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Proposal for a torsion shear fixture for bonded joints

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The aim of the present work is to design a simple torsion fixture for bonded joints that is adaptable to conventional testing machines and at the same time gives reliable results. This is achieved by transforming a traction or compression movement of a conventional machine to a torsion one on the bonded joints. In the present study, a full description of the mechanism of transferring a compression/traction movement to a pure torsion is explained, and the validation approach is outlined. Finally, experiments on bonded joints are performed and compared to step lap shear test. The results showed that the proposed torsion apparatus is very promising in measuring the strength of bonded joints.

1. Introduction

When measuring the shear strength of bonded joints, two fundamental requirements are needed, the simplicity of testing and reliability of the results. However, these requirements seem to be irreconcilable in bonded structures. The shear strengths measured using standard methods such as lap shear specimens give rise to non-uniform shear stresses that affect the reliability of the results, but can be performed easily using conventional standard testing machines. On the other hand, measuring the torsion shear strength requires particular testing apparatus, which is not available for all laboratories. In the following a brief review on adhesively bonded joint testing is given in order to appreciate the relevance of torsion testing of these joints.

The mechanical performance of adhesives is usually determined on bulk specimens or adhesively bonded joints, or both. Testing of bulk adhesives is easier than testing films because much larger deformations can be attained, and are much easier to measure. On the other hand, mechanical testing of adhesive bonded constructions is the most straightforward method to measure the strength of a joint. In the design and strength analysis of bonded joints, the major requirements are the elastic shear and tensile moduli. Also necessary are Poisson's ratio, ultimate shear strain, and ultimate shear stress. The difficulties in achieving reliable results on adhesive bonded joints is related to many factors, the most important being: specimens configuration, strain measurements, stress state to which the bondline is subjected and thickness of the bondline [1, 2].

There are four basic types of loading an adhesive joint: tensile, shear, peel and cleavage. Tensile shear strength is most widely adopted as a measure of the ultimate shear strength of an adhesive bond loaded in tension. In this type of loading, the forces act in the plane of the adhesive layer. The single lap joints are the most commonly used adhesive joints and have been the best studied so far. A review of the published literature related to the analysis of bonded joints suggests that there are three widely accepted methods for predicting the strength: closed form solutions [3], finite element method (FEM) [4, 5] and linear elastic fracture mechanics (LEFM) concept [6, 7]. More recently, a new approach based on an interface corner stress intensity factor has been investigated within the context of elasticity theory [8]. The precise analysis of simple lap is very complex. First, because the stress distribution along the overlap length and through the thickness is non uniform [4, 9–16]. In addition there are interactions between each of the three potential failure modes: adherend yielding, adhesive peel, and adhesive shear. Consequently, the joint strength reliability is affected. Another factor affecting the simple lap joint strength is the existence of a bending moment arising from the two non collinear forces, in addition to the in plane tension. The use of step lap joint is an alternative test method of the single lap joint, in which the undesirable bending effect is avoided. However, the stress distributions are almost unchanged [17, 18].

In terms of load transfer between two members, double lap joints are probably the most desirable joints. The loads in the adherend parts can be in tension, compression, or in plane shear. However, there is a further difficulty of making two uniform bondlines of the same thickness [19].

Peeling tests are the most selective for measuring the surface treatment quality differences. Uniformity of peel strength values for a given adhesive in contrast to lap shear strength made the peel test attractive to measure the properties of adhesive bonded joints. Various forms of peel tests are used to assess the performance of structural adhesives. In fact, this form of test deliberately stresses the adhesive in a very small region, subjecting it to a large tensile stress, although a complex stress situation is usually present. This situation is not easily assessed, and peel is normally used to compare adhesives rather than to measure their properties [18, 20].

Cleavage strength is very rarely quoted in reference works. In this type of test severe localized loading occurs on one side of the joint, while the other side is merely loaded. This type of test is usually performed to investigate the ability of the bonded joint to withstand further processing after assembly [21]. Axisymmetric joints also with square or rectangular adherends, are widely used specimens for testing the response of adhesives to shear, tensile and compressive stresses. Butt joints or napkin ring, as specified by ASTM E229, can provide an apparently convenient means for determining the mechanical properties of structural adhesives. The test is used to measure the moduli of rigidity, elasticity and Poisson's ratio [22]. The advantage of using butt joints is that the adhesive is tested in the thin film form as used in most joints, thus overcoming any possible objection to bulk specimen. Although the stress distribution is simple, end effects are once again a problem. As stated by Adams and Wake [4], "if joints are to be loaded to failure and if the failure stress is to mean anything, then it must be true stress and not a convenient but misleading approximation."

It should be mentioned also that in practice, most structural adhesives exhibit considerable plastic deformation when subjected to shear stress, and it is quite probable that the presence of a spew fillet will yield without causing premature failure of the joint when loaded in torsion and this would lead to an overestimation of the shear modulus of the adhesive [23, 24].

Finally, one should mention the torsion shear fixture needed to investigate the mechanical properties of adhesive bonded joints. Historically, the torsion apparatus has been developed by Bossler *et al.* in 1968 [22]. Other designs have been also proposed in ASTM known as ASTM E229 [25]. The torsion shear apparatus is designed to ensure that no bending or cleavage loads are imposed on the napkin ring specimens that must be subjected to a pure shear.

The reason that, the shear testing of adhesive bonded joint is mainly performed under tension or compression loading of either single or step lap joints is due to the fact that this approach meets the requirement of ease of testing. However, the reliability of the tests is questionable because of the non-uniform stress distribution which affect the shear strength of the bonded joints. Indeed, non-uniform stress distribution create stress concentration that are not taken into account when estimating the shear strength from these tests. On the other hand, the torsion testing method is known to involve less stress concentration and consequently, is considered to be more reliable as resported in [22]. However, the difficulties associated to the torsion testing method has prevented its widespread popularity. The main feature of the proposed torsion apparatus in this paper is its simplicity, moderate cost and adaptability to universal testing machines.

