NPL Manual

Design and Testing of Bonded and Bolted Joints

This Electronic Guide was produced as part of the Measurements for Materials System Programme on Design for Fatigue and Creep in Joined Systems

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Introduction

This document addresses design and testing issues that affect the strength, stiffness and life expectancy of bonded and bolted joints under quasi-static (monotonic), cyclic and creep loading conditions. Factors, such as specimen geometry, material properties, processing variables and surface treatment are considered in addition to test parameters (i.e. loading mode and alignment) and bolt parameters (i.e. torque or clamping force, and single- and multi-bolt arrays). It includes advice on data requirements and test methods required to generate design data, and provides advice on design, testing and manufacture of bonded and bolted structures. A list of recommended test methods for determining input data for the design and analysis of bonded and bolted joints is provided in Appendix 1 (see also [15, 20, 22]). A number of joint geometries, including butt-tension, single-lap, scarf., t-peel and T-joints are used to demonstrate various aspects associated with design, fabrication and testing of joined systems.

The guide is primarily concerned with metals and fibre-reinforced polymer composites either bonded with structural adhesives (i.e. rubber-toughened epoxies) or bolted. The document presents models for determining deformation and failure behaviour of rubber-toughened epoxy adhesives. Quasi-static, cyclic and creep loading are covered.

The intention of the guide is to provide designers and users with sufficient information which, when coupled with their own expertise, can be used to reliably assess the key parameters that affect the performance of adhesive joints. If the intention is to generate design data, then the guide should be used in conjunction with the appropriate structural design codes. The guide assumes some basic knowledge of the materials and mechanical engineering, and is not intended as a textbook or as a design protocol. There are a number of published works, which provide a comprehensive coverage of adhesive technology and preliminary design [1-17]. Other NPL Measurement Good Practice Guides "Preparation and Testing of Bulk Specimens of Adhesives", "Thermal Analysis Techniques for Composites and Adhesives", "Fibre Reinforced Plastic Composites - Machining of Composites and Specimen Preparation", "Characterisation of Flexible Adhesives for Design", "Preparation and Testing of Adhesive Joints" and "The Use of Finite Element Methods for Design with Adhesives" [18-23], provide advice on issues relating to the preparation and testing of bulk adhesive and adhesive joint specimens, acquisition of design data from bulk specimens, finite element modelling of adhesives, flexible adhesives and durability testing. The intention of the guide is to complement these published works.

It is recommended that specialist advice be sought from adhesive manufacturers on adhesive selection, use of associated technologies and health and safety requirements. Organisations that can provide specialist advice are listed at the back of the guide along with relevant standards and publications. Expert advice should be obtained from the adhesive manufacturer on selection and use of surface treatments and that the detail requirements specified by the manufacturer are completely satisfied. Where tests are performed to characterise the adhesive material then it is recommended that the surface preparation is as good as possible to minimise premature adhesion failure. Where tests are performed to evaluate a bonding system then the surface preparation procedures for test specimens will need to mirror those for the final bonded component.

Glossary of Terms (Based on BSI and ASTM definitions)

Accelerated ageing test: Short-term test designed to simulate the effects of longer-term service conditions.
Adherend: Body that is or intended to be held to another body by an adhesive.
Adherend failure: Failure of a joint in the body of the adherend.
Adhesion: State in which two surfaces are held together by interfacial bonds.
Adhesive: Non-metallic substance capable of joining materials by surface bonding (adhesion), the bonding possessing adequate internal strength (cohesion).
Adhesive failure: Failure of an adhesive bond, such that separation appears to be at the adhesive/adherend interface.
ASTM: American Society for Testing and Materials.
Bond: The union of materials by adhesives.
Bond-line: The layer of adhesive, which attaches two adherends.
Bond strength: The unit of load applied to tension, compression, flexure, peel, impact, cleavage, or shear, required to break an adhesive assembly with failure occurring in or near the plane of the bond.
BSI: British Standards Institute
Butt joint: Joint in which the plane of the bond is at right angles to a major axis of the adherends.
Bulk adhesive: The adhesive unaltered by the adherend.
Cleavage: Mode of application of a force to a joint between rigid adherends, which is not uniform over the whole area, but results in a stress concentrated at one edge.
Cohesion: The ability of the adhesive to resist splitting or rupture.
Cohesive failure: Failure within the body of the adhesive (i.e. not at the interface).
Creep: The time-dependent increase in strain resulting from a sustained load.
Cure: To set or harden by means of a chemical reaction.
Cure time: Time required to affect a cure at a given temperature.
Double lap joint: Joint made by placing one or two adherends partly over one or two other adherends and bonding together the overlapped portions.
Durability: The endurance of joint strength relative to the required service conditions.
Environmental test: Test to assess the performance of an assembly under service conditions.
Exothermic: A chemical reaction that emits heat.
Fatigue life: Number of cycles necessary to bring an adhesive bond to the point of failure when the bond is subjected to repeated cyclic stressing under specified conditions.
Fatigue strength: Force that a joint will withstand when the force is applied repeatedly for an infinite number of cycles.
Fillet: Portion of an adhesive that bridges the adherends outside the bond-line.
Gel: A semi-solid system consisting of a network of solid aggregates in which liquid is held.
Gelation: Formation of a gel.
Glass transition: A reversible change in an amorphous polymer or in amorphous regions of a partially crystalline polymer from (or to) a viscous or rubbery condition to (or from) a hard and relatively brittle one.

Glossary continued…..

National Physical Laboratory

Hygroscopic: Material capable of absorbing and retaining environmental moisture.
ISO: International Standards Organisation.
Lap joint: Joint made by placing one adherend partly over another and bonding together the overlapped portions.
Open time: Time interval from when an adhesive is applied to when the material becomes unworkable.
Peel: Mode of application of a force to a joint in which one or both of the adherends is flexible and which the stress is concentrated at a boundary.
Peel ply: A layer of resin free material used to protect a laminate for later secondary bonding.
Plasticisation: Increase in softness, flexibility, and extensibility of an adhesive.
Post-cure: Further treatment by time and/or temperature of an adhesive to obtain the required properties by curing.
Porosity: A condition of trapped pockets of air, gas or vacuum within a solid material.
Primer: A coating applied to a surface, prior to the application of an adhesive, to improve the performance of the bond.
Scarf joint: Joint made by cutting identical angular segments at an angle less than 45° to the major axis of two adherends and bonding the adherends with the cut areas fitted together to be coplanar.
Service life (N): Number of stress cycles applied to a specimen until it has reached the chosen end of the test.
Shear: Mode of application of a force to a joint that acts in the plane of the bond.
Shelf life: The period for which the components of the adhesive may be stored, under the conditions specified by the manufacturer, without being degraded.
Strain: Unit change due to force in size of body relative to its original size.
Stress: Force exerted per unit area at a point within a plane.
Stress-cycles (SN) curve: Curve, allowing the resistance of the material to be seen, which indicates the relationship observed experimentally between the service life N and maximum stress.
Stress-strain diagram (or curve): A diagram in which corresponding values of stress and strain are plotted against each other.
Structural bond: A bond, which is capable of sustaining in a structure a specified strength level under a combination of stresses for a specified time.
Substrate: An adherend, a material upon which an adhesive is applied.
Surface preparation (or treatment): Physical and/or chemical treatments applied to adherends to render them suitable or more suitable for adhesive bonding.
Tack: The property of an adhesive that enables it to form a bond of measurable strength immediately after adhesive and adherend are brought into contact under low pressure.
Tension: Mode of application of a tensile force normal to the plane of a joint between rigid adherends and uniformly distributed over the whole area of the bond-line.
Thermoset: A resin that is substantially infusible and insoluble after being cured.
Thermoplastic: A material that can be repeatedly softened by heating.
Traveller: A test specimen used for example to measure moisture content as a result of conditioning.
Viscosity: Resistance of a liquid material to flow.
Wet strength: Strength of an adhesive bond determined immediately after removal from a liquid in which it has immersed under specified conditions.
Wetting: A surface is considered completely wet by a liquid if the contact angle is zero, and incompletely wet if the contact angle has a finite value.
Yield stress: The stress (either normal or shear) at which a marked increase in deformation occurs without an increase in load.
Yield strain: The strain, below which a material acts in an elastic manner, and above which it begins to exhibit permanent deformation.

