rounded adherend corners is larger than that with ductile adhesives. Akpinar *et al.* (2014) presented an experimental and numerical analysis study of the strength of the adhesively bonded step-lap joints for different step numbers. The authors showed that in the SLJ geometries, one-step lap geometry reduces the stress concentration developing at the edges of the overlap area while the highest decrease occurred in the three-step lap geometry. Additionally, the amount of reduction in the stress values supports the increase in the experimentally obtained load carrying capacity of the joint. Karachalos *et al.* (2013), gives a new insight and more details description of the mechanisms of failure associated with adherend yielding. A variety of material (adhesive and adherend plasticity) and geometric (overlap length, adherend thickness) parameters have been investigated in their study. They proposed a simple failure methodology in order to predict the strength of single lap joint.

In this study, the effect of using different parameters geometrical of single-lap joint (SLJ) and subjected to tensile loading was investigated numerically. Stress analyses in the SLJ were performed non-linear finite element method by considering the geometrical non-linearity and the material non-linearities of the adhesive (Adekit A140) and adherend (AA2024-T3). Several parameters geometrical were taken into consideration namely, the effect of the fillet angle of the adhesive, the beveling angle of the adherend, the thickness of the adhesive and the adherend. The experimental design method is used to give an understanding on how modifications in the parameters geometrical can influence the joint performance. In additional, this method allows to define best geometric shape of SLJ for low distribution of stresses.

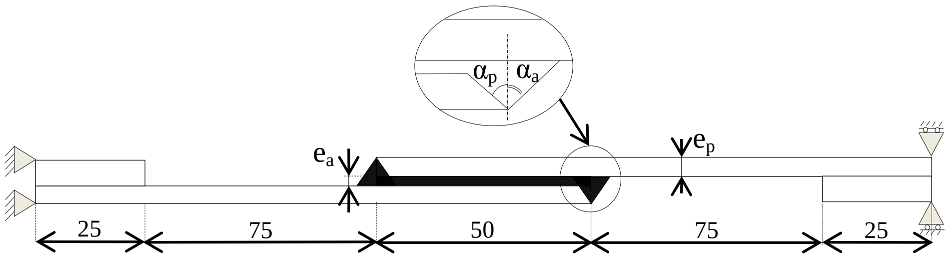


Fig. 1 Specimen geometry (dimensions in mm)

Table 1 Dimensions of the bonded joint

Dimension	Material	
	Aluminum	Adhesive
Length (mm)	150	50
Width (mm)	25	25
Thickness (mm)	e_p=[2-3-4]	e_a=[0.1-0.2-0.3]
α_a(dgrs)	[0-30-60]	-
α_p(dgrs)	-	[0-30-60]

2. Geometric model and mechanical properties

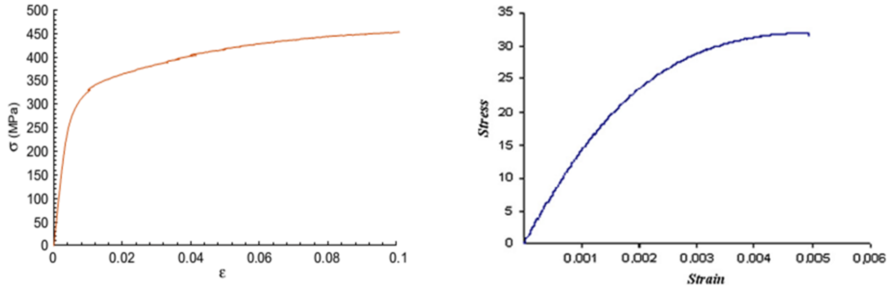
A preliminary study was conducted by Madani *et al.* (2013) where they showed that the modification of the edges of the adherend and/or the adhesive causes a reduction of stresses at the edges of the adhesive. They introduced several geometric models to study the influence of the modification of edges of the joint on the reductions of the stresses.

In this study, the analysis of the effect of various parameters on the shear and peel stress distribution in the adhesive layer has been studied numerically. The analysis of the influence of these parameters was performed by the method of experimental design.

The geometrical model used for this study is shown in Fig. 1. The dimensions are shown in Table 1.With as such variable fillet angle of adhesive “α_a“ and beveling angle of the adherend “α_p”. The tensile tests on the 2024-T3 aluminum adherend and the adhesive ADEKIT A140 allow drawing the characteristics curves shown in Fig. 2.

3. Mesh of the assembly

Finite element analysis of the joint configuration, shown in Fig. 1 is done using the finite element code ABAQUS. The finite element model is shown in Fig. 3. A layered structure is actually a three-dimensional structure. A three-dimensional finite element model of such a structure involves several degrees of complexity. In this study we make simplifying assumptions which still allow us to capture the essential features of the response. The stress analyses in the SLJ were performed with the non-linear finite element method by considering both the geometrical non-linearity and nonlinear material behavior of both adhesive (ADEKIT A140) and adherend (AA2024-T3). Each layer is considered as



(a) (b)

Fig. 2 Tensile stress-strain curve for: (a) aluminum adherend, (b) adhesive Adekit A140