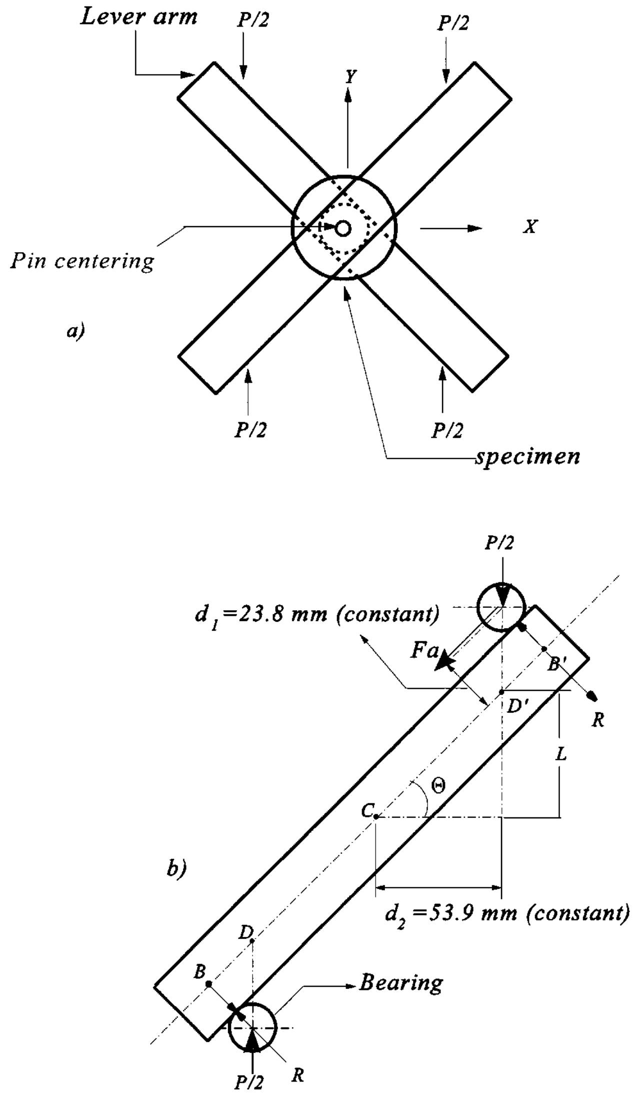

Figure 1 Diagram showing the mechanism of the proposed torsion fixture.

2. Proposal of a torsion fixture

Fig. 1 shows the mechanism of the proposed torsion fixture. This fixture is designed to be fitted to any uniaxial testing machine with the aid of some clamping accessories (Fig. 2). The mechanism to create a torsion loading on a bulk specimen or a bonded joint is based on the torque generated by two lever arms in a cross configuration moving in opposite rotational directions. The rotation of the lever arms is provided by the axial displacement of the cross-head of the uniaxial testing machine. In order to easily describe the operation of the proposed fixture, let take the specimen configuration shown in Fig. 3 and consider the measurement of its shear stress-shear strain response. The specimen is inserted between the two lever arms through cross grooves and a centering pin as shown in Fig. 1a. To accomplish a perfect rotation movement, the lever arms are laid on needle bearings hold by lower and upper shafts (Fig. 2). The shafts came with a lower and upper

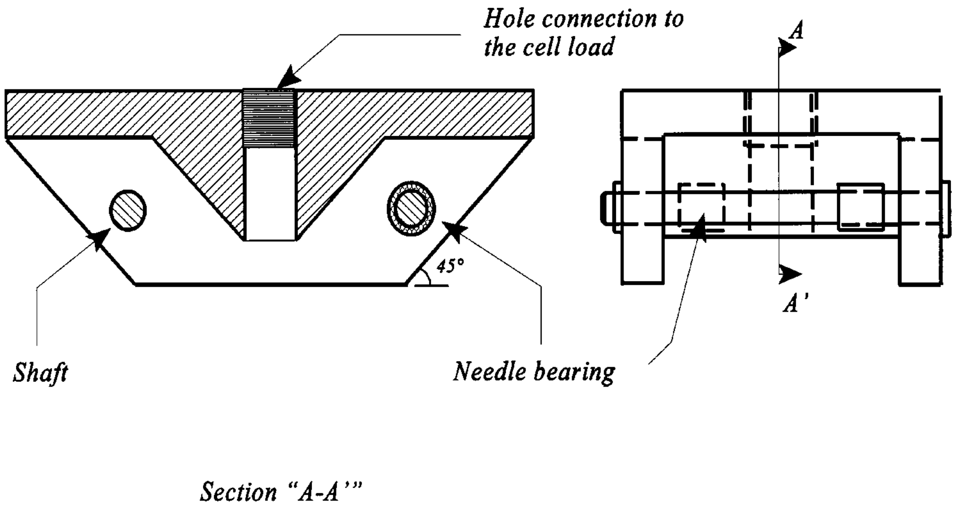

Figure 2 Upper steel frame diagram.