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Design Requirements for Bonded Joints

This section covers the various aspects that need to be considered in the design of adhesive joints. These include: joint geometry, adherend and adhesive properties, and mode of loading (i.e. monotonic, cyclic and creep loading). A number of examples in the form of different joint geometries will be used to demonstrate the various effects that geometric and material parameters have on joint performance.

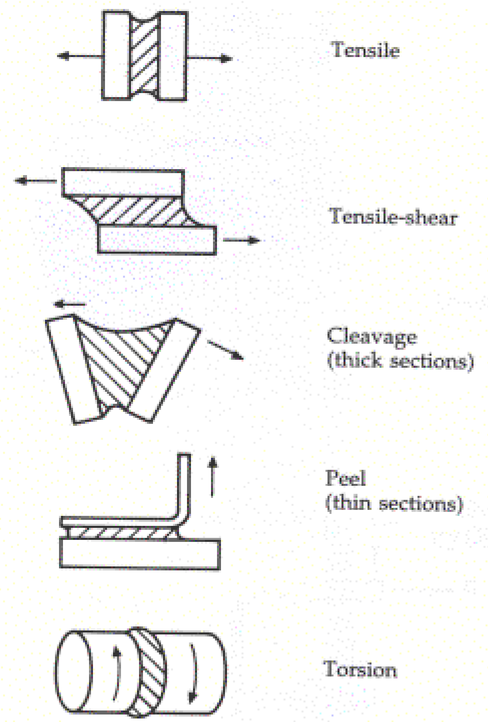
Design Considerations

Loading Modes

The four main loading modes of bonded joints are (see figure right):

- Peel loads produced by out-of-plane loads acting on thin adherends.
- Shear stresses produced by tensile, torsional or pure shear loads imposed on adherends.
- Tensile stresses produced by out-of-plane tensile loads.
- Cleavage loads produced by out-of-plane tensile loads acting on stiff and thick adherends at the ends of the joints.

In practice, a bonded joint will simultaneously experiences several of these loading components.



Basic loading modes experienced by adhesive joints

The aim of designing adhesive joints is to maintain the adhesive in a state of shear or compression. Bonded joints are strongest under these loading conditions. Tension, cleavage or peel forces should be avoided, or their effect minimised. The presence of these stresses will compromise joint strength and fatigue performance. Structural adhesives have relatively poor resistance to through-thickness (peel) stresses, and therefore to obtain maximum efficiency, joints need to be designed to minimise tensile stresses. For composite laminates, resistance to peel stresses may be considerably lower, so even greater care must be taken with these materials to minimise these stresses.

Failure Modes

The aim is to design a joint to fail by bulk failure of the adherends. A margin of safety is generally incorporated in the design to account for factors, such as service environment, type of loading, degree of control in adhesive application, etc. It is important to ensure that the adhesive is not the weakest link. This is because of the high variability in adhesive strength and concern as to the speed of damage growth that can occur under cyclic loading. For composite adherends, failure is often observed to occur in the near surface plies of laminate materials. This is due to the low toughness associated with the thin resin layer present at the surface of these materials. Considerable care needs to be taken to ensure that the thin surface resin layer does not become the weakest link.

Loading and failure Modes continued…..

Design Considerations

Loading & Failure Modes | Design Basics | Simplified Design Procedure | Design Data Requirements| Safety Factors |Certification Requirements

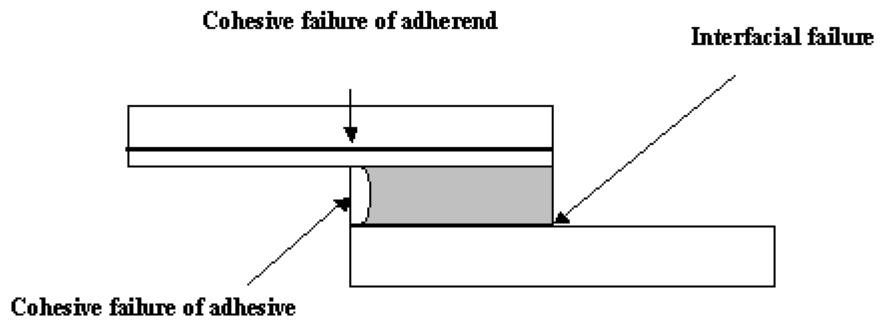
There are a number of potential failure modes for adhesively bonded metallic or composite joints, including (see figure below):

- Adhesive failure at the metal/adhesive or composite/adhesive interface
- Cohesive failure of the adhesive
- Cohesive failure of the adherend

Adhesive failure is the rupture of the adhesive bond, such that separation occurs at the adherend/adhesive interface. This form of failure, which can result from either inadequate surface treatment or material mismatch, should be avoided. Information on interfacial strength, although qualitative, is normally obtained from adhesive joint tests (i.e. lap shear). The term "interface" is used for the layer of material bordering the adherend and adhesive, which encompasses the true interface, the interphase and the near surface area. The material properties in this region tend to differ significantly from the bulk adhesive.

Cohesive failure of the adhesive occurs when the load exceeds the adhesive strength. This tends to be a localised effect, occurring near stress concentrations (ends of joints).

Cohesive failure of the adherend occurs when the load exceeds the adherend strength. For metals, adherend strength usually corresponds to the yield strength. In laminate materials, this form of failure generally initiates from the matrix between layers as a result of out-of-plane tensile or interlaminar shear stresses. Other forms of failure can occur if the composite adherend is not a layered structure (e.g. through-thickness tensile cracking).



Locations of failure initiation in a laminated bonded joint
Note: Generally, failure tends to be mixed mode - a combination of interfacial and cohesive - location of initial failure can also be difficult to detect.

Design Basics >>>

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Design Considerations

Design Basics

Design of a joint should satisfy the following conditions (Appendix 2):

- Allowable shear stress of adhesive not exceeded
- Allowable tensile (peel) stress of adhesive not exceeded
- Allowable in-plane shear stress of adherend not exceeded
- Allowable through-thickness tensile stress of adherend not exceeded

Basic design considerations for maximising the static strength and fatigue performance of adhesively bonded joints include [8, 12]:

- Minimise shear and peel stress concentrations - shear and peel stress concentrations present at bondline ends can be reduced by the use of a combination of tapered or bevelled external scarf or radiused adhesive fillets. Significant increases in the joint strength compared with square-ended bondlines can be achieved. It is recommended that the taper ends of lap joints should have a thickness of 0.76 mm and a slope of 1/10.
- Increasing either the adherend stiffness (i.e. elastic modulus) or adherend thickness results in an increase in load-bearing capacity of the single-lap joint. The use of stiff or thick adherends will minimise peak stress levels and yield a more uniform adhesive stress distribution. **However, the use of absolutely rigid adherends will not prevent the formation of stress concentrations at the bondline.**
- The total overlap length must be sufficiently long to ensure that the shear stress in the middle of the overlap is low enough to avoid creep. Short overlaps can result in failure through creep-rupture. It is recommended that the overlap length, L, is approximately 10 times the minimum adherend thickness to ensure a uniform shear distribution. Increasing the overlap lengths beyond this value does not result in substantial increases in static and fatigue performance. The low stress region in the middle of a long overlap contributes to joint strength by providing elastic restoring force or reserve. It is recommended to maximise bond area. Longer overlap lengths are highly desirable (provided cost and weight penalties are not too high).
- Ensure the joint is loaded in the direction of maximum strength of the adherend. The bonded joint needs not only to be loaded in the direction of maximum strength, but also loads in the weak directions need to be minimised.
- Maintain a uniform bond thickness and wherever possible join identical adherends to minimise skewing of the peak and normal stresses, and to minimise thermal residual stresses due to differences in coefficient of thermal expansion (CTE) values.
- Avoid interlaminar shear or tensile failures of composite adherends. Also, ensure the laminated adherend is symmetric, thus ensuring the coupling stiffness components of the laminate are zero (i.e. no twisting).
- Account for differences in thermal expansion coefficients of the adhesive and adherends (see [11]). Differences can lead to bending stresses and residual stresses, which will compromise joint performance.

