Adhesively bonded single lap joints with non-flat interfaces

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A series of experiments and detailed finite element simulations were carried out to study the performance of single lap joints with non-flat interfaces loaded at a small angle to the mean bond plane. In the experimental part, fiber reinforced epoxy composite adherends with sinusoidal bonding surfaces (but no change to the exterior sample dimensions) were fabricated in a purpose-built mold. This construction method allowed the fibers to follow the sinusoidal profile for maximum strength. Then the sinusoidal surfaces were joined by a controlled-thickness layer of epoxy. Single lap joints with a flat interface were also fabricated using the same composite material and with the same overall shape. The experiments showed that the interface non-flatness has significant effect on the mechanical behavior and strength of the bonded joints. Finite element simulations of a simplified two-dimensional model with isotropic adherends were also carried out to estimate the distribution of shear and peeling stresses along the bonded joints, and the results were related to the experimental observations. Parametric variations were studied via FEA to highlight the role of interface shape, adhesive elastic modulus and a central void size on the distribution of stresses, and inherently the overall strength and behavior of the bonded joints.

Keywords: Adhesively bonded joint; composites; interface profile; failure; finite element
1. Introduction

Lightweight materials are widely used in the design of marine and aerospace structures where strength-to-weight ratio is a critical design factor. Many of these structures must achieve diverse – and often divergent – properties and function, and thus are made from dissimilar materials which might have very different properties. Such hybrid structures can minimize weight by using the lowest density material with the appropriate strength, stiffness or ductility in each area of the structural assembly. But a resulting key challenge of hybrid structures is joint performance and its integrity during the service life of the structure. Traditional metal-joining methods (e.g. welding, brazing) are not suitable for joining fiber-reinforced polymer (FRP) composites or organic materials. Adhesive bonding, on the other hand, is capable of joining materials with different geometry, properties and functions [1-3]. Unfortunately, in most cases the bonded joint is the ‘weakest link’ of the structure, with inferior strength compared to other components, thus limiting the overall load-carrying capacity of the structural system.

The quest to design more effective bonded joints has led to development of strong, heat-stable adhesives with enhanced adhesion properties [2, 4]. Here, we tackle the joint-strength challenge from another perspective and investigate the role of interface shape on the performance and strength of adhesively bonded joints. While it is common practice to roughen the surface of adherends when attaching them by adhesives (i.e., in the context of surface preparation) [5] the roughness amplitude and wavelength are generally much smaller than the adhesive layer thickness and are induced to assure appropriate adhesion at the interface of the adhesive-adherend. Our focus, in contrast, is on the possibility of altering the mechanics of load transfer, to increase strength and perhaps ductility, without however altering the dimensional envelope of the joint.

To examine the feasibility of enhancing the structural performance of bonded joints by varying interface shape but not overall joint envelope, we created sample joints with sinusoidal interfaces using unidirectional glass FRP shaped in a custom made mold. Flat-joint specimens with the same adherend and adhesive thickness, bond length and specimen width were also fabricated. The adherends were bonded together to form a single lap joint using a commercially available adhesive while maintaining careful control of bond-line thickness. The force-displacement response of the specimens was measured in axial straining without grip rotation up to overall failure. The details of the experiments are provided in Section 2. In Section 3, we
present simplified finite element analyses of the distribution of shear and peeling stresses in both flat and non-flat bonded joints. An extensive parametric study was also carried out to highlight the role of interface shape (i.e. wavelength and amplitude) and the adhesive stiffness on the distribution of peeling and shear stresses, particularly their maximum values. In Section 4, we present our finite-element examination of the effect of a central void in the adhesive layer on the distribution of stresses. It should be noted that the use of substantially sinusoidal joint interfaces is not entirely novel. For example, Zeng and Sun [6] introduced what they called a ‘wavy lap joint’, differing from our explorations in that the adherends maintained constant thickness, with a bulkier resulting joint (our design maintains overall joint thickness). Ávila, and de O. Bueno [7] performed both experiments and FEA to show a strength about 40% higher. And, Shankar et al. [8] also reported on the strength increase in wavy lap joints. Our investigation may be seen as separating the factor of interface shape from that of overall joint shape.