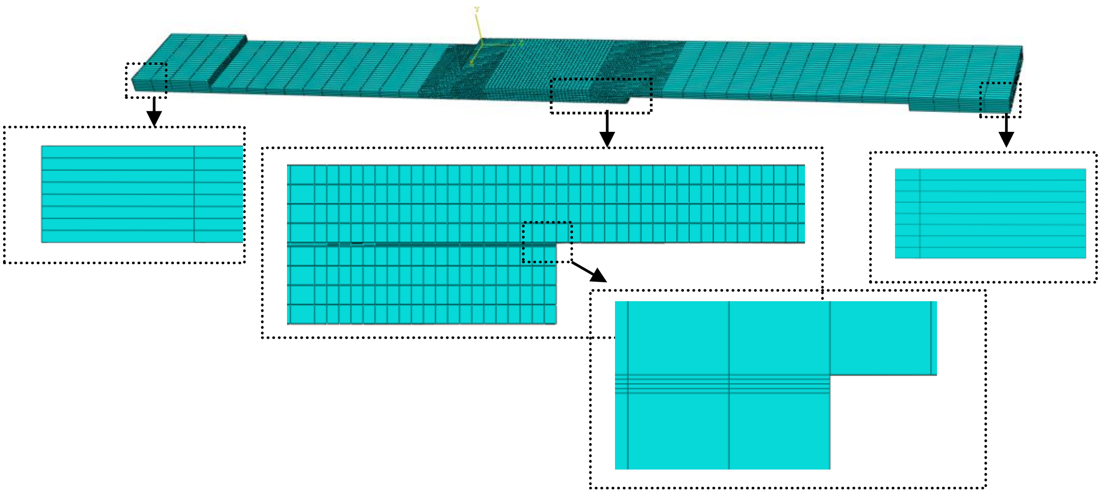


Fig. 3 Typical mesh model of the global structure of the bonded joint

an individual three-dimensional structure under a state of plane-stress. Also, for stress analysis the von Mises yield criterion with Isotropic Hardening was used to calculate the shear (τ_{xy}) and peel stress (σ_{yy}) distributions in the adhesive layers. The bonding is considered as being perfect, smooth transition grid in the model is assured, a refined mesh is made at the edge of the adherend and the selected element is the volume element ("brick") C3D20. The substrates are modeled by elements to 8 nodes (24 degrees of freedom). The mesh is refined on the two ends of the overlap length (Fig. 3).

The conditions applied boundaries are classic for traction models single lap joints. The joint is oriented along the *x*, *z* is the direction of the width, y is the direction normal to the joint plane.

Blocking left: the nodes located on the extreme left face of the joint are blocked in translation in the *x*, *y* and *z*, and rotation in the z direction.

Traction to right: the nodes located on the extreme right face of the joint are blocked in translation in the *y* and *z* directions, and rotation in the *z* direction. 15 MPa of stress is applied in the *x* direction.

4. Results and analysis

With the finite element calculation using the ABAQUS computer code, one could determine the shear and peel stresses in the adhesive layer for different geometrical parameters namely the thickness of the substrates and the adhesive as well as the fillet angle of the adhesive and bevel angle of the adherend.

The results of different stresses based on different parameters have been introduced in a code named MODDE to establish an experience plan to determine the most influencing factors on the values of peel and shear stresses.

The different statistical calculations are performed by the MODDE software. There is two methods to make regression in this software: regression 'PLS' (partial least squares) is used when data is missing and regression 'MLR' (multiple linear regression).The chosen method is MLR, least square fit regressions on several factors. We try to establish a relationship between the input variables (fillet angle of the adhesive, the angle of beveling of the adherend, the thickness of the adhesive and of the adherend) and the output variables (peeling stresses and shear stresses).

For this, we adopt a comprehensive plan experiments called second degree (faces composite plan centered) of 4 factors with 2 levels which offers a surfaces modeling response "RSM", the experimenter's model is quadratic and has following the form

$$Y = a_0 \sum_{i=1}^{4} a_i x_i + \sum_{1 \le i \prec j \le 4} a_{ij} x_j + \sum_{i=1}^{4} a_{ii} x_i^2 + e, \tag{1}$$

Where Y is the response of the process and x_i is the normalized centered value for each factor u_i

$$x_i = \frac{(u_i - u_{ic})}{\Delta u_i} = u_i^* \tag{2}$$

$$u_{ic} = \frac{(u_{i\,\min} + u_{i\,\max})}{2} \tag{3}$$

$$\Delta u_i = \frac{(u_{i\,\max} - u_{i\,\min})}{2} \tag{4}$$

The results of experiments are carried out according to the plan of factorial experiments. Table 2 shows the matrix of experiment.

After studying the effect of the various geometrical shapes possible of adhesive and the adherends edges (with and without beveling of the adherend, with and without adhesive fillet), we analyzed the influence of each factor of the different geometries on shear and peel stresses in the adhesive layer.

4.1 Effect of the adhesive thickness

The thickness of the adhesive layer is an important geometric parameter. The numerical studies of Madani *et al.* (2015, 2009) show that an increase of the adhesive thickness involves a reduction in the shear stress, 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. Halioui (1990) experimentally studied the influence of adhesive thickness on the strength of the single lap joint and showed that it decreases when the thickness of the adhesive film increases.

This effect is shown in Fig. 4, this shows the variation of the peel and shear stress according to the adhesive thickness. Indeed, the maximum value of the stresses obtained for thin layers of the adhesive, the increase of the latter leads to a decrease of the stresses of the order of 21% of the shear and 19% for the peeling stress. These stresses decreases with thickness by stabilizing to its minimum value as the thickness adhesive is e_a=0.23 mm.

It is preferable to increase the thickness of the adhesive layer to improve the strength of the joint. An increased thickness of the adhesive layer, the latter may act as a third material with low mechanical properties.