The level of allowable stress in the adhesive layer at the limit load (i.e. the highest load expected to be experienced during the service life of the structure) is generally established from the ultimate load (i.e. load at failure) multiplied by a suitable safety factor

Simplified Design Procedure >>>

Design Considerations

Simplified Design Procedure

Chamis and Murthy [53] presented the following simplified procedure for designing adhesively bonded joints:

1. Establish joint design requirements: loads, adhesive, safety factors, etc.
2. Obtain laminate dimensions and properties for the adherends.
3. Obtain properties for the adhesive.
4. Degrade adhesive properties for moisture, temperature and cyclic loading (using equations given).
5. Select design allowables. These are either set by design criteria or are chosen as follows: (a) a load factor on the force **F** (usually 1.5 or 2); or (b) a safety factor of one-half of the degraded adhesive strength (b is preferable).
6. Select length of joint using equation:

   ***l* = F/Sas**

   where **F** = load in adherends per unit width and **Sas** = design allowable shear stress in the adhesive.
7. Check minimum length and maximum shear and normal stresses in adhesive using shear lag equations.
8. Calculate the bending stresses in doublers and adherends using given equations.
9. Calculate margin of safety (MOS) for all calculated stresses. This is usually done at each step where stresses are calculated and compared to allowables using:

   MOS = (allowable stress)/(calculated stress) - 1
10. Calculate joint efficiency **JE** using:

    JE = (joint force transferred, F)/(adherend fracture load) x 100
11. Summarise joint design.

Design Data Requirements>>>

Design Considerations

Design Data Requirements
In design of adhesively bonded joints, consideration should be given to the adherends (geometry and material properties) and adhesive, actual and potential failure modes, thermal properties, magnitude and nature of loading involved, and environmental conditions. Stress analyses of adhesive joints require a database of basic engineering properties of the adhesive, adherend and joint geometry. Basic property requirements for the design of bonded structures are listed below, although not all of these properties would necessarily be required for any given joint configuration.
In-plane and through-thickness (T-T) elastic (i.e. moduli and Poisson's ratios) and strength (or yield stress) properties of the adherends (tension, compression and shear)

- Elastic and strength properties of the adhesive (tension and shear)
-
- Maximum strain in the adhesive and adherends (tension, compression and shear)
-
- Adhesive and adherend(s) non-linear elastic/elastic-plastic stress and strains
-
- Coefficients of thermal expansion (CTE) of adherends and adhesive
-
- Mode I and mode II strain-energy release-rates (fracture mechanics based design)
-
- Thickness of adherends and adhesive layer
-
- Length and width of bonded regions
-
- Safety factors (see Table below)

Cyclic fatigue, creep or high-rate (impact) data may be required, depending on the loading conditions. Fatigue or creep modelling of joint behaviour would require S-N data or time-dependent properties (i.e. creep moduli) in addition to the static material properties of the adhesive-adherend system.

Appendix 1 provides a list of test methods and associated standards for generating design data for bonded and bolted metal and composite structures.

Design Considerations

Safety factors

Recommended Values For Partial Safety Factors [23]

Joint Configuration	Safety Factor (γm)
Adhesive Properties, γm1	1.5
Adhesive thickness, γm2	1.5
Long-term loading, γm3	1.5
Environmental conditions, γm4	2.0
Fatigue (non-fail safe joints), γm5	
Periodic inspection, good access	2.0
Periodic inspection, poor access	2.5
No inspection/maintenance	3.0

Note: In designing a joint, the partial safety factor γm by which the adhesive properties should be divided to give design values is shown below (refer to [12]):

$$\gamma m = \gamma m1 \gamma m2 \gamma m3 \gamma m4 \gamma m5$$

For long-term testing, the overall partial safety factor γm should be no less than 4.0.

For example, if the joint fails at an ultimate load of 100 kN, then the allowable load limit should be 25 kN (or less) for a safety factor of 4.0. As joints will deform, limits may need to be placed on deflections to ensure that at the limit load the structure does not become non-functional. It is recommended that proof tests should be conducted on the structure in order to determine the critical design load.

Certification Requirements>>>

Design Considerations

Certification Requirements

The certification requirements for the design, manufacture and inspection of adhesive joints must be considered from the onset. For example, the JAR (Joint Aviation Requirements) in JAR23.573 states the following. The limit load capacity of each bonded joint, the failure of which would result in catastrophic loss of the aeroplane, must be substantiated by one of the following methods:

- The maximum disbonds of each joint consistent with the capability to withstand the loads must be determined by analysis, tests or both. Disbonds of each bonded joint greater than this must prevented by design features; or
- Proof testing must be conducted on each production article that will apply the critical limit design load to each critical bonded joint; or
- Repeatable and reliable non-destructive inpection techniques must be established which assure the strength of each joint.

Design Examples >>

Design Examples

Lapjoint Analysis | General Comments on Lapjoint Configurations | Scarfjoint Analysis | T-joint Analysis

Lapjoint Analysis

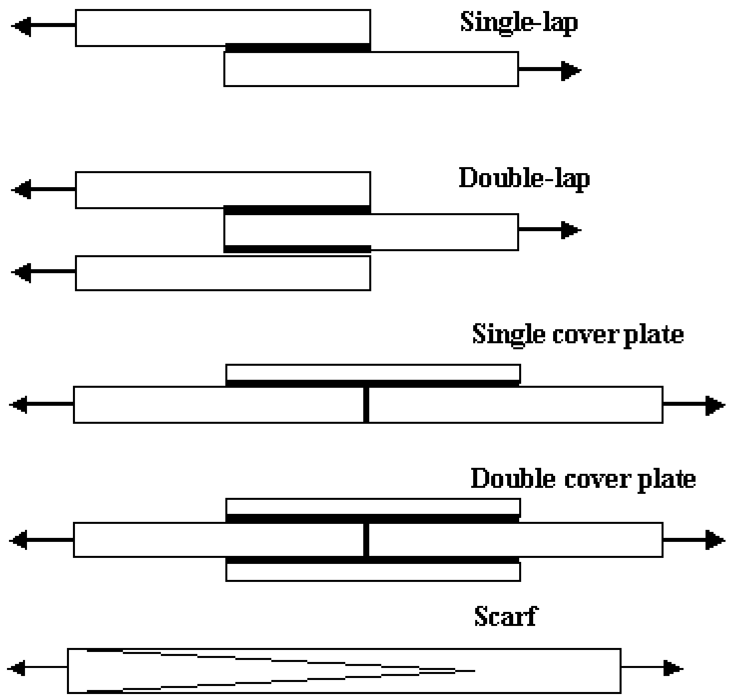
The overall lapjoint geometry is critical to performance. Loads causing peel stresses will compromise joint performance.

In a single-lap joint eccentric forces acting on the joint induce a bending moment. The bending moment causes additional tensile (peel) stresses to be induced in the adhesive layer, concentrated at the ends of the joint.

There are various methods for minimising the negative influence of bending forces caused through eccentric loading. These include:

- increasing the bondline thickness;
- stiffening the adherends (i.e. increase adherend thickness or use of stiffer materials (see figures below));
- use of double overlapping, single and double cover plates, and scarf and step configurations (see figure right); and
- modifications to the adhesive fillet at the ends of joints.

Appendix 2 describes an analytical procedure for producing satisfactory single-lap joints.