steel frames that can be connected to an uniaxial testing machine. Moreover, clamping accessories are designed to hold the upper and lower frames aligned and at the same time keep the frames from rotation during loading. In Fig. 1b a detailed mechanism of equilibrium forces acting on the system is shown. The force applied by the uniaxial machine is transferred to the needle bearings supported by the steel shafts and give rise to two concurrent forces:

$$\begin{aligned} R &= P/2\cos\theta \\ F_a &= \mu(P/2\sin\theta) \end{aligned} \tag{1}$$

where μ is the coefficient of friction, $P/2$ is the half force applied by the machine and θ is the angle between the segment BC and the horizontal. To ensure pure torsion and avoid bending and cleavage, the axial force, F_a, should be negligible. Since the needle bearing is free to displace on the steel shaft and free to rotate, this requirement is satisfied, therefore, we can write the following:

$$R \ggg F_a \leftrightharpoons \mu \lll 1 \tag{2}$$

The determination of the torsion moment is easily obtained from the diagram forces acting on the lever arm of the torsion apparatus shown in Fig. 1b. The half distance of the lever arm is represented by the segment BC and is given by:

$$BC = \left(\frac{d_1}{\cos\theta}\right) + (d_2 \cdot \mathrm{tg}\theta) \tag{3}$$

Since the length L decreases during the experiment, the

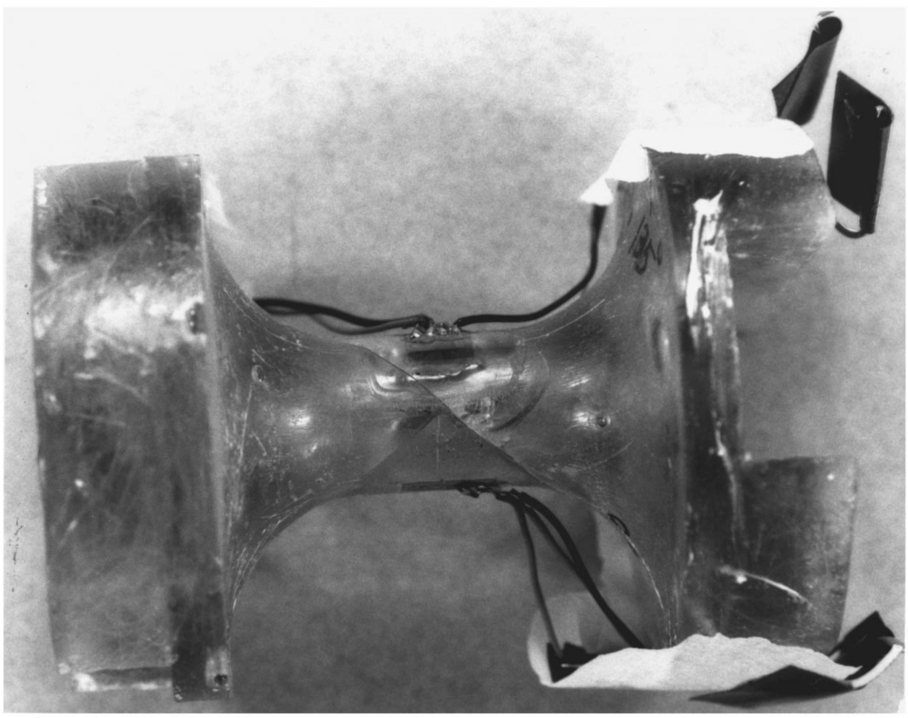

Figure 3 A photograph showing the strain gages mounted on the validation specimen also show a 45° failure direction.

angle θ then should be expressed by the following:

$$\theta = \text{arctg}\left(\frac{L}{d_1}\right) \tag{4}$$

$$L = d_1 - (d/2) \tag{5}$$

where d is the displacement of the head of the machine. Hence, the torsion moment M induced by the two lever arms is expressed by:

$$M = 4R \cdot BC \tag{6}$$

Therefore, the maximum shear stress, acting on the extreme edge radii of the validation specimen shown in Fig. 3 is obtained from the following relationship:

$$\tau_{r\text{o}\max} = \frac{Mr_\text{o}}{J} = \frac{Mr_\text{o}}{\pi\left(r_\text{o}^4 - r_\text{i}^4\right)/2} \tag{7}$$

where r_o is the outer radius of the necked region of the validation specimen, r_i is the inner radius and J is the polar inertia moment.

The shear modulus is then given by Equation 8 and is determined by the secant modulus method.

$$G = \frac{\tau_{r\text{o}}}{\gamma_{r\theta}} \tag{8}$$

$\gamma_{\text{r}\theta}$ is measured by the strain gages response.

It should be noticed that the condition of thin walled torsion specimen are not encountered for the validation specimen. However, this will be the case for bonded joint in Section 5. In other terms, if extraneous stresses are to be induced in the torsion region, they would be amplified in the validation specimen compared to the napking ring joints in which this fixture is intended.

Experimental tests with the proposed torsion fixture were performed using a cross head speed of 5 mm/min. Actually, it is more relevant to consider the angular speed rather than the cross head speed. To achieve this objective, the following expression has been established on the basis of kinematic relationships.

$$\omega = \frac{V_\text{m} \cdot \cos^2\theta}{d} \tag{9}$$

where ω is the rotational speed and V_m is the cross head speed of the machine.

The recommended time to failure of the napkin ring specimens given by ASTM E 229 should be between 2 and 5 min. In the present study, the rotational speed was almost constant during the 4 min of the test experiment (0.047 rad/min at the start of the experiment to 0.055 rad/min at the failure of the specimen).