Adams and Peppiatt (1974) have shown that the effect of the adhesive thickness on joint strength is small in the range 0.1-0.4 mm. Therefore, the thickness variation expected to have little effect on the joint strength.

Further increasing the adhesive thickness decreases the risk of its plasticization. The adhesive becomes too rigid and the load transfer adherend / adhesive / adherend become weak and the stresses are localized in the adhesive thus causing it to break. According Madani *et al.* (2009) the optimum thickness of the adhesive layer is between 0.1 and 0.2 mm.

Table 2 Matrix of the runs

Exp. No.	e_a (mm)	e_p (mm)	α_p (dgrs)	α_a (dgrs)	Shear stresses (MPa)	Peel stresses (MPa)
1	0.1	2	0	0	12.7	15.01
2	0.3	2	0	0	9.24	11.49
3	0.1	4	0	0	17.42	33.12
4	0.3	4	0	0	15.23	25.37
5	0.1	2	60	0	6.75	6.695
6	0.3	2	60	0	4.711	8.662
7	0.1	4	60	0	11.06	18.68
8	0.3	4	60	0	8.346	16.33
9	0.1	2	0	60	6.765	3.556
10	0.3	2	0	60	4.992	2.436
11	0.1	4	0	60	12.62	10.52
12	0.3	4	0	60	9.218	7.41
13	0.1	2	60	60	4.955	1.439
14	0.3	2	60	60	3.666	1.572
15	0.1	4	60	60	9.243	2.017
16	0.3	4	60	60	5.885	2.139
17	0.1	3	30	30	12.56	7.873
18	0.3	3	30	30	9.608	8.834
19	0.2	2	30	30	6.316	5.517
20	0.2	4	30	30	11.12	9.771
21	0.2	3	0	30	10.51	8.425
22	0.2	3	60	30	7.574	8.676
23	0.2	3	30	0	12.2	13.47
24	0.2	3	30	60	6.246	2.423
25	0.2	3	30	30	10.51	8.425

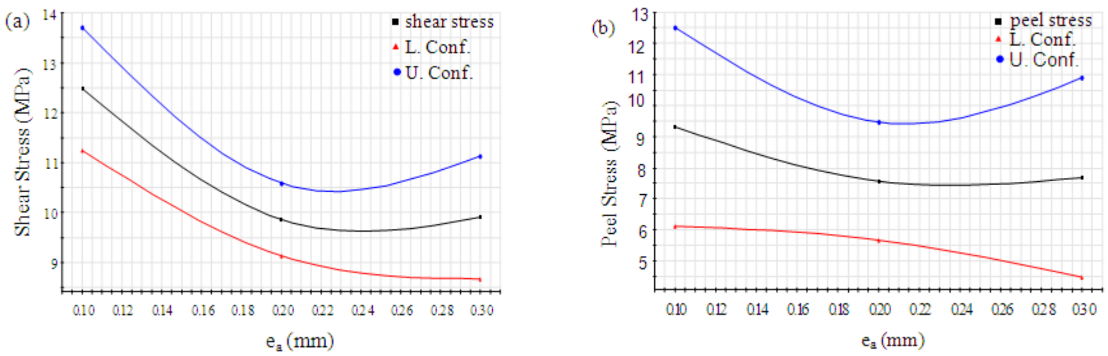


Fig. 4 Prediction plots according to: (a) shear stress; (b) peel stresses versus adhesive thickness

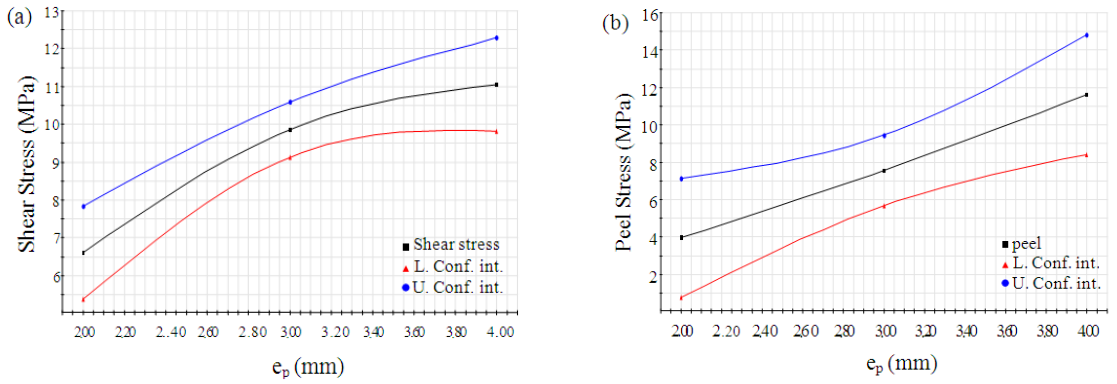


Fig. 5 Prediction plots according to: (a) shear stress; (b) peel stresses versus adherend thickness

4.2 Effect of the adherend thickness

The Fig. 5, illustrates the effect of the adherents thickness “e_p” on the peel and shear stresses. It is noted that increasing the thickness of the adherend “e_p” causes an increase in stresses; this rise in stresses may reach 40% for the shear and 65% for the peeling stress. Increasing the thickness of the adherend causes a significant bending moment in addition to the eccentric loading which can lead to deflections of the assembly (Karachalios *et al.* 2013).

The local stress variations near the edges of the overlap region are characterized by very high gradients. The gradients of the stresses of components depend on the elastic properties of the adherents, the adhesive and the common geometry.