2. Experimental investigations

In this study, the adherends were made from E-glass/epoxy composite (EG-SR-183/32-24”, ALDILA) with a ply thickness of approximately 130 μm. To create a traditional flat-interface lap joint, flat composite sheets were fabricated using hot press molding. First, a mold release (Xtend 19SAM, AXEL) was applied to the steel mold surfaces. Then, composite sheets of 4.5 mm -- 5.5 mm thick were made by stacking 42 unidirectional plies (i.e. the fibers are in the same direction for all the plies). The composite sheets were cured for 1 hour at 130 °C and then allowed to cool, while subjected to a continuous ply-consolidating pressure of 1 MPa. The elastic modulus in the fiber direction of the cured composite sheet was measured by uniaxial Instron test to be about 23 GPa.

To create specimens with a sinusoidal boundary, an aluminum specimen mold with a surface including two wavelengths of a sinusoid (wavelength 25.4 mm, amplitude 2.0 mm) was designed and fabricated, Fig. 1A. Composite sheets with one flat surface and one non-flat surface matching the mold plate profile were fabricated using the same material and number of plies as above, Fig. 1B. The adherend's thickness varies along its length as the fibers get spread out or pressed together to follow the non-flat profile. Cutting strips of the resulting molded parts in in the center yields two different kinds of adherend: a ‘thin then thick’ type, and a ‘thick then thin’ type. Two adherends of the same type are needed to make a bond with sinusoidal interface.
This technique for making a sinusoidal surface allows the composite fibers to follow the surface profile and remain intact through the length of the sheet. Molding the shape instead of cutting avoids the material being potentially cracked, or excessively roughened, in a machining step. In a complementary study, we have used a similar molding technique to fabricate carbon fiber reinforced pyramidal truss structures. The fabricated trusses were shown to have superior strength and utilize the inherent strength of the carbon fibers. The fabricated structures also had much higher strength compared to pyramidal truss cores made by machining [9].

Whether flat or partially sinusoidal in the 25-mm bond region, the composite sheets were cut to blocks of 20 mm wide × 120 mm long, which were then bonded to create the single lap joints using EPON 828 epoxy resin (Miller Stephenson Inc.) with EPIKURE 3140 (Miller Stephenson Inc.) as the curing agent. Before bonding, the surface of the adherends was sand blasted and cleaned by acetone. (The elastic modulus of a thick bonded layer of adhesive was measured and found to be ~ 3.4 GPa which is in line with the manufacturer's value of 3.66). Before being tested, the bonded joints were left at room temperature for two days without pressure.

Figure 1C shows the schematic of a single lap bonded joint with a sine wave interface profile, where \( H \) is the adherend thickness, \( L \) is the lap joint length (equal to the sinusoidal wavelength ), and \( A \) is its amplitude. It is easily seen that the outer envelope for a bonded specimen with sinusoidal interface is identical to that with a flat interface.

The space for the adhesive is formed by separating the adherends a distance \( t = 0.254 \text{ mm} \) in the perpendicular direction, a distance which was controlled and kept constant during curing using brass shims. Note that when two perfectly conforming surfaces are separated in this way, the resulting gap width is not constant, but rather equals \( t / \sqrt{1 + \left(\frac{dy}{dx}\right)^2} \), where \( x \) is along the bond mean plane, and \( y \) is in the separation direction. The gap equals \( t \) only where slope \( dy/dx = 0 \), and otherwise it is less (for an amplitude of 1.8 and wavelength 25.4, the maximum slope is 0.44, and the gap there is approximately \( t/1.1 \)).

In this paper, a joint with a positive \( A/H \) ratio has an interface profile \( y = A \sin\left(\frac{\pi x}{L}\right) \), where \( x \) and \( y \) are measured from the left edge of the adhesive, as shown in Fig. 1C. For positive \( A/H \), the adherend is thinned close to its free end. A joint with a negative \( A/H \) ratio has
an interface profile \( y = -A \sin \left( \frac{\pi x}{L} \right) \), and of course \( A/H = 0 \) refers to a flat interface. All specimens fabricated in the experimental part of this study have a wavelength equal to the lap joint length, \( L = 25.4 \text{ mm} \). Figure 1D shows specimens with \( A/H = +0.4, -0.4 \) and 0.