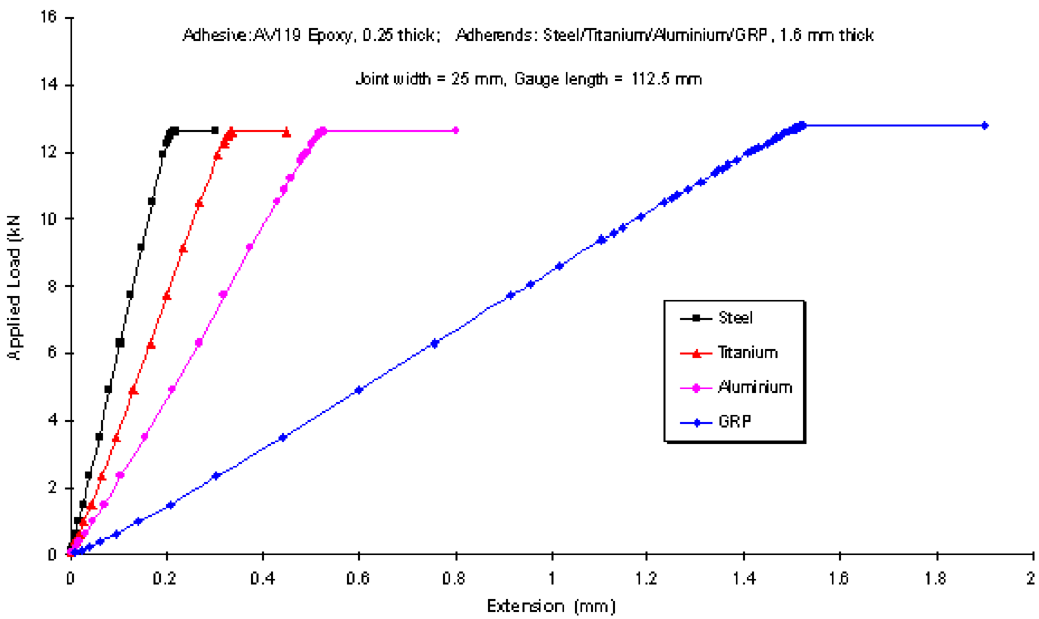
Various lap joint configurations (with increasing joint improvement from top to bottom)

Lapjoint Analysis Continued…..

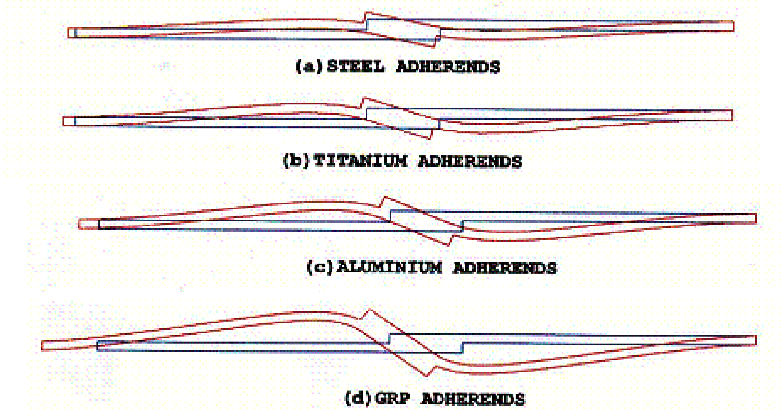
Design Examples

Lapjoint Analysis Continued

The deformed shape (exaggerated) of a single-lap joint manufactured from four different adherend materials is shown in the figure on the right. The corresponding load-displacement response, as predicted using FEA, for the different materials is shown in the figure below. The adherend properties are shown in the table below right.



Typical load-displacement response for a single-lap joint for different substrate materials



Relative deformation of a single-lap joint for different substrate materials (comparison of elastic properties are shown in the table below)

Property	CR1 Mild Steel	6Al-4V Titanium	5251 Aluminium	Tufnol 10G/40
E11 (GPa)	206.0	120.0	72.0	25.2
E22 (GPa)	206.0	120.0	72.0	10.7
E33 (GPa)	206.0	120.0	72.0	25.2
ν12	0.38	0.38	0.35	0.40
ν13	0.38	0.38	0.35	0.14
ν23	0.38	0.38	0.35	0.40
E12 (GPa)	74.6	43.5	26.7	3.25
E13 (GPa)	74.6	43.5	26.7	4.41
E23 (GPa)	74.6	43.5	26.7	3.25

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Lapjoint Analysis Continued……

Design Examples

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Lapjoint Analysis Continued

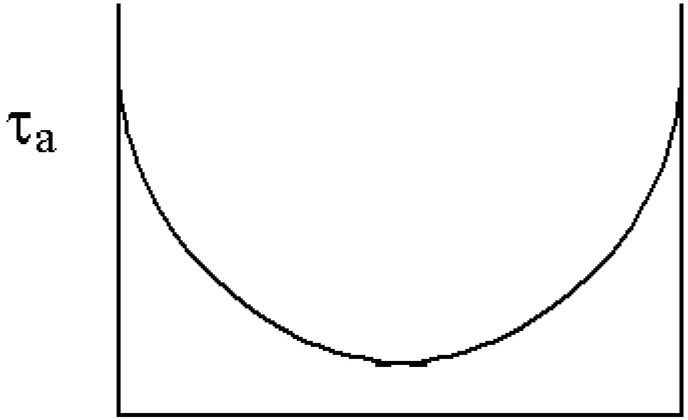
In order to reduce (not eliminate) the eccentricity of the load path that causes out-of-plane bending moments resulting in high peel stresses and non-uniform shear stresses in the adhesive layer, end tabs are bonded to the adherends in single-lap joints as shown in the figure below.

Side-view of single-lap joint with end tabs to reduce eccentric loading

Increasing adhesive thickness results in a more compliant joint to shear stress. The extra adhesive thickness distributes the shear strain over a larger dimension, lowering the strain per unit length and the stress concentration at the ends of the bondline. Alternatively, using an adhesive with a lower modulus will have a similar effect.

An increase in joint width results in an increase in joint strength. Failure load increases in the same proportion as the joint width increases (i.e. doubling the width will double the failure load). This is achieved without affecting the shear stress distribution within the adhesive joint.

Failure load does not increase proportionally with increasing **bond length** (table below). Although increasing the overlap length reduces the average shear stress, the increase is not in proportion to the increase in bond length (NB. The shear stress distribution is non-uniform (figure right) with the ends of the joint resisting a greater amount of stress than the middle of the bond). In order to increase the load capacity of the joint, it is better to increase bond width rather than bond length. As the lap joint length increases, the mean shear stress decreases, and thus the shear stress concentration at the end of the joint increases.



Non-uniform shear distribution along bondline

Lapjoint Analysis Continued…..

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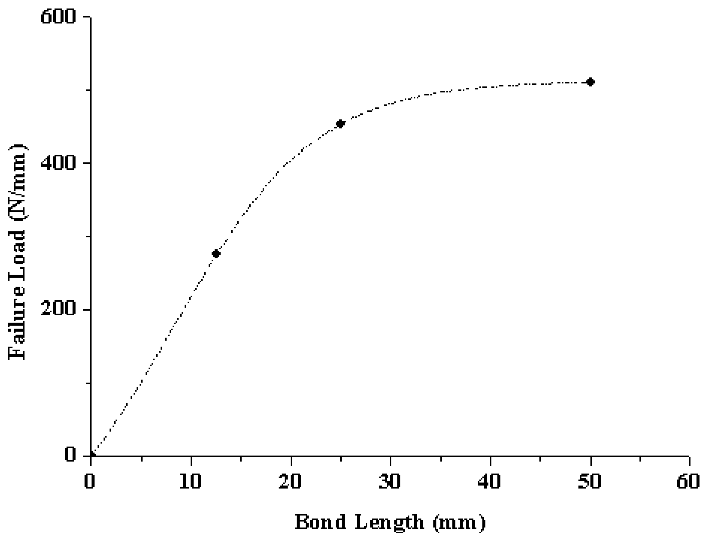
Lapjoint Analysis Continued

The figure below shows that for the woven fabric joints an increase in bond length results in a non-proportional increase in failure load, and that if the overlap length is increased beyond 50 mm, there is only a small change in failure load. This is because yielding of the adhesive occurs at the end of the overlap where the adhesion or cohesive strength of the adhesive is exceeded.