3. Experimental validation of the proposed torsion fixture

Even though the preceeding analsis predicts a quasi-pure torsion loading, extraneous stress components can be generated by the loading mechanism and inflect the results reliability. The validation procedure is intended to verify if the proposed torsion fixture induces extraneous stresses other than pure torsion. Probably, the best validation approach is to compare results from the proposed fixture to those obtained on proven standard equipment such as that proposed by Bossler or ASTM E229 [22, 25]. Unfortunately, such equipment is not available and an alternative validation approach is proposed. First as shown in Fig. 3, a specimen configuration with appropriately mounted strain gages capable of detecting the principal strain components during loading is proposed. This permits the measurement of axial, transverse and shear strains. The magnitude of these strains components will show to what extent extraneous stresses, other than torsion, are present. From the same test, the shear modulus G_A can be determined. Secondly, independent standard tensile tests will be performed on the same material. Strain gages for the tensile tests will be used in order to determine the shear modulus of the material using the Timoshenko formula $G_\text{T} = (E/2(1+\nu))$. If the proposed torsion test is reliable, G_A and G_T should be comparable.

The specimens used to validate the proposed torsion apparatus were casted in a steel mold laboratory made to fit the lever arms in a cross shape form (Fig. 1). The material used is a general purpose thermoset vinylester resin, the same one used for the standard tensile specimens. To avoid subjection of the validation specimen to any stress concentration especially at the extreme edges, it was decided to reinforce this part with chopped fibers (Fig. 3). As stated earlier, to accomplish the validation two types of strain gages are used. First, strain gage rosettes (0°/90°) are bonded onto the validation specimen along the y and z directions. In addition (±45°) strain gage rosettes were also used to measure the shear strain in the main direction of the specimen (z). In all the experiments, three specimens were tested to detect the bending in the yz plane and other three to detect the bending in the xz plane.

3.5. Results and discussion

Fig. 4 shows a finite element computation of a typical validation specimen. It is clearly seen that any applied force P on the lever arm induces a stress concentration on the necked region of the specimen. Mechanical tests performed on specimens show that all specimens present a typical torsion failure at a 45° direction (Fig. 3).

The results of the torsion test are shown in Figs 5 and 6. Both figures illustrate the extraneous strains in xz and yz planes. Since only the elastic domain is of interest, the shear strains greater than 1% were not taken into consideration. Table I summarizes the results of both the torsion and tensile results.

It can be seen from the Figs 5 and 6 that the axial and transverse strains measured up to $\gamma_{\text{r}\theta}$ of 1% are negligible. This represent, in the extreme case, 3% of the shear strain in the xz plane and 4% in the yz plane respectively. A careful analysis of the experiment shows that most of the contribution to the experimental error came from the instabilities in the Wheatstone bridge.

TABLE I Results of the torsion and tensile tests

Tests	Ultimate strength $\tau_{r\theta}$ (MPa) or σ_{ult} (MPa)	Tensile modulus E (GPa)	Poisson's ratio ν	Shear modulus G_T or G_A (MPa)
Torsion test	32.0 ± 8.1	—	—	958 ± 39
Tensile test	43.5 ± 6.7	2.9 ± 0.2	0.40 ± 0.02	1059 ± 60

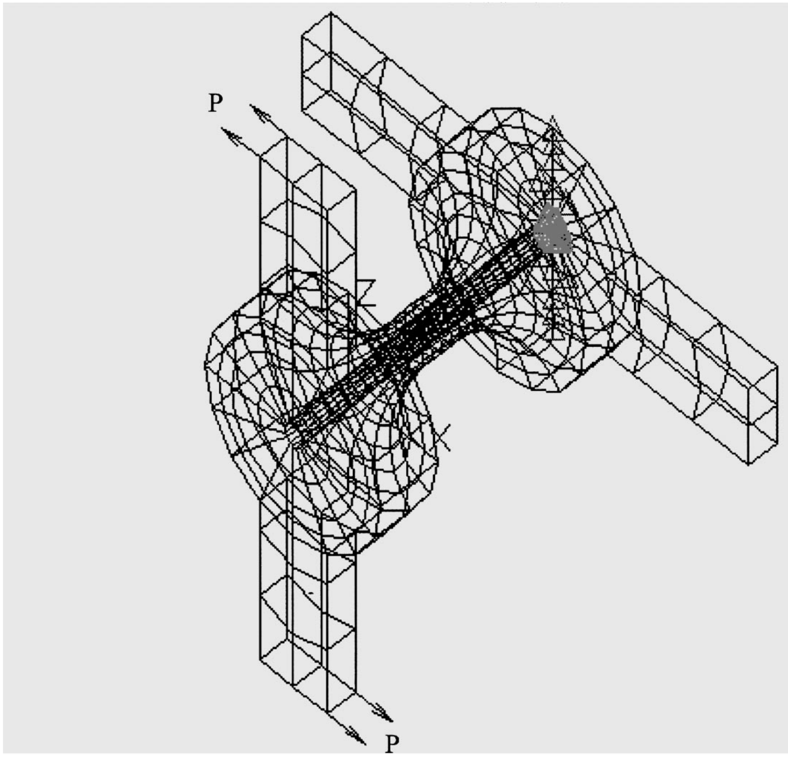

Figure 4 Finite element model of the validation specimen.

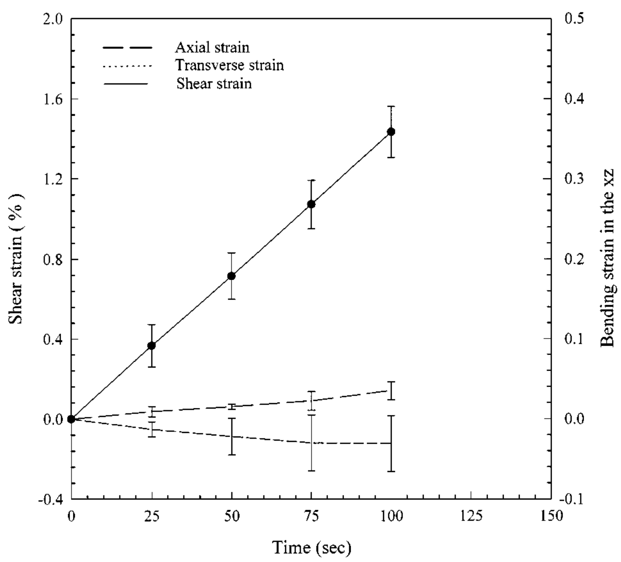

Figure 5 Axial, transverse and shear strains as a function of time (bending in the *xz* plane).