An Instron universal tensile tester, model 5582, was used to obtain the force-displacement response of the bonded specimens. Aluminum tabs 25 mm long and with the same thickness as the adherends were attached at the end tips of the specimens, to place the mean bond plane on the load-cell centerline. The resulting center-point antisymmetry means that the load resultant transmitted by the bond has no moment about its midpoint. However, it does not mean that the transmitted force is parallel to the bond mean plane – that is only the case when pivoted grips are employed. With the fixed grips of this study, an unmeasured net tensile (peel) force of order 10% - 15% the shear force is anticipated to be imposed on the bond. The tests were carried out at a constant displacement rate of 5 mm/min. It should be noted that while the loading exerts predominantly a shear force on the overall bond, tensile (peel) stresses across the adhesive are typically very large in a small region near the bond end. That is because the adherend near the joint is subjected to a bending moment, whereas the end of the nearby mating adherend is moment-free. This would lead to curvature non-conformity at the joint edge, if extreme normal stresses in the adhesive did not enforce bending compatibility. Note that this concentrated peel stress is a problem caused by adherend flexibility, and is mitigated if the adherends are made relatively rigid.

Five specimens with \( A/H = 0 \) and five with \( A/H = -0.4 \) were tested, while for the \( A/H = +0.4 \) configuration seven specimens were tested. Fig. 2A shows examples of the measured force-displacement responses of the bonded joints with different interface profiles. Specimens with \( A/H = 0 \) and \( +0.4 \) have comparable initial stiffness, followed by an approximately linear force-displacement response up to the sudden failure of the joint. In contrast, the specimens with \( A/H = -0.4 \) generally showed a lower initial stiffness, with multiple cracking and sudden reductions in the load carrying capacity of the bonded joint prior to the final failure of the specimen. For this configuration, the profile of the load-displacement response varied from one specimen to another, and the partial drops in the response curves were observed at different displacements. However, the final failure loads of all specimens were comparable. Figure 2B shows the average values and standard deviations of the measured failure loads for each configuration. On average,
the specimens with $A/H = +0.4$ have a 40% higher failure load than the conventional flat joint configuration with the same lap length and adhesive thickness. The specimens with $A/H = -0.4$ were generally able to tolerate several stages of cracking, but had a significantly lower failure load than the other two configurations.

3. Finite element analysis of stress distribution

Two-dimensional models of single lap joint specimens with different interface configurations were performed using the commercial finite element package Abaqus (SIMULIA, Providence, RI). The models had an overlap length, adherend thickness and bond gap similar to the fabricated specimens in Section 2. Many used $A = 1.6$ mm and $H = 5.6$ mm, making $A/H = 0.28$. To model the Instron loading, the models were fixed at one end, while the other end was permitted only to translate axially without rotation, and an axial-direction load was applied. (Of course the total adherend loading at that end then includes a transverse force and a moment about the adherend centerline, due to the translational and rotational constraints. But overall antisymmetry means that the bond transmits no net moment.)

The adhesive was meshed with about ten elements through the thickness. Then the adherends were meshed with a pattern that became progressively coarser away from the adhesive. 4-node plane stress elements (type CPS4R) were used to mesh the models. The adherends and adhesives were modeled as isotropic linear elastic materials. The elastic modulus of the adherend and adhesive were taken as $E = 23.1$ GPa and $E_a = 3.66$ GPa, respectively, based on the experimental measurement presented in section 2. Poisson’s ratio for both adhesive and adherends was taken as 0.33. We also carried out a parametric study by varying $E_a$ in the range $0.183 \text{ GPa} \leq E_a \leq 3.66 \text{ GPa}$, to highlight the effect of the relative stiffness of the adhesive compared to the adherends.