Failure load versus bond length for woven fabric single-lap joints

Adherend Thickness/Overlap length	Load/Width (N/mm)
CR1 Mild Cold Rolled Steel	
1.5 mm thick/12.5 mm overlap	334±11
2.5 mm thick	
12.5 mm overlap	354±10
25.0 mm overlap	428±38
50.0 mm overlap	633±63
5251 Aluminium Alloy	
1.6 mm thick/12.5 mm overlap	191±14
3.0 mm thick/12.5 mm overlap	325±28
6Al-4V Titanium Alloy	
2.0 mm thick/12.5 mm overlap	457±52
Unidirectional T300/924 Carbon/Epoxy	
2.0 mm thick/12.5 mm overlap	369±41
Plain Woven Fabric (Tufnol 10G/40)	
2.5 mm thick	
12.5 mm overlap	275±28
25.0 mm overlap	454±27
50.0 mm overlap	511±32
5.1 mm thick/12.5 mm overlap	327±27

Failure Load Per Unit Width for AV119 Epoxy Adhesive Joints
Note: The "apparent" shear strength measured using lap joints is given in terms of load per unit width (N/mm) rather than load per unit area (i.e. stress).

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General Comments on Lap joint Analysis >>

Design examples

General Comments on Lap Joint Configurations

Single-Lap Joints

- The highest stress concentrations occur in this type of joint (at the free ends of the joint).
- The centre of the joint transmits very low loads.
- Tapered or bevelled external scarf or radiused adhesive fillets reduce stress concentrations at the free ends of the joint.
- Unsupported single-lap joints should only be employed for thin metallic adherends.
- Peel stresses in fibre-reinforced plastic adherends are generally too severe, and therefore it is not advised to use this geometry for structural applications with these materials.

Double-Lap Joints

- Bending and peel stresses are reduced, but not eliminated. Although peel stresses are smaller than for single-lap joints, these stresses limit the thickness of material that can be joined.
- Peel and shear stresses at the ends of the joint can be reduced in a similar manner to those employed for single-lap joints.
- Joint suitable for a wider range of applications compared with the single-lap configuration.

Strap and Scarf Joints

- If correctly designed, peel stresses will be negligible and may be used to join a wide range of materials including fibre-reinforced plastics.
- Joint strengths are higher than either single-lap or double-lap joints.

Stepped Lap Joints

- Suitable for use with thick adherends, but is inherently more difficult to machine or mould than other forms of lap joints. Not applicable to thin adherends.
- Joints exhibit very good stress distribution and high joint efficiency (i.e. good strength/weight ratio).

Scarf joint Analysis >>

Design Examples

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Scarf Joint Analysis

A simple strength-of-materials analysis approach, resolving stresses and areas, can be used to determine shear stress τ and normal stress σ_T in a simple scarf joint, such as that shown in the figure above. The analysis predicts a uniform shear stress in the adhesive layer given by [11]:

$$\tau = P \sin\theta \cos\theta / t \qquad \text{E(1a)}$$

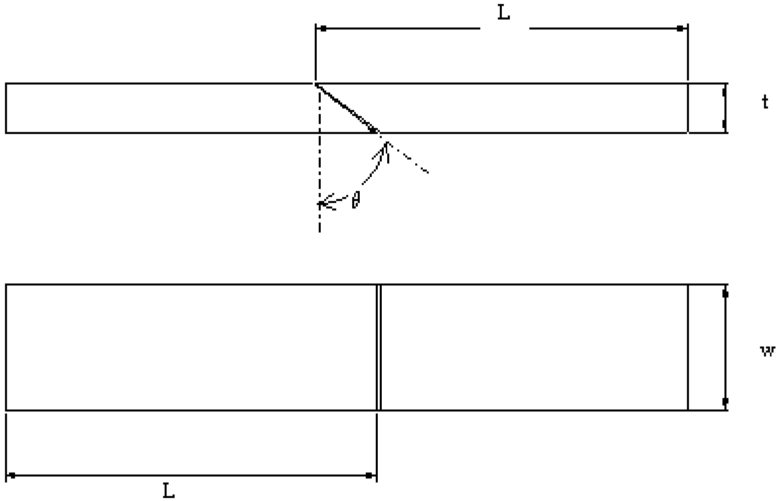
and a uniform normal stress in the adhesive given by

$$\sigma_T = P \cos^2\theta / t \qquad \text{Eq (1b)}$$

where P is the applied (end) load per unit width, t is the adherend thickness and θ is the taper angle.

The ratio of normal stress to shear stress is given by

$$\sigma_T / \tau = \cot\theta \qquad \text{Eq (1c)}$$



Schematic of scarf joint

For a taper angle θ = 60°, σ_T/τ is equal to 0.57. From a designer's perspective, the taper angle should be as large as possible. For repair applications, θq is required to be greater than 87° (i.e. σ_T/τ = 0.05).

In principle, the load bearing capacity increases in proportion with the width w and thickness t of the adherends, and is not limited by local high stress concentrations at the ends, as in a lap joint. The above analysis assumes that as the shear stress distribution is uniform. However, there is no elastic trough in the stress distribution along the adhesive layer to alleviate continuous strains (deformations) under prolonged loading (e.g. creep or low frequency cyclic fatigue loads). Under these conditions eventual failure can be expected if the stress in the adhesive exceeds the shear yield stress. Joint strength is also sensitive to damage in the tapered edge. It is advisable that the design be based on conservative estimates of elastic properties. For adherends with dissimilar thermal/elastic properties, the analysis will be more complex.

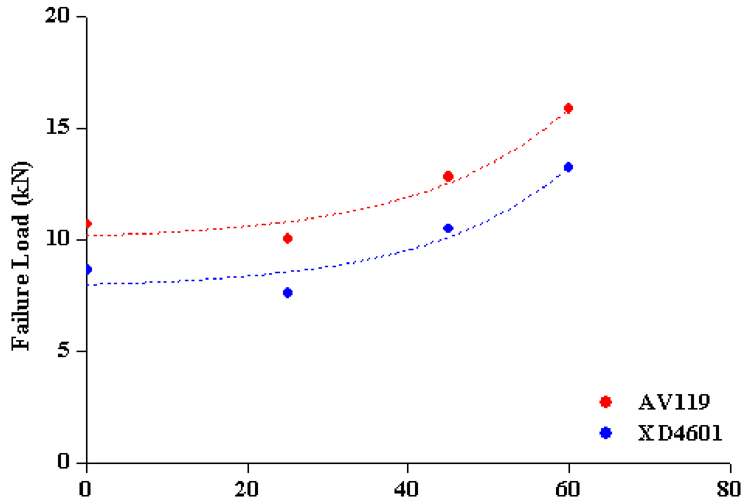
Scarf joint Analysis Continued…..

Design Examples

[Lapjoint Analysis](#) | [General Comments on Lapjoint Configurations](#) | [Scarfjoint Analysis](#) | [T-joint Analysis](#)

Scarf Joint Analysis Continued

The figure (right) shows the relationship between measured failure load and scarf angle for mild steel joints bonded with AV119 and XD4601 epoxy adhesives. The load values shown at θ = 0° were obtained using the butt-tension test.