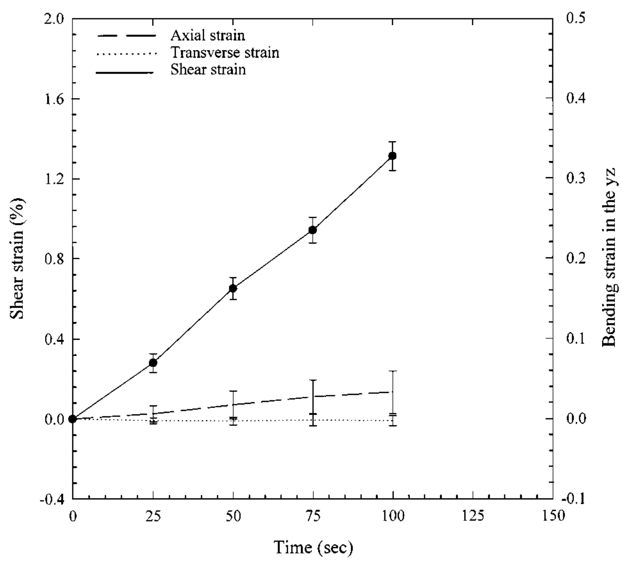

Figure 6 Axial, transverse and shear stains as a function of time (bending in the *yz* plane).

On the other hand, the lower and upper frame alignment can be improved so that the extraneous stress effects are no more exit or at least very negligible. The specimen porosity observed during preparation may also interfere in the scattering shown in the results.

It is shown from the Figs 7 and 8 that the vinylester is relatively a brittle material and has a relatively linear behavior up to the nominal axial strain of 1%. Viscoelastic behavior was more evident at higher strains approaching the limit of failure.

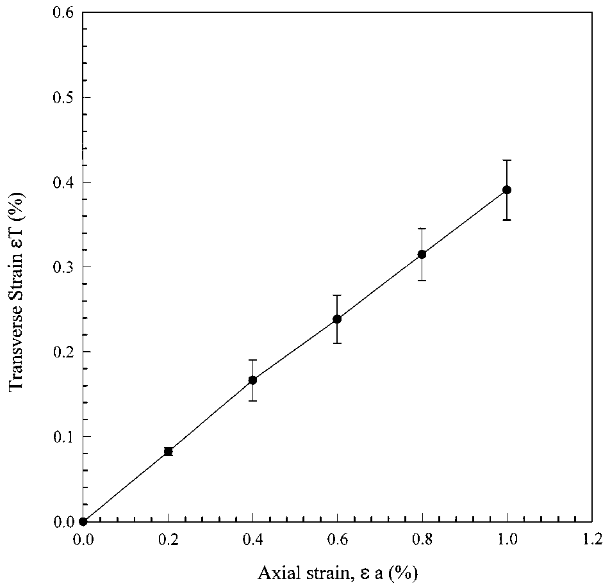

Figure 7 Transverse strain vs. axial strain curve.

The shear modulus obtained in both experiments showed a good agreement, the average values thus obtained of 1059 MPa for the tensile test and 958 MPa for the torsion test are of the same magnitude. A relatively small error of 9% mainly caused by the non recorded friction loses of the forces arising in the needle bearings and the accessory block alignments is quite acceptable. Equally, the undesirable bending effects, even as small as recorded may contribute to this difference. Although, the small values of the standard deviation for the torsion experiments show a consistency in the obtained results.

As a summary, the results obtained in the both tensile and torsion experiments on the validation specimens are in agreement. The fact that a small variation of 9% difference in shear modulus is found between both experimental tests suggests that the proposed torsion apparatus is very promising. It is obvious that the small extraneous bending stresses recorded in the case of the validation specimens would vanish almost completely for the type of napkin ring specimens mainly due the small region of the adhesive thickness and the larger aspect ratio of the ring. Moreover, a minor modification is done on the lever arms which are brought to a round shaped form to minimize the contact and thus minimizing the bending stresses.

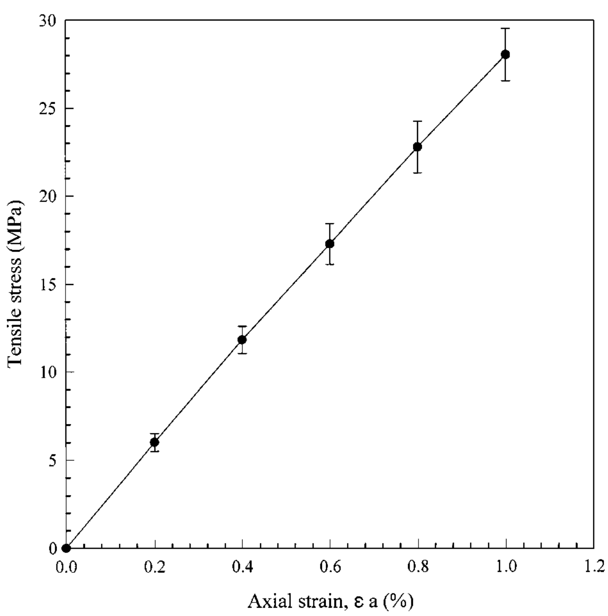

Figure 8 Tensile stress vs. strain curve.

