INFLUENCE OF SOME MODIFICATIONS OF LOCAL GEOMETRY ON THE STRESS STATES IN ADHESIVE BONDED LAP JOINTS

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In most structures the joints between the parts are usually critical zones because of intensive local loadings. During the last decades, adhesive bonding was widely used, especially in the aerospace and automotive industries for joining dissimilar materials. Adhesive bonding offers important advantages in comparison with the mechanical fastening (as example, by rivets or bolts), as follows: continuity of bonding lines, a more uniform distribution of stresses over the overlap area, significant savings in the energy consumption and costs reduction. In this paper, several ideas to improve the performance of adhesive bonded lap joints in light structures from aluminium were discussed. Some geometrical modifications as: tapering of the substrates, pre-bending the adherends in the overlap zone or mould the adhesive spew as a fillet at the adhesive layer ends, were identified as having beneficial effects in the stress concentration diminution. The finite element modeling and analyses were undertaken in order to emphasize the influence of various geometrical modifications.

Keywords: adhesive bonded; lap joints; stress concentration diminution.

1. INTRODUCTION

Bonded assemblies allow for a gradual transfer of load from one structural element to another and the diminution of stress concentrations due to the material discontinuities inherent to mechanical fastening methods. Although most common adhesives are homogenous and isotropic materials, the stress and deformation states in the bond line and the joint failure mechanisms are very complex [1].

A great number of papers that deals with the improvement of the shape of the adherends as well as the modification to the adhesive composition to obtain better properties under working conditions.

The most studied is the single lap joint that is efficient only if the assembled sheets are very thin. The main drawback of single lap bonded joints are the high localized stresses at the adhesive layer ends with a little stress carried in its large central zone, if a relatively stiff adhesive is used. Different ideas have been investigated in order to obtain a better distribution of the stresses and to diminish the stress peaks. The modifications which were proposed include pre-bending of the adherends in the overlap zone [2], use of local “wavy” shape of the substrates [3], tapering the adherends and mould the adhesive spews as fillets at the joints ends [4]-[7] or other special machining of the sheets in the superposition region [8].

An old photoelastic study [9] on the “reverse-bent” joint showed significantly reduced stress peaks by comparison with the single lap joint manufactured by using straight sheets with the same thickness.

Two similar studies [2] and [3] were developed in order to evaluate the efficiency of pre-bending in the case of steel sheets bonded by epoxy adhesives. The results of any numerical simulations developed in paper [2] by using elastic-plastic finite element analysis showed that the peak stresses were reduced by pre-bending the adherends ends, and for an optimum performing angle (near to 7°) the tensile load capacity of the joint can be increased with about 64 %. In [3] only an increasing of 40 % is reported.

In this paper a simple solution to enhance the performance of reverse bent joint by is proposed and analyzed based on Finite Element Analysis (FEA). The ends of sheets will be prepared both by pre-bending and by tapering. The results of the improved joint design were compared with the ones obtained for the simple lap joint having the same overlap length.
2. DESCRIPTION OF THE FINITE ELEMENT MODELS

The adherends were considered as made from aluminium 2024-T3 having the modulus of elasticity \( E = 73000 \) MPa, Poisson’s ratio \( \nu = 0.33 \) and tensile strength \( R_m = 480 \) MPa. For joining the aluminium sheets was used a structural epoxy adhesive that have the following mechanical properties: Young’s modulus \( E_a = 3300 \) MPa, Poisson’s ratio \( \nu_a = 0.35 \), tensile strength \( \sigma_{fa} = 70 \) MPa and shear strength \( \tau_{fa} = 47 \) MPa.

The configurations of the joints that will be discussed are presented in Table 1. The objective of the study is to evaluate the stress peaks for these geometries and to emphasize the influence of overlap length on the stress distribution along the adhesive layer.

Table 1 : The joints that were analyzed and compared

<table>
<thead>
<tr>
<th>Variant number</th>
<th>Symbol</th>
<th>Sketches of the joints</th>
</tr>
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<tbody>
<tr>
<td>1</td>
<td>SL (single lap)</td>
<td>![Sketch of SL joint]</td>
</tr>
<tr>
<td>2</td>
<td>PB (pre-bended)</td>
<td>![Sketch of PB joint]</td>
</tr>
<tr>
<td>3</td>
<td>TPB (tapered pre-bended)</td>
<td>![Sketch of TPB joint]</td>
</tr>
<tr>
<td>4</td>
<td>TPBL (tapered pre-bended long)</td>
<td>![Sketch of TPBL joint]</td>
</tr>
</tbody>
</table>

Linear elastic analyses were undertaken by using parametric models and eight-node quadrilateral finite elements. The adhesive and the substrates were divided into 4 and 6 elements through thickness, respectively. Along the bonding line were considered 40 elements for variants 1 to 3 and 60 elements in case of variant 4. The force \( F \) was taken as to induce a nominal stress \( \sigma_n = 100 \) MPa into the substrates.