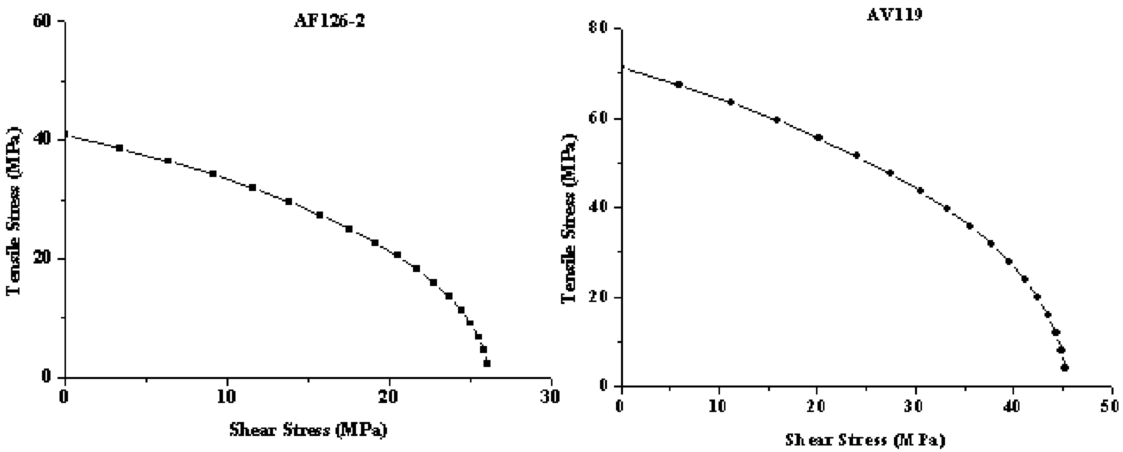
Failure load versus scarf angle for different epoxy adhesives (w = 25 mm, t = 10 mm, L = 80 mm)

Failure envelopes for scarf joints (see figure below) can be estimated using the Hill's quadratic failure criterion, shown below, in conjunction with Equations (1a) and (1b) [above](#).

$$\left(\frac{\sigma_T}{X_T}\right) + \left(\frac{\tau}{S}\right) = 1$$

Eq (1d)

X_T and S are the tensile strength and shear strength of the adhesive, respectively. σ_T and τ in the above relationship correspond to maximum load values.



Failure envelopes for scarf joints bonded with epoxy adhesives

[T-Joint Analysis >>](#)

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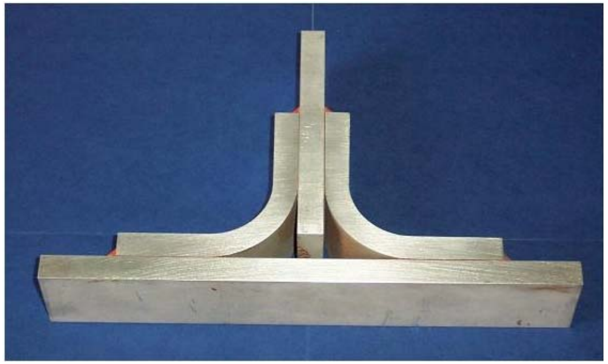
Design Examples

T-Joint Analysis

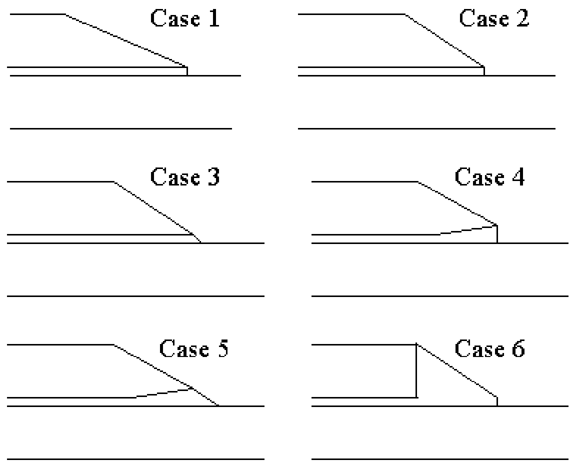
The general approach is to use finite element analysis to determine the effect of geometric and material properties on the strength and stiffness of complex joints, such as the T-joint shown in the figure (right). Closed form (analytical) solutions can be used for preliminary design purposes for assessing the effect of geometry and material properties on joint stiffness (see Appendix 3). This approach assumes that the properties in the longitudinal direction of the material determine how a structure will deform (see [54]).



There are a number of approaches that can be used to modify joint geometry in order to improve stiffness and strength (alternative designs). It should be noted that the size of internal fillets play significant role in joint performance. Fatigue life is particularly sensitive and it is therefore advised to ensure that the internal fillets are large. External fillets have far less effect on the fatigue performance of T-joints. Completely filling the internal cavities of the joint will improve joint stiffness.

EXAMPLES OF ALTERNATIVE T-JOINT GOEMETRIES

Altering the geometry of joints, such as the T-joint, can have beneficial effects on the stiffness and strength. An overview of the designs and FEA results will be presented in this section. A standard geometry and material system (2024-T6 aluminium/XD4601) will be used to illustrate these effects.



Fillets/Tapers

The effects of the presence of fillets and tapers on the T-joint predicted strength and stiffness were investigated. All geometries were based on the original adhesively bonded T-joint with a fully filled Bermuda Triangle region. In the first case a large taper with a gradient of 1 in 2.5 (making an angle of 21.8° with the horizontal) was applied to the flanges. In cases 2 to 5, the taper was steeper, with an angle of 32.5°. In case 2, the adhesive was squared off at the end of the taper. In case 3 an adhesive fillet was present. In case 4, the flange was also tapered upwards with the adhesive squared off. Case 5 was the same as case 4 but with an adhesive fillet continuing the taper. Case 6 had no taper on the flange, but had a large tapered adhesive fillet. The different geometries are shown (right) - the thickness of the adhesive layer has been exaggerated for clarity.

T-Joint Analysis Continued…..

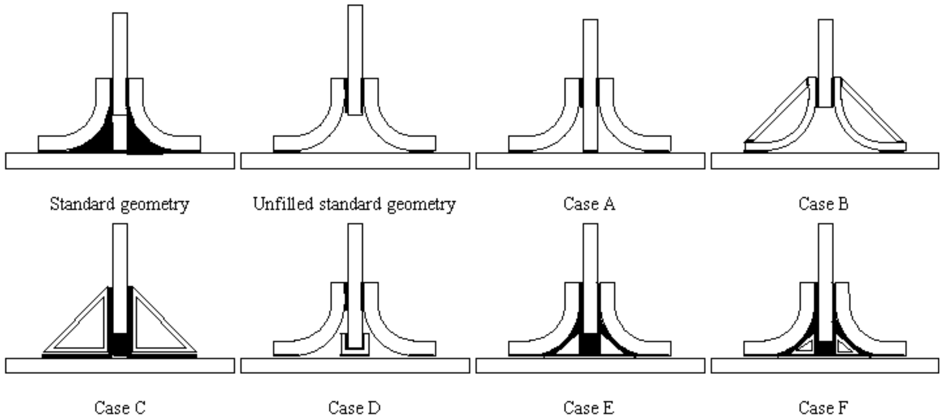
Design Examples

T-Joint Analysis Continued

Vertical loading was applied to these geometries using FEA. Predictions of failure loads, using maximum principal stress, and joint stiffness and were calculated. These were also calculated as specific stiffness and specific strength. The results are shown below.

In general these geometry changes only have marginal effects on strength and stiffness. When the results are calculated as specific strengths and specific stiffness they show a slight improvement over the original geometry due to the reduced specimen weight. The taper with the largest effect, when comparing both actual values and specific values, is case 1 with the longest taper (a 1 in 2.5 slope). Although the actual stiffness value for case 1 is slightly lower than the original geometry, the specific stiffness is the highest of all the cases. The actual strength and specific strength are also the highest of all geometries. The worst case appears to be case 6, with the large adhesive fillet. In this case, the stiffness is average, but the additional weight of the adhesive brings the specific stiffness down. The strength and specific strength are the lowest of all cases.