The finite element calculation delivers an estimate of the stress distribution within the adhesive, denoted here in global coordinates by $(\sigma_x, \sigma_y, \tau_{xy})$. About 50 nodes approximately following the mid-line of the adhesive were used to report the stress state. Those midline adhesive stress components were used to calculate peeling and shear stresses, denoted by $\sigma_a$ and $\sigma_s$ [10]:

$$\sigma_a = \sigma_x \sin^2 \theta + \sigma_y \cos^2 \theta - 2 \tau_{xy} \sin \theta \cos \theta$$
\[ \sigma_z = -\left(\sigma_x - \sigma_y\right)\sin \theta \cos \theta + \tau_{xy} \left(\cos^2 \theta - \sin^2 \theta\right) \]

where \( \theta = \arctan(dy/dx) \) is the angle of the bonded joint interface. The calculated results of course do not include any effects of large deformation, material nonlinearity, or material and interface failure. They are used primarily for a qualitative ranking of the stress level created by different configurations.

Figure 4 shows the distribution of interface peeling and shear stresses along the joint midlines as a function of \( x \). Those stresses are normalized by the applied axial force divided by adherend area. These plotted quantities are symmetric about the center of the bonded joint (i.e. \( x/L = 0.5 \)) as expected. The maximum values of peel and shear occur at the edge of the joint for all configurations. The maximum values of the shear stresses are comparable for the three joint configurations. The maximum peeling stresses are generally much higher than the calculated maximum shear stresses, so whether the adhesive is either ductile or brittle, this suggests that the failure of the bonded joints is mainly due to the peeling stress and initiates from the bonded joint edge. For example, the maximum peeling stress for the bonded joint with \( A/H = +0.28 \) is approximately 50% higher than the maximum peeling stress in the flat joint, which is in good agreement with the ratio of the initial failure loads obtained in our experiments. In Figure 5, the distribution of normalized peeling and shear stresses at the joint interface is shown for the bonded joint with \( A/H = +0.28 \) with different values of adhesive elastic modulus, \( E_a \) (The adherend elastic modulus is \( E = 23.1 \) GPa in the calculations). A bonded joint with a compliant adhesive has a more uniform stress distribution in the bonded joint. The maximum values of the peeling and shear stress, which occur close to the bond edge, increase abruptly for bonded joints with large adhesive elastic modulus. However, the distribution of the stresses at the central part of the bonded joint is relatively independent of the adhesive stiffness. Figure 6 shows the maximum peeling stresses in the bonded joint for bonded joints with different \( A/H \) for two different elastic moduli of the adhesive. The maximum peeling stress decreases (in a relatively linear fashion) by increasing \( A/H \). This effect is stronger for bonded joints with a stiffer adhesive layer.

Figure 7 shows the shear and peeling stress distributions in bonded joints with \( A/H = +0.28 \) and different interface profiles with \( n=1, n=1.5 \) and \( n=2 \). In this set of calculations, the lap
joint length is kept constant, $L = 25.4$ mm, and the wavelengths of the joints are $L$, $L/1.5$ and $L/2$. For bonded joints with $n = 1$ and 2, the stress distributions in the joints are symmetric with respect to the center of the joint (i.e. $x/L = 0.5$). The bonded joint with $n = 2$ has a lower value of both maximum shear and peeling stresses. In contrast, the joint with $n = 1.5$ has asymmetric stress distributions. At the left edge of the adhesive (i.e. $x = 0$), the value of the peeling stress for this joint is between the calculated values of the peeling stress for the joints with $n = 1$ and $n = 2$. In contrast, at the right edge of the adhesive (i.e. $x = L$), the joints with $n = 1.5$ has much higher peeling stresses compared to the other two specimens. In fact, the maximum peeling stress in the joint with $n = 1.5$ and $A/H = +0.28$ has comparable values to the maximum peeling stress estimated for a joint with $n = 1$ and $A/H = -0.28$ – See Fig. 6. The results suggest that the slope of the interface profile at the two edges of the adhesive (i.e. $\theta = dy/dx$ at $x = 0$ and $x = L$) is the key factor governing the distribution of peeling stresses, and thus the overall strength. In contrast to the considerable differences observed in the peeling stress distributions, the shear stresses are comparable in all joints mentioned above. The direction of the shear stress changes in the central region of the specimen with $n = 1.5$ and $n = 2$, which was not observed for the specimen with $n = 1$. However, the adhesive in the central region of the adhesive layer generally contributes minimally to the overall behavior and strength of the bonded joints – as will be seen in the next section.