4.3 Effect of the beveling angle of the adherend

The geometry of the joints with rounded adherends studied experimentally by Zhao *et al.* (2011) showed that for a ductile adhesive the strength of joints with sharp adherend corners was slightly higher than those of joints with different degrees of rounding of the adherend corners. It seems that the stresses or strains in a finite area around the stress concentration point govern the failure of lap joints.

Both geometrical modifications of the adhesive and the adherend edge at the same time have

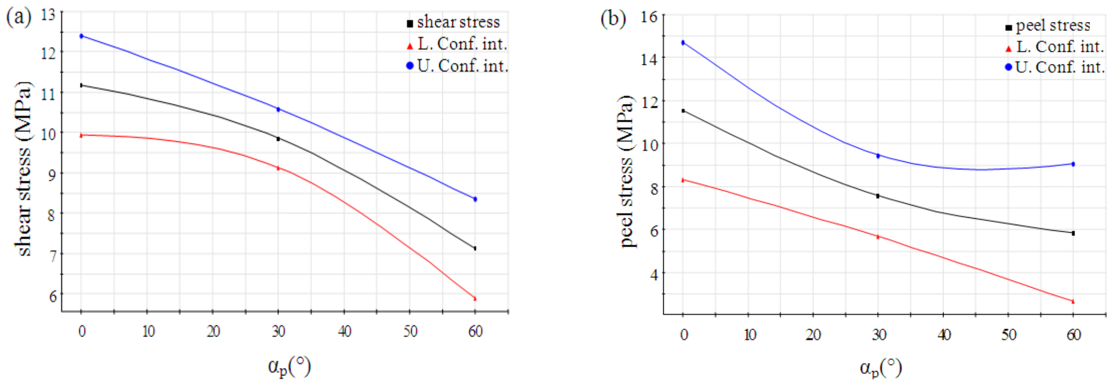


Fig. 6 Prediction plots according to: (a) shear stress; (b) peel stresses versus beveling angle of the adherend

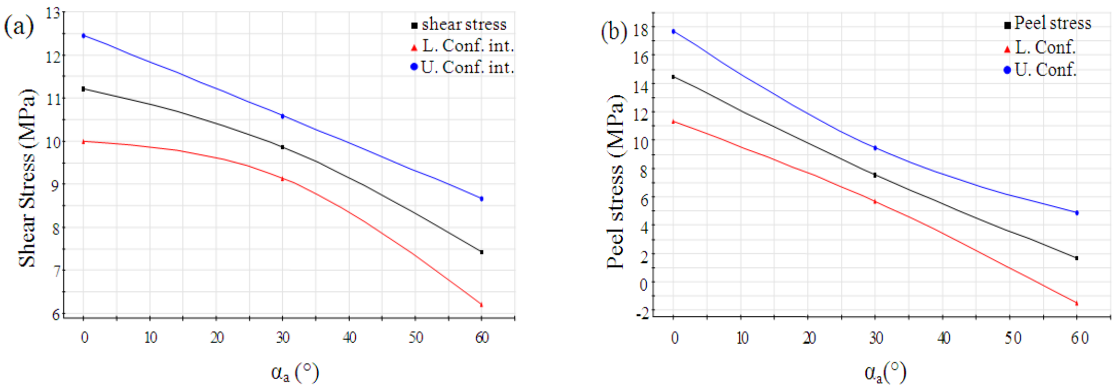


Fig. 7 Prediction plots according to: (a) shear stress; (b) peel stresses versus fillet angle of the adhesive

shown a significant effect on overlap levels of the edge. In this study the variation in the beveling angle causes automatically the variation of the fillet angle of adhesive.

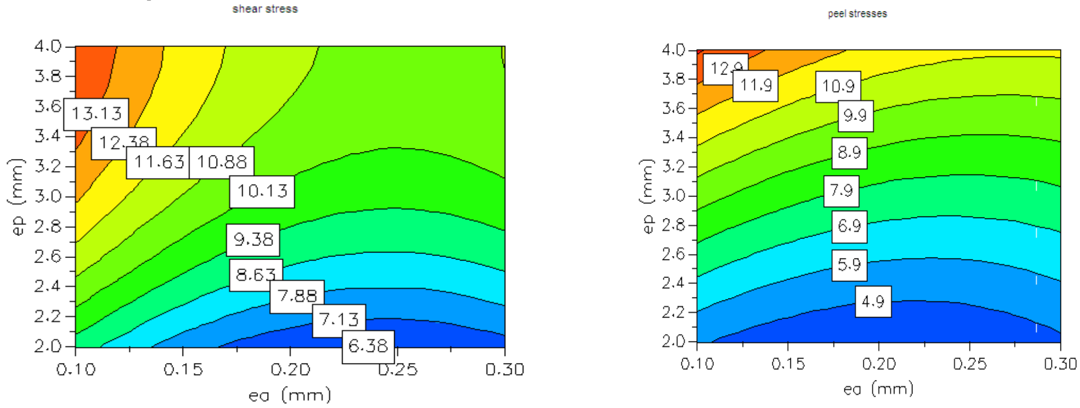
The Beveling of the edges of the adherends may seem interesting, since the reduction in the thickness at this level minimizes the bending moment around the joint thereby causing a decrease in peel overstressing and shear stresses (Fig. 6). It is noted that the increase in the beveling angle of the adherend α_p causes significant reduction of stresses. This reduction of the stress in the adhesive is of the order of 36% of the shear stress and 44% for the peeling stress.