Joint	Weight (kg)	Stiffness (kN/mm)	Specific Stiffness (kN/mm/kg)	Strength (kN)	Specific Strength (kN/kg)
Original Geometry	1.688	85.80	50.83	86.58	51.29
Case 1	1.612	84.63	52.50	88.04	54.62
Case 2	1.640	85.58	52.18	86.92	53.00
Case 3	1.640	84.34	51.43	86.42	52.70
Case 4	1.640	85.23	51.97	87.08	53.10
Case 5	1.640	82.84	50.51	86.13	52.52
Case 6	1.697	85.00	50.09	86.13	50.76



Alternative T-joint Designs

A variety of alternative T-joint designs were investigated using FEA and results compared to the standard geometry and the standard unfilled standard geometry. Case A is the standard unfilled T-joint geometry except the web is bonded to the base. Case B has thinner flanges to reduce weight with a strut to increase stiffness. Case C uses triangular flange pieces that are fully bonded to the web and base. Case D is similar to Case A except there is a metal 'cup' at the base of the web. Case E has polyurethane wedge shape blocks in the Bermuda Triangle region to reduce the volume of adhesive. Case F is similar to case E except that the blocks are hollow aluminium wedges. All geometries are illustrated (left). The dark areas are adhesive regions

T-Joint Analysis Continued…..

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T-Joint Analysis Continued

In the FE analyses these joints were loaded in the vertical direction. The analyses were also repeated under transverse loading. Maximum principal stress was used to predict failure loads. The results are shown in the tables below.

Vertical Loading (Results table, right)

The lightest of the geometries are the two cases with external struts, case B and case C. The presence of the struts greatly increases the stiffness of the T-joint under vertical loading. The strength of these joints is also slightly higher than the comparable unfilled standard geometry. Case E and case F are closest to the filled standard geometry, but the presence of the blocks within the adhesive appears to make little difference to the results.

Joint	Weight (kg)	Stiffness (kN/mm)	Specific Stiffness (kN/mm/kg)	Strength (kN)	Specific Strength (kN/kg)
Original Geometry	1.688	85.8	50.83	86.58	51.29
Unfilled Geometry	1.606	56.8	35.37	24.34	15.16
Case A	1.606	57.9	36.05	25.03	15.59
Case B	1.432	109.3	76.33	27.79	19.41
Case C	1.491	142.9	95.84	35.73	23.96
Case D	1.638	57.7	35.23	25.12	15.34
Case E	1.656	81.0	48.91	73.71	44.51
Case F	1.768	88.1	49.83	89.45	50.59

Transverse Loading (Results table, left)

Cases A, B and D should be compared with the unfilled standard geometry, while case E and case F are closest to the filled standard geometry. As with the vertical loading cases, the presence of struts (cases B and C) increases the specific stiffness to levels above those predicted for the filled standard geometry. In terms of specific strength, case A and case D show the largest improvements over both the unfilled and filled original geometries. These are both cases where the web is bonded to the base. Under transverse loading the presence of the blocks within the adhesive appears to make little difference to the results.

Joint	Weight (kg)	Stiffness (kN/mm)	Specific Stiffness (kN/mm/kg)	Strength (kN)	Specific Strength (kN/kg)
Original Geometry	1.688	6.31	3.74	6.80	4.03
Unfilled Geometry	1.606	3.74	2.33	6.74	4.20
Case A	1.606	6.61	4.12	8.35	5.20
Case B	1.432	8.33	5.82	5.82	4.06
Case C	1.491	10.07	6.75	6.11	4.10
Case D	1.638	5.79	3.53	7.66	4.68
Case E	1.656	5.49	3.32	6.73	4.06
Case F	1.768	6.50	3.68	6.73	3.81

T-Joint Analysis Continued.....

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Design Examples

T-Joint Analysis Continued

Thermal Residual Effects

Thermally induced residual stresses have been known to have an adverse effect on the strength of bonded joints. FEA was carried out to compare the strength of a joint with and without taking into account the difference in coefficient of thermal expansion (CTE) between aluminium adherend and XD4601 adhesive. The table below compares the two cases for the standard unfilled joint configuration. When using the maximum principal stress as a failure criterion the joint strength increases slightly when thermal stresses are taken into account.

Standard Unfilled Joint Geometry	Stiffness (kN/mm)	Strength (kN)
Excluding thermal effects	56.8	24.34
Including thermal effects	56.8	25.15

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Fatigue and Creep >>>

Fatigue and Creep

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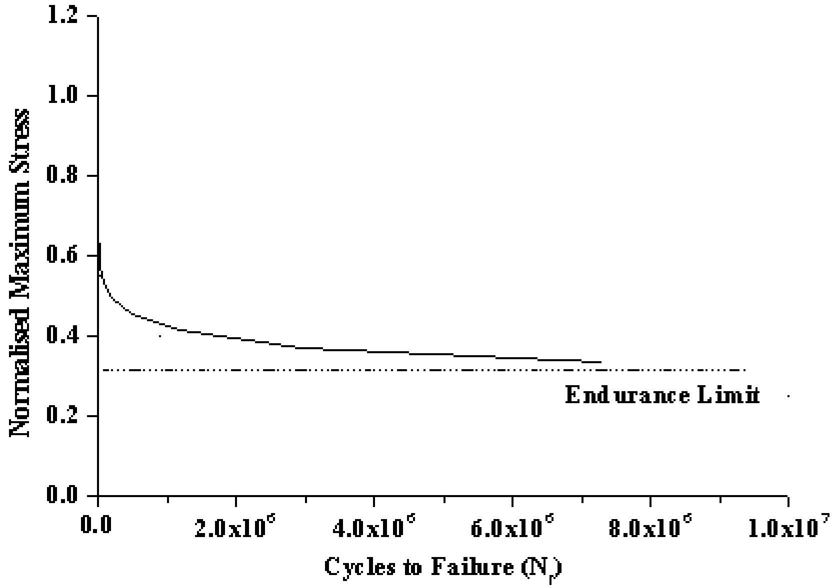
Fatigue and Creep

It is recommended that verification of fatigue and creep performance should be undertaken on critical joints to demonstrate that the joint can carry the ultimate load required throughout its design life, under representative conditions of stress, temperature and humidity. Design of bonded joints is still far form the point where fatigue and creep performance can be inferred from data obtained for simple coupon tests (e.g. single-lap joints), which can only provide comparative data.

Fatigue damage or creep in an adhesive layer can be avoided, or at least minimised, by ensuring that the adhesive remains in an elastic state for most of its service life. A practical approach is to ensure the overlap length of the joint is sufficiently long, so that the adhesive remains elastic. The elastic region on unloading acts as an elastic reservoir to restore the joint to its unstrained state, thus preventing accumulation of shear strain. Ideally, significant plastic deformation of the adhesive should only be permitted when the joint is stressed to the limit load. Where the limit load is the highest load expected to be experienced during the service life of the structure. Even at the ultimate load, which is 1.5 times the limit, the strain in the adhesive should not approach the failure strain. Provided stress levels are sufficiently low (i.e. below the "endurance limit" (see figure below)) then fatigue should not to be a problem.



Typical S-N curve with fatigue or endurance limit

Repeated cyclic loading to high plastic strains can result in creep failure occurring within a relatively short number of cycles due to the cumulative effect of cyclic shear strains. From a design perspective, a sufficiently long overlap length will ensure that most of the adhesive remains elastic. The elastic region acts as an elastic reservoir during unloading, enabling the bond layer to recover (i.e. stress relief) and thereby preventing creep strain accumulating. Provided the minimum shear stress at the middle of the overlap remains within the elastic limit of the adhesive and the maximum shear strain at the ends of the overlap is limited to a value below the adhesive yield strain, then the joint should be suitable for use under cyclic loading conditions. Creep within the low stress region of the bonded region should be kept to a minimum. This effect is more pronounced as the frequency of testing is reduced.