In all cases were maintained constant the thickness \( t = 3 \) mm, the width \( b = 25 \) mm, and the adhesive layer thickness \( t_a = 0.15 \) mm. The stress states were evaluated for five different values (10, 15, 20, 25 and 30 mm) of the overlap length \( l \). The performing angle is depending of the overlap length. In case of design variants 2 and 3, the inclination angle can calculate by the relation

\[
\sin \phi \approx \frac{t}{l}.
\]  (1)
Increased height for variant number 3 was taken $s = 0.5$ mm. Parameters $s_i$ and $l_i$ in case of the variant 4 were established (in mm) by using relations

$$s_i = s + 5 \cdot \sin \varphi, \quad l_i = l + 10. \quad (2)$$

### 3. THE RESULTS OF FINITE ELEMENT ANALYSES

For inter-comparative purposes, the peel, shear, tensile and equivalent stresses ($\sigma_y$, $\tau_{xy}$, $\sigma_x$, $\sigma_{eq}$) were normalized with respect to the nominal stress $\sigma_n = F / (b_t) = 100$ MPa.

In the case of the adhesive, following recommendation of the EUROCOMP Design Code [10], the Hill’s failure criterion will be applied. The stress state is allowable in a point of the adhesive layer if the condition

$$\left( \frac{\sigma_y}{\sigma_{fa}} \right)^2 + \left( \frac{\tau_{xy}}{\tau_{fa}} \right)^2 \leq 1, \quad (3)$$

is accomplished.

Although the values of $\sigma_x$ are significant in many cases, it is to observe that this stress is not involved in the failure theory (3).

The diagrams of the variation of the maximum values of normalised stresses over the length $l$ of the adhesive are presented in Figure 1, and emphasize the great discrepancy between the poor joint (SL) and the best (TPBL). The equivalent stress as example is for the joint TPLB about of 10 times less than in case of single lap joint (SL). It is to remark that the peel stress considered the most dangerous for the integrity of adhesive has a spectacular decreasing if the adherends are tapered (variant TPB) and adhesive wedges are formed at the joints ends (variant TPBL).

![Figure 1. Influence of the overlap length on maximum values of stresses $\sigma_{eq}$, $\sigma_x$, $\sigma_y$, $\tau_{xy}$ in adhesive](image-url)
For all variants the maximum stresses in the adherends, due mainly to the loading in bending, were obtained in the close vicinity of the adhesive layer ends. The overlap length has a little influence on the maximum equivalent stress in the adherends, but the geometry of the joint is very important. In case of single lap joint (SL) an equivalent stress in the adherends of 4.5 times greater than \( \sigma_n \) was obtained, while the corresponding result in case of variants PB, TPB and TPBL are in between \( 1.1 \sigma_n \) and \( 1.5 \sigma_n \).

The discussion will be continued in terms of absolute values of the stresses induce by the loading \( F = 7500 \) N, chosen as to produce a nominal stress \( \sigma_n = 100 \) MPa into the substrates.

Variant SL is unacceptable because of the strong stress concentration both in adhesive and adherends at the joint ends. For the improved variants PB, TPB and TPBL the distribution of stresses \( \sigma_{eq}, \sigma_x, \sigma_y, \tau_{xy} \), are presented in figures 2, 3 and 4, for half of the overlap length. It is not necessary to show the stresses variation along all overlap length because the diagrams are symmetric. The dimensionless coordinate \( x/(2l) \) is equal to 0 at the edge of the joint and becomes equal to 1 at the middle.

The presented results were obtained by linear elastic FEA. Consequently, in order to evaluate the load capacity of a joint configuration the value of force \( F \) will be increase or decreased proportionally while the criterion (3) is accomplished.

The single lap joint (SL) is unbalanced and because of the loading in bending, the deformed structure will be S-shaped. An improvement obtained by using pre-bending and tapering of the adherends edges before bonding is the reduction of sheets deflections.

Table 2 contains the maximum absolute displacements along the \( y \) axis normal to the loading line of the joint. It is evident the beneficial effect of geometrical modifications in the overlap zone.
Figure 3. Distribution of stresses $\sigma_{eq}$, $\sigma_x$, $\sigma_y$, $\tau_{xy}$ in adhesive, for variant TPB, along half of the overlap length

Figure 4. Distribution of stresses $\sigma_{eq}$, $\sigma_x$, $\sigma_y$, $\tau_{xy}$ in adhesive, for variant TPBL, along half of the overlap length
Table 2 : Values of maximum deflections for the analyzed cases [in mm]

<table>
<thead>
<tr>
<th>The joint type</th>
<th>The overlap length ( l ) [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10</td>
</tr>
<tr>
<td>SL</td>
<td>0.368</td>
</tr>
<tr>
<td>PB</td>
<td>0.0035</td>
</tr>
<tr>
<td>TPB</td>
<td>0.0036</td>
</tr>
<tr>
<td>TPBL</td>
<td>0.0088</td>
</tr>
</tbody>
</table>

Because of significant deformation under loading, the single lap joint is recommended only in case of very thin substrates. The modifications above proposed and analyzed are efficient and easy to apply in the case of the sheets with mean thicknesses.

4. CONCLUSIONS

The main drawback of a single-lap joint is that bending loads occur in the adherends and peeling stresses act into the adhesive with the result that the joint has a reduced load capacity. The equivalent, shear and peeling stresses have peak values at the edge of the overlap region over a very short distance. However, this kind of joint becomes acceptable in the case of an adequate pre-bending and machining of the ends of the adherends in the overlap zone. The study that was undertaken showed a spectacular decreasing of the maximum equivalent stress and of the peel stress, considered as the most dangerous for the integrity of adhesive, if the adherends are tapered and adhesive wedges are formed at the joints ends. Additionally, the tendency to curve the adherends by bending is strongly diminished.

REFERENCES

10. * * * Guide to the structural use of adhesives, The Institution of Structural Engineers, 1999.