4.4 Effect of the fillet angle of the adhesive

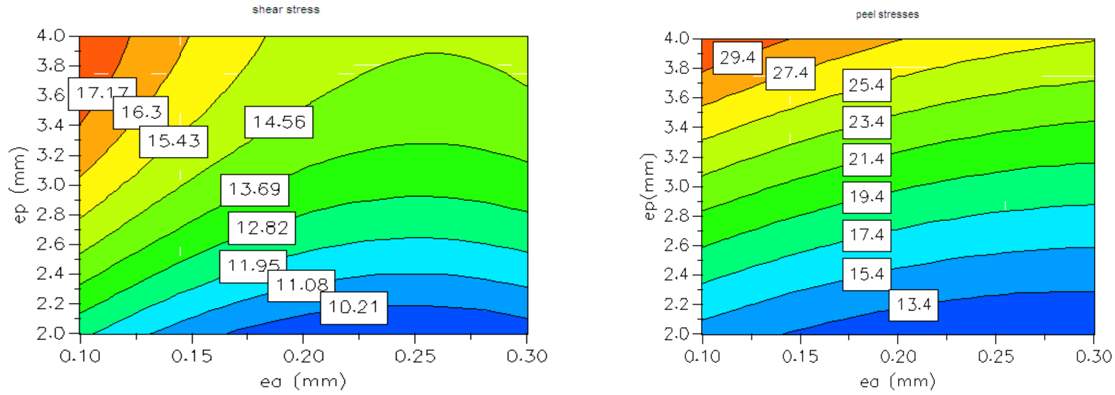
In many study (Karachalios *et al.* 2013), the fillets have been shown to reduce significantly the peak stresses at the ends of the joints. The use of chamfered spacers is one of the most effective ways to control the bondline thickness (Dorn and Liu 1993). Zhao *et al.* (2011) and da Silva *et al.* (2007) shows that a fillet of adhesive around the joint further improves the mechanical strength of the assembly that the beveling of the adherends.

Indeed, the presence of an adhesive fillet angle increases the bonding area thus minimizing stress concentration at the edge of the adhesive. This reduction varies with the variation of the angle of the fillet, it is preferable to increase the adhesive fillet angle to have a fairly substantial overlap length.

For: α_a=30° / α_p=30°



For: α_a=0°/ α_p=0°



For: α_a=60°/ α_p=60°

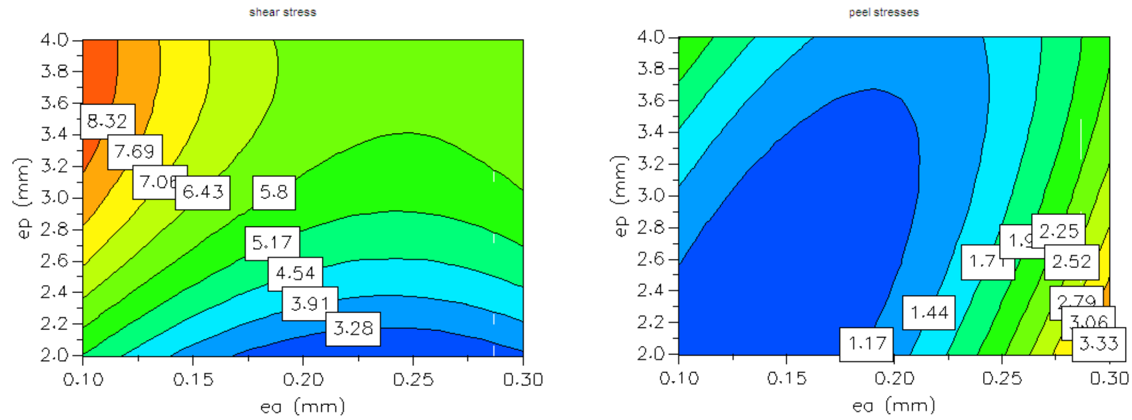


Fig. 8 contours response (Iso-response) versus (e_a,e_p,α_a,α_p)

The adhesive fillet angle influence "α_a" on the peeling stress and the shear stress is illustrated in Fig. 7. This shows that the increase of the adhesive fillet angle resulting in lower stresses 34% of the shear stress and 90% for the peeling stress.

4.5 Interaction effect of different factors on the response

The results of the various parameters mentioned previously on the value of the peel and shear stresses are shown in Fig. 8. This analysis considers the effect of interaction between two factors (thickness of the adherend and the thickness of the adhesive) on the peel stress and shear stress, while keeping the other two parameters constant (angle of the beveling of the adherend and adhesive fillet angle).

Note that the minimum value of the shear stresses corresponds to values of the adherend thickness between 2 and 2.15 mm and an adhesive thickness of 0.2 and 0.23mm with a beveling angle of the adherend α_p=60° and the fillet angle of adhesive α_a =60°. For the peeling stresses the minimum value corresponds to the values of the thickness of the adherend comprised between 2 and 2.5 mm, an adhesive thickness comprised between 0.18 and 0.2 mm with a beveling angle of the adherend α_a=60° and fillet angle of adhesive α_p=60°.