3.6. Application to the case of bonded joints

3.6.1. Materials

The materials used in the present study are an epoxy resin of a chemically pure bisphenol A diglycidyl ether (DGEBA) (DER 331) from Dow chemicals, a polyamide resin (V125) from Henkel Co. as an amine hardener and a tertiary amine (2,4,6-diphenyl amine) Epi-cure 3253 from Shell chemicals as a catalyst. Two types of fillers are used, silane treated glass beads CP3003 with a diameter less than 45 μm, and ceramic filler (pure alumina powder) with a diameter less than 100 μm. Also are used aluminum T6061-T6 plates and napkin ring as adherents.

3.6.2. Specimens preparation

The surface treatment of the aluminum plates is performed according to a recommended ASTM D2651 procedure (hot etching mixture of dichromate-sulfuric acid, referred as FPL etch). In performing the surface treatment, care should be made in order to obtain uniform results. It was reported that variation of the surface treatment parameters has a drastic effect on shear and peel strength properties [26]. The adhesive composites are prepared by mixing the epoxy resin, curing agent, catalyst and fillers. The mixture is stirred completely and placed into the plates to be joined. It has to be mentioned that because of the high viscosity of the adhesives, no air bubble removal is undertaken. The adhesive thickness between the two plates of the step lap shear samples was 0.2 mm and this is done by placing a thin shim of steel at both ends of the plate as shown in Fig. 9. To prevent any excess epoxy adhesive from sticking on the edges of the specimen (spew fillet), a 3 mm free space is coated with grease and removed before curing takes place. On the other hand, napkin ring specimens as shown in Fig. 10 are also joined with the same mixtures under the same conditions.

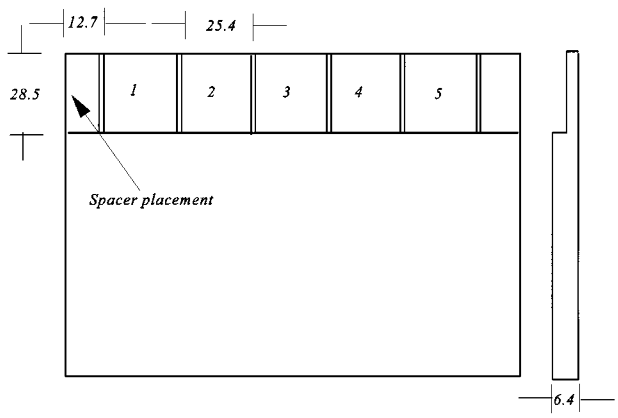

Figure 9 Step lap specimens (All dimension are in mm).

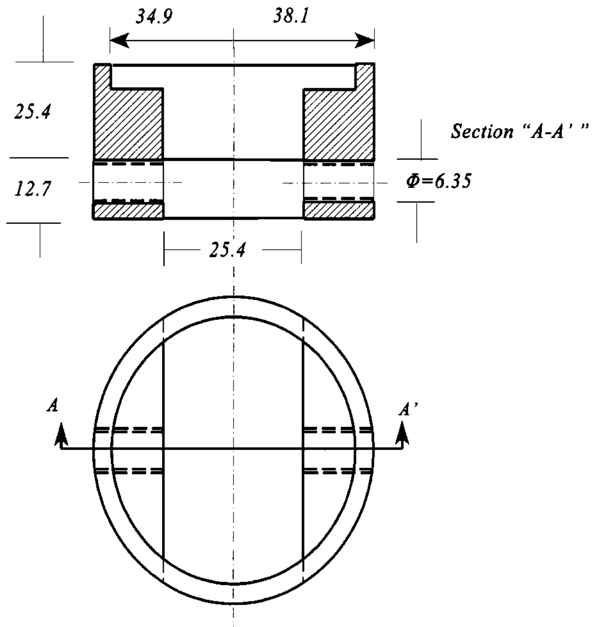

Figure 10 Napking ring specimen (All dimension are in mm).

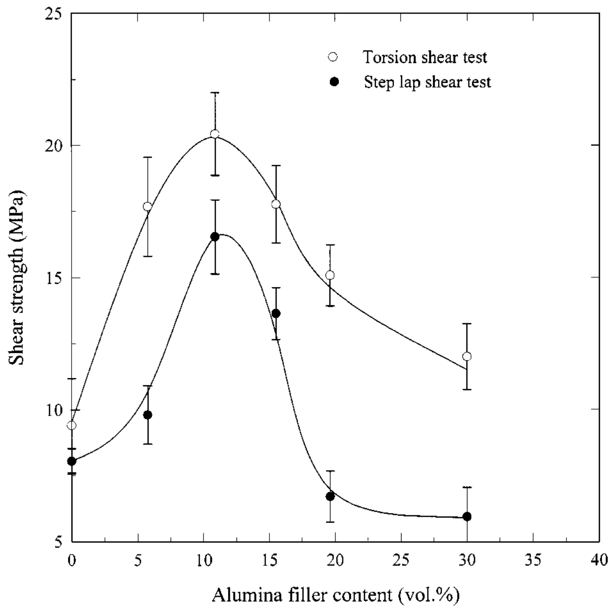

Figure 11 Shear of strength as function of filler content in the case of ACP.