4. Effect of a central void in the bonded joint on the joint strength

In this section, we have studied the effect of a central void on the distribution of stresses in a single lap joint with a non-flat interface. For most adhesively bonded joints, a central void (of relatively large size compared the overall lap length) does not affect the distribution of stresses in the joint significantly and thus is perceived to have minimal effect on the bond structural integrity and mechanical response [11, 12]. This finding has implications in development of bonded joints with lower material cost and weight by removing the part of the adhesive [12-14]. Here, we studied the role of a central void on the distribution of stresses in a bonded joint with non-flat surfaces. Figure 8 shows the distribution of peeling and shear stresses along two bonded joints with $n = 1$ and $A/H = 0$ and $+0.32$ and different normalized void sizes. The normalized void size is denoted by $\gamma = L_1/L$, where $L_1$ is the length of the central void and $L$ is the overlap length. The distribution of the stresses in the bonded joint with a non-flat interface
exhibits lower sensitivity to the presence of the void compared to its flat joint counterpart. In general, the distribution of stresses in the bonded joint does vary significantly with the void size. We have carried out a similar set of analysis for bonded joints with a wide range of $A/H$. Figure 9 shows the dependence of the maximum normalized peeling stress on the interface profile normalized amplitude, $A/H$, for bonded joints with different void sized. For all bonded joints, a central void with $\gamma < 0.65$, has minimal effect on the maximum peeling stress and thus, the strength of the bonded joint. This observation is in agreement with Rossettos et al [14] and Lang et al [15] where a void size as large as 70% of the overlap length showed to have no significant effect on bonded joint strength.

5. Conclusions

Finite element analysis and experiments were used to evaluate the mechanical performance of adhesively bonded joints with a non-flat interface profile. In the experimental part of the study, single lap joints with three different interface profiles were created using a custom-made mold. The experimental results showed considerable differences between the uniaxial force-displacement of bonded joints with different interface profiles. The stress distributions in the bonded joints with the same configurations as the specimens tested in our experiments were calculated using the finite element method. The finite element results suggested that the failure of the specimens is mainly due to peeling. Further parametric studies were carried out to highlight the role of interface shape and adhesive stiffness, as well as a central void on the distribution of stresses in a bonded joint.

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References


Figures

Figure 1. A) Image of the custom-made steel mold. B) Composite plate after curing. C) Schematic of an adhesively bonded joint with a non-flat interface. D) Three specimen configurations manufactured and tested.

Figure 2. A) Uniaxial force-displacement response of the adhesively bonded joint with different interface profiles. B) Failure load for three configurations of the bonded joints under uniaxial tension. Error bar represents total spread of results.

Figure 3. Schematic of the finite element model and the applied boundary condition.

Figure 4. Distribution of (A) normalized peeling and (B) normalized shear stresses along the adhesive midline for three different interface configurations, $A/H = +0.28, 0, -0.28$.

Figure 5. Distribution of (A) normalized peeling and (B) normalized shear stresses along the adhesive midline for $A/H = +0.28$ and different adhesive elastic moduli.

Figure 6. Maximum normalized peeling stress in a bonded joint versus the normalized amplitude of the joint interface profile. The results are presented for bonded joints with two different adhesive elastic moduli.

Figure 7. Distribution of (A) normalized peeling and (B) normalized shear stresses along the joint for bonded joints with different number of wavelength and $A/H = +0.28$.

Figure 8. A) Distribution of normalized peeling and shear stress along a flat bonded joint with different central void sizes. $\gamma = (\text{void length}) / (\text{bond length})$. B) Similar as A for a bonded joint with $A/H = +0.32$.

Figure 9. Maximum normalized peeling stress in a bonded joint versus the normalized amplitude of the joint interface profile. The results are presented for bonded joints with different central void sizes.
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