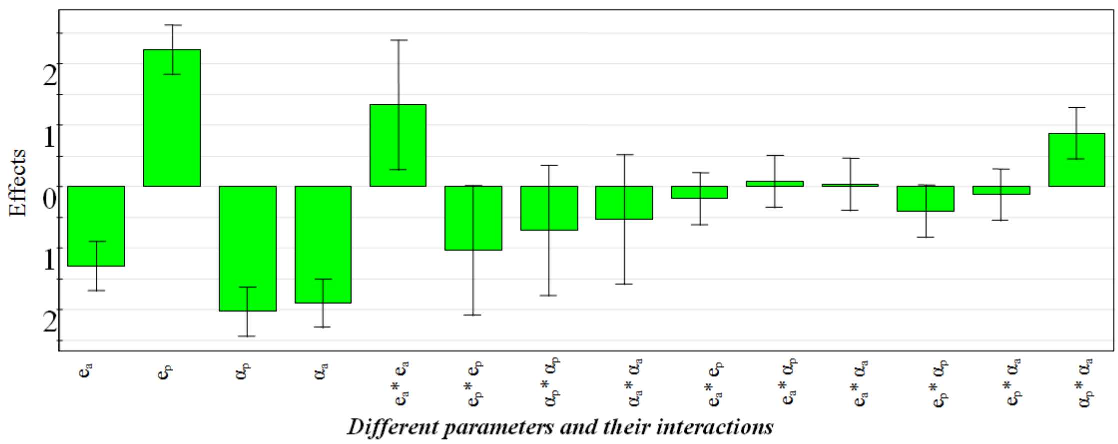
4.6 Effect of different factors on the peel and shear stresses

It is important to study the effect of different factors on the performance of a simple lap joint. These effects are represented by a histogram. This diagram shows the effects in decreasing order of their importance in absolute value. The effects of all the terms of the factors (linear, quadratic and cross) on the shear stress and peel stresses are shown in Fig. 9.

The analysis of these diagrams shows that the effect of the two factors the beveling angle of the adherend "α_p" and the adhesive fillet angle "α_a" are the most significant and the most dominant in the optimization of the geometric parameters of a bonded assembly. The second dominant factor is the thickness of the adhesive "e_a" and finally, the adherend thickness "e_p" as the factor having a significant effect on a stresses distribution in a single bonded joint.

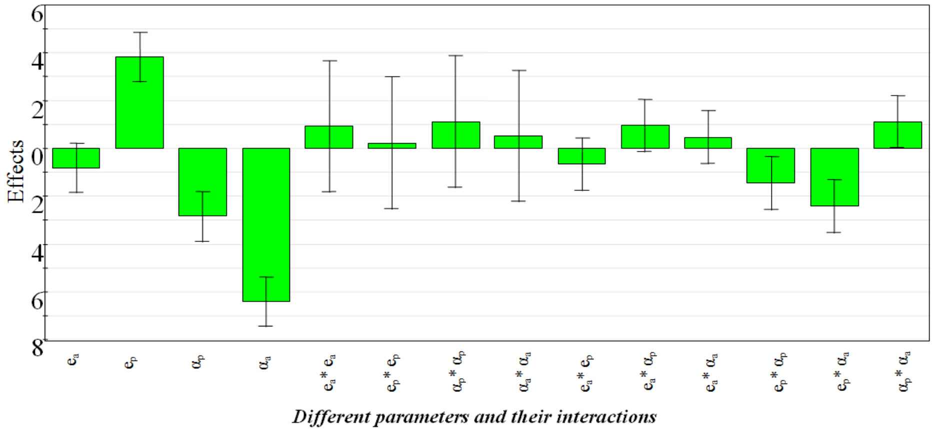
4.7 Checking the optimal point

Table 3 illustrates the optimal point found by MODDE software. Indeed, the lowest values of the



(a)

Fig. 9 Effects of the different parameters and their interactions: (a) Shear stress (b) Peel stress



(b)

Fig. 9 Continued

Table 3 Optimal geometric factors dimensions

e_a	e_p	α_p	α_a	τ_{xy}	σ_{yy}	iter
0.243	2	0.0003	59.999	4.1112	1.798	165
0.189	2.015	59.3476	59.998	3.0568	1.167	163
0.271	3.9994	31.1623	60	8.0857	3.005	201
0.250	4	59.9768	59.999	5.8847	1.515	170
0.187	2.0001	59.993	59.475	3.0008	1.233	269
0.242	2.0002	0.0004	59.999	4.1124	1.797	162
0.197	2.0504	59.778	59.957	3.0658	1.272	89
0.228	2.0048	52.056	59.515	3.2375	1.297	39

Table 4 Peel stress coefficients list

Peel stresses	Coeff. SC	Std. Err.	*P*	Conf. int
Constant	7.56171	0.848976	4.54334e-006	1.89162
e_a	-0.814833	0.463435	0.109213	1.03259
e_p	3.83222	0.463435	8.80182e-006	1.03259
α_p	-2.84039	0.463435	0.000111356	1.03259
α_a	-6.40639	0.463435	7.64358e-008	1.03259
$e_a * e_a$	0.93567	1.23206	0.465117	2.74517
$e_p * e_p$	0.22617	1.23206	0.858019	2.74517
$\alpha_p * \alpha_p$	1.13267	1.23206	0.379564	2.74517
$\alpha_a * \alpha_a$	0.528671	1.23206	0.676951	2.74517
$e_a * e_p$	-0.659251	0.491547	0.209529	1.09523
$e_a * \alpha_p$	0.960751	0.491547	0.0791495	1.09523
$e_a * \alpha_a$	0.479875	0.491547	0.351963	1.09523
$e_p * \alpha_p$	-1.44563	0.491547	0.014761	1.09523
$e_p * \alpha_a$	-2.41	0.491547	0.000620445	1.09523
$\alpha_p * \alpha_a$	1.11675	0.491547	0.0464197	1.09523
N = 25 DF = 10	Q2 = 0.848 R2 = 0.972 R2 Adj. = 0.934		Cond. no. = 6.5927 Y-miss = 0 RSD = 1.9662 Conf. lev. = 0.95	

shear stress (τ_{xy}=3 MPa) and the peeling stress (σ_{yy}=1.23 MPa) are obtained for the adhesive thickness e_a=0.1879 mm, the thickness of the adherend e_p=2 mm, a bevel adherend α_p=59.993 ° and an adhesive fillet α_a=59.475 °.