3.6.3. Shear test procedures

Step lap shear specimens inspired by ASTM D1002 are hold to grips of an universal testing machine MTS 810 coupled to a data acquisition system. A 1.3 mm/min cross head speed is set for all the experiments. At least five specimens were tested for reproducibility. Similarly, napkin ring samples are also tested at a 5 mm/min cross head speed with the proposed laboratory torsion fixture. It should be noted that the above analysis of Section 3.3 is also valid in the case of bonded joint. Furthermore, the shear strain $\gamma_{r\theta}$ can be expressed as follow:

$$\gamma_{r\theta} = \frac{r\, d\theta}{2e} \tag{10}$$

where $d\theta$ is the angular displacement of the two adherent surfaces, e is the adhesive thickness, r is radial distance to the center line of the joint.

3.7. Bonded joint results and discussion

A more detailed analysis of the effect of filler content on the mechanical properties of adhesive bonded joints is given in the second part of this research work [27]. Concisely, this part of work is intended to verify the reliability of the proposed torsion apparatus. Figs 11 and 12 compare the two types of experiments on samples of various filler type and filler content. It is shown from these figures that the general trend of the shear strength in both experiments increases with filler content, reaches a maximum and then starts to decrease. It is clear that the shear strength measured with the proposed torsion shear fixture is higher than that measured with the conventional step lap specimen. It is well known that step lap joints produce underestimated joint strengths [10]. This is confirmed by the results of Figs 11 and 12 which at the same time confirms the reliability of the proposed torsion fixture. It should be noted that Nayeb-Hashemi *et al.* [28] have reported similar trends as in Fig. 12. Their results show that an almost 50% higher shear stress is required at the interface for bond failure under torsion, than under uniaxial load. It should be noticed that reference [28] deals with the torsion of tubular adhesively bonded joints instead of annular joint of Fig. 12. The two tests are different since they do not involve the same stress state. Actually, the proposed torsion fixture can perform the kind of tests mentionned in [28].

Additional evidence of the reliability of the torsion measurements made with the proposed torsion fixture is given by comparing the relative modulus (G'_c/G'_a), where G is the shear modulus and the subscripts c and a represent filled adhesive and matrix respectively; measured on the napking ring specimen and the relative modulus measured on the bulk materials by DMA. The

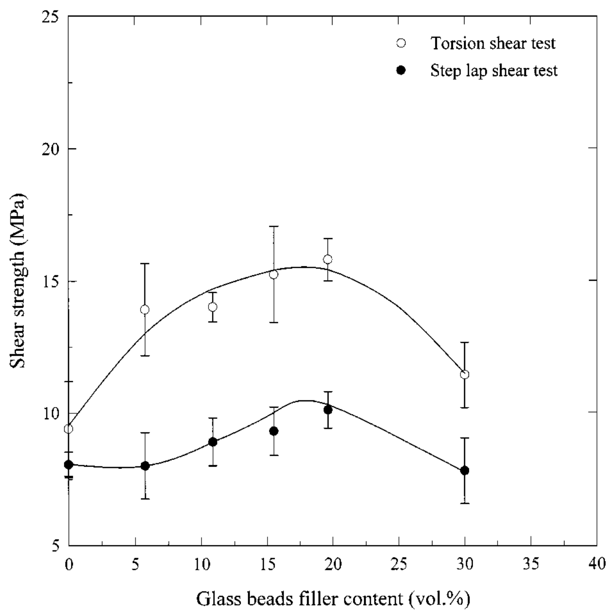

Figure 12 Shear strength as function of filler content in the case of glass beads.

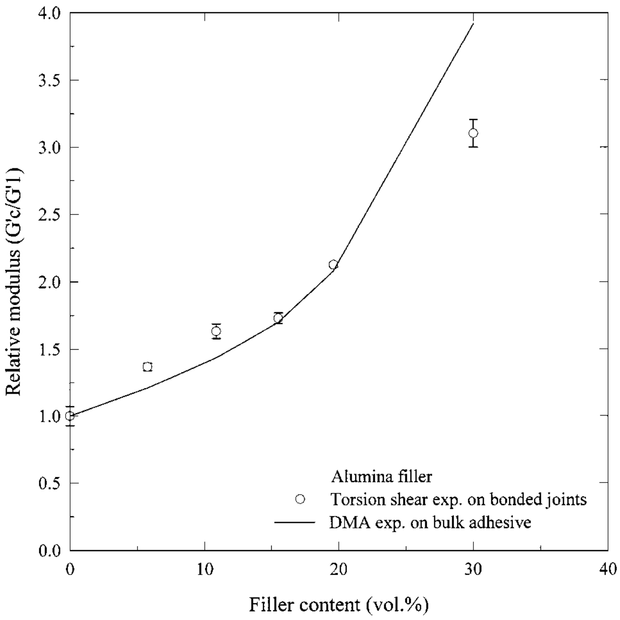

Figure 13 Relative modulus as function of filler content in the case of ACP.

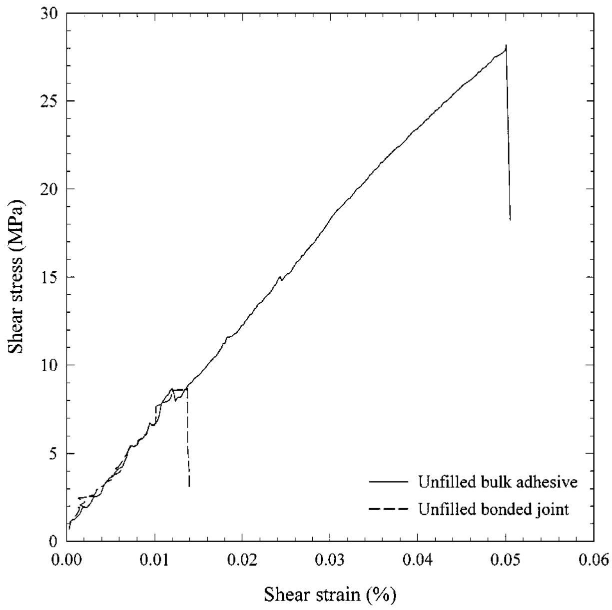

Figure 15 Torsion shear stress versus shear strain (comparison between bulk adhesive and bonded joint).