The optimization process is to maximize the beveling of the adherend and the adhesive fillet and minimize the thickness of the adhesive and the thickness of the adherend. This optimization study of the geometric factors to minimize stress responses can be achieved by experience with the geometric parameters shown in the Table 3.

According to the analysis made by MODDE software, we allow for a list of mathematical models of the coefficients for the shear stress (τ_{xy}) and peel (σ_{yy}). Table 4 and Table 5 represent the coefficients models list mathematics for each answers.

After the combination of Eqs. (2), (3) and (4) in Eq. (1) and from the tables representing the list of coefficients of the various expressed as interaction effects on the responses of peeling stress (σ_{yy}) and shear stress (τ_{xy}) it has been possible to obtain the following mathematical models

$$\begin{aligned}\sigma_{yy} &= 93.5669 e_a^2 + 0.22617 e_p^2 + 0.00058741 \alpha_a^2 + 0.001258522 \alpha_p^2 - 6.5925 e_p e_a + 0.1899583 e_a \alpha_a \\ &+ 0.3202497 e_a \alpha_p - 0.0481876 e_p \alpha_p - 0.08033 e_p \alpha_a + 0.00124083 \alpha_a \alpha_p - 40.2038 e_a + 7.6493 e_p \\ &- 0.077007 \alpha_a - 0.126903 \alpha_p + 2.85664\end{aligned} \tag{5}$$

$$\begin{aligned}\tau_{xy} &= 132.954 e_a^2 - 1.03646 e_p^2 - 0.000591 \alpha_a^2 - 0.00079162 \alpha_p^2 - 1.93938 e_p e_a + 0.0121043 e_a \alpha_a \\ &+ 0.029687366 e_a \alpha_p - 0.0043062336 e_p \alpha_a - 0.01320626 e_p \alpha_p + 0.00096368 \alpha_a \alpha_p - 61.4937 e_a \\ &+ 9.356841 e_p - 0.0460691466 \alpha_a - 0.0153340733 \alpha_p + 2.8082972\end{aligned} \tag{6}$$

Table 5 Shear stress coefficients list

shear stress	Coeff. SC	Std. Err.	P	Conf. int
Constant	9.86239	0.326446	3.69572e-011	0.72736
e_a	-1.28761	0.178199	2.83864e-005	0.397048
e_p	2.22483	0.178199	2.01075e-007	0.397048
α_p	-2.02806	0.178199	4.80003e-007	0.397048
α_a	-1.89261	0.178199	9.12791e-007	0.397048
e_a * e_a	1.32954	0.473747	0.0185879	1.05556
e_p * e_p	-1.03645	0.473747	0.0535366	1.05556
α_p * α_p	-0.712456	0.473747	0.163523	1.05556
α_a * α_a	-0.531454	0.473747	0.288158	1.05556
e_a * e_p	-0.193938	0.189008	0.329033	0.421133
e_a * α_p	0.0890628	0.189008	0.647602	0.421133
e_a * α_a	0.0363128	0.189008	0.851491	0.421133
e_p * α_p	-0.396188	0.189008	0.0624813	0.421133
e_p * α_a	-0.129188	0.189008	0.509816	0.421133
α_p * α_a	0.867312	0.189008	0.000997153	0.421133
N = 25 DF = 10	Q2 = 0.885 R2 = 0.980 R2 Adj. = 0.952		Cond. no. = 6.5927 Y-miss = 0 RSD = 0.7560 Conf. lev. = 0.95	

5. Conclusions

In this study, the analysis of the effect of various geometrical parameters and their interactions on the shear and peel stress distribution in the adhesive layer of SLJ has been studied numerically. The analysis of the influence of these parameters was performed by the method of experimental design. Accordingly, the following conclusions can be drawn:

- The reduction of the stresses which can improve the durability of the SLJ .The maximum value of the shear and peel stress obtained for thin layers of the adhesive, the increase of the latter leads to a decrease of the stresses of the order of 21% of the shear stress and 19% for the peeling stress. These stresses decreases with thickness by stabilizing to its minimum value as the thickness adhesive is e_a=0.23 mm.
- The increasing the thickness of the adherend causes an increase in stresses; this rise in stresses may reach 40% for the shear stress and 65% for the peeling stress.
- The presence of fillet adhesive in the SLJ increases the bonding area thus minimizing stress concentration at the edge of the adhesive. The increase of the adhesive fillet angle resulting in lower stresses 34% of the shear stress and 90% for the peeling stress.
- The increase in the beveling angle of the adherend causes significant reduction of stresses. This reduction of the stress in the adhesive is of the order of 36% of the shear stress and 44% for the peeling stress.
- The effect of the two factors the beveling angle of the adherend and the adhesive fillet angle are the most significant and the most dominant in the optimization of the geometric parameters of a bonded SLJ.

References

ABAQUS/CAE (2009), Ver 6.9 User’s Manual, Hibbitt, Karlsson and Sorensen, Inc.

Adams, R., Atkins, R., Harris, J. and Kinloch, A. (1986), “Stress analysis and failure properties of carbon-fibre reinforced plastic/steel double lap-joint”, *J. Ahdes.*, **20**, 29-30.

Adams, R.D. and Peppiatt, N.A. (1974), “Stress analysis of adhesive-bonded lap joints”, *J. Strain Anal.*, **9**(3), 185-196.

Akpinar, S. (2014), “The strength of the adhesively bonded step-lap joints for different step numbers”, *Compos. Part B*, **67**, 170-178.

Benchiha, A. and Madani, K. (2015), “Influence of the presence of defects on the stresses shear distribution in the adhesive layer for the single-lap bonded joint”, *Struct. Eng. Mech.*, **53**(5), 1017-1030.

Campilho, R.D.S.G., de Moura, M.F.S.F. and Domingues, J.J.M.S. (2009), “Numerical prediction on the tensile residual strength of repaired CFRP under different geometric changes“, *Int. J. Adhes. Adhes*., **29**, 195-205.

Campilho, R.D.S.G., de Moura, M.F.S.F., Domingues, J.J.M.S. and Morais, J.J.L. (2008), “Computational modelling of the residual strength of repaired composite laminates using a cohesive damage model”, *J. Adhes. Sci. Technol*., **22**(13), 1565-1591.

da Silva, L.F. and Adams, R.D. (2007), “Techniques to reduce the peel stresses in adhesive joints with composites”, *Int. J Adhes. Adhes.*, **27**, 227-235.

da Silva, L.F., das Neves, P.J., Adams, R. and Spelt, J. (2009), “Analytical models of adhesively bonded joints-Part I: Literature survey“, *Int. J. Adhes. Adhes.*, **29**, 319-330.

da Silva, L.F., das Neves, P.J., Adams, R., Wang, A. and Spelt, J. (2009), “Analytical models of adhesively bonded joints-Part II: Comparative study“, *Int. J. Adhes. Adhes.*, **29**, 331-341.

Dorn, L. and Liu, W. (1993), “The stress state and failure properties of adhesive-bonded plastic/metal

joints", *Int. J. Adhes. Adhes.*, **13**(1), 21-31.
Grant, L.D.R., Adams, R.D. and da Silva, L.F.M. (2009), "Experimental and numerical modeling of single-lap joints for the automotive industry", *Int. J. Adhes. Adhes*., **29**, 405-413.
Halioui, M. (1990), "Contribution à l'évolution du comportement mécanique non-linéaire d'assemblages collés de tôles minces d'acier sous sollicitations monotones ou cycliques", PhD. Université de Reins.
Higgins, A. (2000), "Adhesive bonding of aircraf structures", *Int. J. Adhes. Adhes*., **20**, 367-76.
Hildebrand, M. (1994), "Non-linear analysis and optimization of adhesively bonded single lap joints between fibre-reinforced plastics and metals", *Int. J. Adhes. Adhes.*, **14**(4), 261-267.
Karachalios, E.F., Adams, R.D. and da Silva, L.F.M. (2013), "Single lap joints loaded in tension with high strength steel adherends", *Int. J. Adhes. Adhes.*, **43**, 81-95.
Karachalios, E.F., Adams, R.D. and da Silva, L.F.M. (2013), "The behaviour of single lap joints under bending loading", *J. Adhes. Sci. Technol*., **27**(16), 1811-1827.
Kumar, S. and Pandey, P.C. (2010), "Behaviour of bi-adhesive joints", *J. Adhes. Sci. Technol.*, **24**, 1251-81.
Li, G. and Li, C. (2014), "Linking bilinear traction law parameters to cohesive zone length for laminated composites and bonded joints", *Adv. Aircraft Spacecraft Sci.*, **1**(2), 177-196.
Madani, K., Boukhoulda, B.F., Touzain, S. and Feaugas, X. (2009), "Numerical analysis of the notch effect on tangential stress distribution of the adhesive layer used for bonding two aluminum adherend 2024-T3", *Comptes Rendus des JNC 16-Toulouse*, France.
Madani, K., Mokhtari, M., Belhouari, M. and El hannani, M. (2013), "Effect of modifying the edges of the adherents and the adhesive on the stress distribution over the width and length of recovery, case of a single lap joint", *Int. J. Min. Metal. Mech. Eng.*, **4**(1), 262-267.
MODDE 6.0 (Moeeling and Design) Umetrics AB, Umea, Sweden.
Pinto, A.M.G., Campilho, R.D.S.G., Mendes, I.R. and Baptista, A.P.M. (2014), "Numerical and experimental analysis of balanced and unbalanced adhesive single-lap joints between aluminium adherends", *J. Adhes.*, **90**(1), 89-103.
Tsai, M.Y. and Morton, J. (1995), "The effect of a spew fillet on adhesive stress distributions in laminated composite single-lap joints", *Compos. Struct.*, **32**,123-131.
Turuga, U. and Sun, C.T. (2008), "Improved design for metallic and composite single-lap joints", *J. Aircraft*, **45**(2), 440-447.
Zeng, Q.G. and Sun, C.T. (2001), "Novel design of a bonded lap joint", *AIAA J.*, **39**(10), 1991-1996.
Zhao, X., Adams, R.D. and da Silva, L.F.M. (2011), "Single lap joints with rounded adherend corners: experimental results and strength prediction", *J. Adhes. Sci. Technol*., **25**(8), 837-56.

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