results are shown in Figs 13 and 14. Even though, no extensometers were used to measure with accuracy the shear strain of the adhesive bonded joints, a good agreement between both results is found. If the fracture has to occur in the adhesive i.e., 100% cohesive failure, the strength of the joint can be predicted easily. However, if this is not the case, a comparison of the shear strength of both bulk adhesive and bonded joint (Fig. 15), it is clearly seen that the fracture occurred earlier in the bonded joint, resulting in lower strength. Whereas, a good agreement of the shear modulus is observed between both results. Moreover the torsion shear results showed that the epoxy based adhesives as illustrated in Figs 16 and 17 are relatively brittle materials. They showed a relatively linear behavior up to fracture.

4. Conclusion

As a summary, numerous mechanical testing methods have been designed to evaluate specific mechanical properties of adhesive bonded joint. The aim of the present study was to design a simple torsion fixture for bonded joints that is adaptable to conventional testing machines and at the same time gives reliable results. It is convincing from the results of the validation work that the proposed torsion apparatus is quite acceptable for measuring the shear properties of bonded joints. Taking into consideration the small inconveniences encountered during the experimental work, it is recommended to improve the block alignment for ease of testing. Regardless of the above mentioned statements, it is shown that the torsion shear strength results are more reliable than those obtained by the standard lap shear test. The

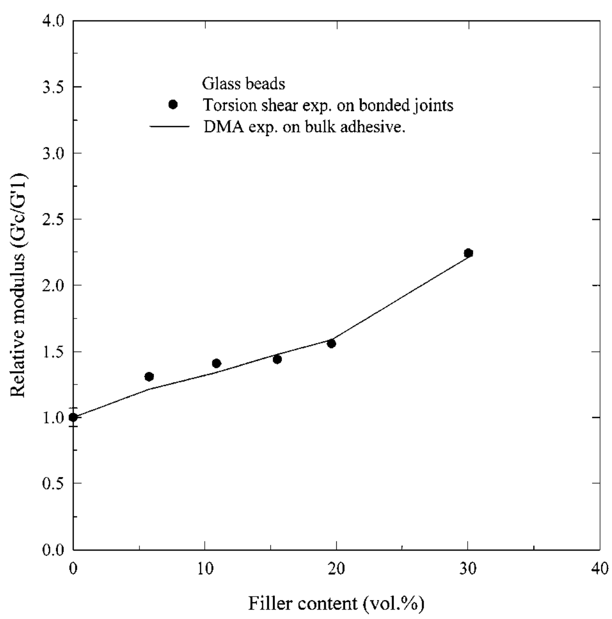

Figure 14 Relative modulus as function of filler content in the case of glass beads.

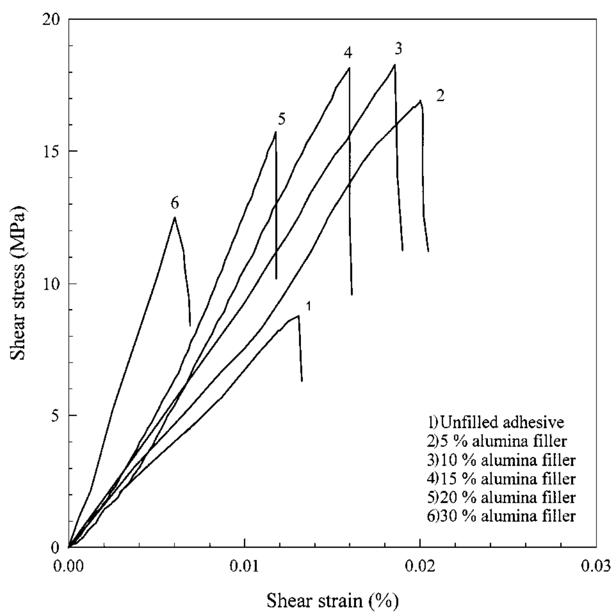

Figure 16 Torsion shear stress vs. shear strain in the case of ACP.

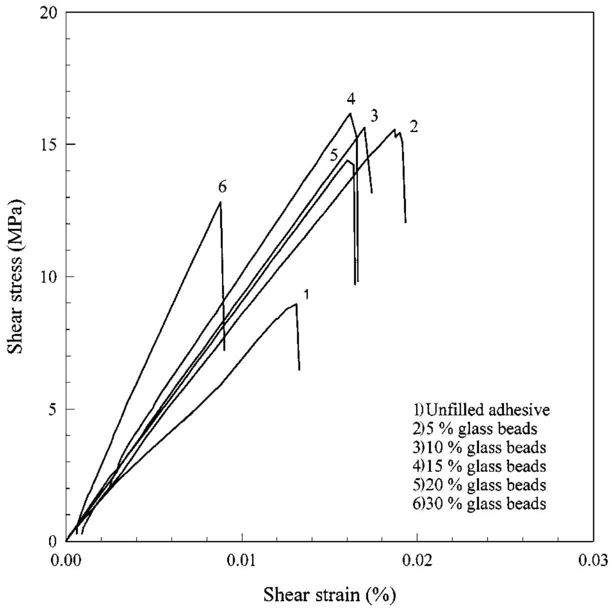

Figure 17 Torsion shear stress vs. shear strain in the case of glass beads.

improvement is mainly due to the fact that step lap test induces non uniform stresses that affect the reliability of the results contrary to torsion shear experiments.

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