[Cyclic Fatigue >>](#)

Fatigue and Creep

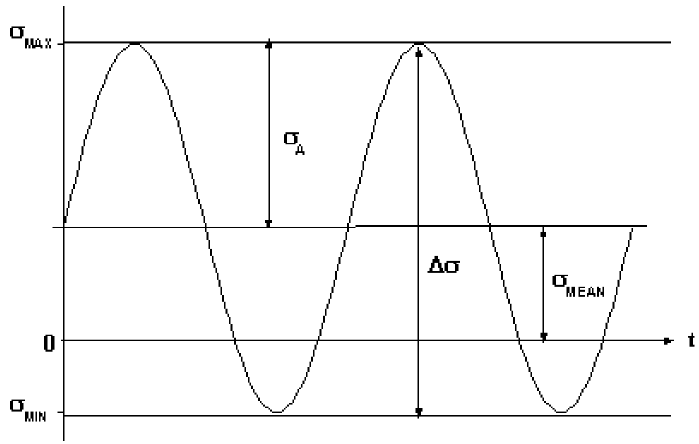
Cyclic Fatigue

Cyclic fatigue loading can be in the form of either constant amplitude or variable amplitude spectrum loading. Constant Amplitude Loading is defined by the terms shown (right).

Variable Amplitude Spectrum Loading: An important aspect to fatigue design is ensuring that the load spectrum is representative of the stresses and strains actually experienced by the component during service. The distribution and number of stress cycles, and the order in which the loads are applied define the stress spectrum loading. For example, stress spectrum loading is used for testing spherical tanks for transporting liquid natural gas and for assessing fatigue performance of aircraft wings. Service load spectra can be estimated from typical operating conditions experienced by the component. This can be achieved by monitoring strain at critical regions of the component under service loads. For the purpose of life prediction, the spectrum loading is simplified.

Standard air spectra programs are available to simulate the load sequence for aircraft transport and military aircraft (e.g. Transport WIng STandard (TWIST) and Fighter Aircraft Loading STAndard For Fatigue Evaluation (FALSTAFF)). TWIST was designed to simulate the loading spectra for transport wing tension skins near the main landing gear attachment. The loading program allows for different types and levels of gust loadings. Both TWIST and FALSTAFF are available as commercial software packages.

Metal airframes are generally fatigue tested under spectrum loading conditions to a minimum of two lifetimes to ensure adequate fatigue life. A high structural reliability is generally guaranteed if the fatigue life of the structure is 2-4 times the lifetime of the structure. However, the high variability associated with fatigue life of composites means that the 2-4 lifetime fatigue criteria may not be sufficiently reliable, and hence the need to use larger life factors for fatigue design.



Minimum stress, σ_{MIN}
Maximum stress, σ_{MAX}
Stress range, $\Delta\sigma = \sigma_{MAX} - \sigma_{MIN}$
Stress amplitude, $\sigma_A = \Delta\sigma/2 = (\sigma_{MAX} - \sigma_{MIN})/2$
Mean stress, $\sigma_{MEAN} = (\sigma_{MAX} + \sigma_{MIN})/2$
Stress ratio, $R = \sigma_{MIN}/\sigma_{MAX}$
$R = -1$ for fully reversed loading
$R = 0$ for zero-tension fatigue, and
$R = 1$ for a static load.

Stress parameters for constant amplitude sinusoidal cyclic loading

Fatigue and Creep

S-N Curves

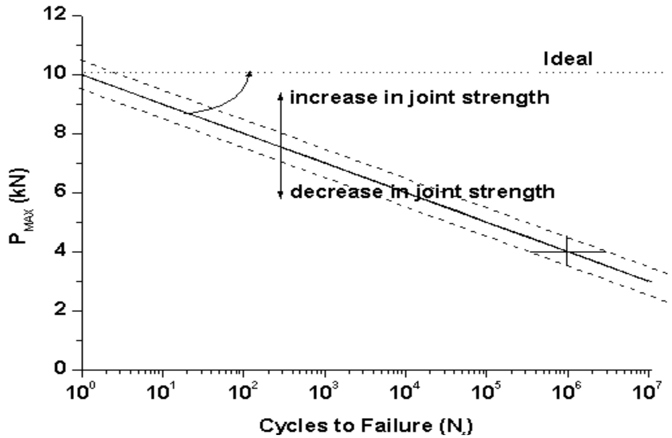
The mean stress level (σ_{MEAN}) and stress amplitude (σ_A) of the imposed fatigue cycle are known to play an important role in influencing the fatigue behaviour of engineered structures. The general approach is to present fatigue data in the form of stress-cycle (S-N) curves (see figure top right) in which the stress range ($\Delta\sigma$) is plotted against cycles to failure (N_f). From these curves it can be observed that fatigue life decreases with an increase in mean stress level and that for a given load, the number of cycles to failure can be expected to increase as the ultimate strength of the joint is increased. The scatter in fatigue life generally lies within a band determined by the scatter in joint strength. Fatigue life (i.e. cycles to failure) at a given load or stress level tends to vary by an order of magnitude (e.g. 1,000 - 10,000 cycles).

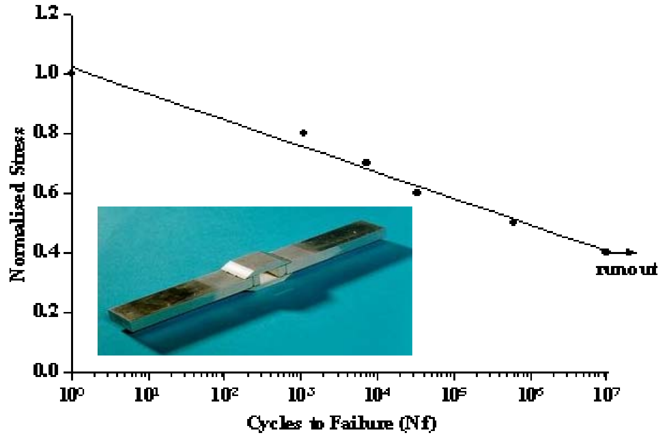
The S-N curve is usually determined from experimental data, by carrying out fatigue tests at different stress levels of stress (i.e. 80%, 70%, 55%, 40% and 25% of the ultimate strength of the joint) on a number of nominally identical specimens representing the detail under consideration. Each point on a S-N curve represents a single test result corresponding to a given specimen. The smaller the slope of S-N curves the better the fatigue performance. The ideal would be to have a slope with a zero gradient.

S-N curves can be presented as either log-log or linear-log plots. One approach is to normalise the applied stress range ($\Delta\sigma$) with respect to the ultimate failure load or failure stress of specimens tested under monotonic loading at an equivalent loading rate to the fatigue cycling and then to plot the normalised stress values with respect to N_f. Normalised S-N curves (see figure right) for tension-tension fatigue can be approximated by the following relationship:

$$\sigma_{MAX} / \sigma_{ULT} = 1 - k \log N_f$$

where σ_{MAX} is the maximum load applied to the specimen, σ_{ULT} is the ultimate strength of the joint, N_f is the number of cycles to failure and k is the fractional loss in strength per decade of cycles. The value of k is a measure of fatigue resistance of the joint and is dependent on the joint geometry and loading conditions.





S-N data for 5251 aluminium alloy/AF126-2 epoxy tapered strap joints (R = 0.1 and f = 5 Hz)

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S-N Curves Continued

The value of k is a measure of fatigue resistance of the joint. The lower the k value the better the fatigue performance. The table (right) shows typical k values for a number of metal and composite joints bonded with epoxy adhesives. The performance of the joint depends on the joint geometry and the range of stresses that occur in the regions of peak stress (i.e. stress concentrations near the ends of adhesive joints). A scarf joint with a 30° taper where failure is dominated by shear stresses has a far better fatigue performance than tests where peel stresses are the dominant stress.

Joint Configuration	k
Scarf (aluminium adherends with 30° taper)	0.055
Double-lap (titanium adherends)	0.075
Double strap (aluminium adherends)	0.088
Single-lap (mild steel adherends)	0.093
Double-lap (woven fabric)	0.097
T-joints (direct tension)	0.104
T-peel (mild steel adherends)	0.130

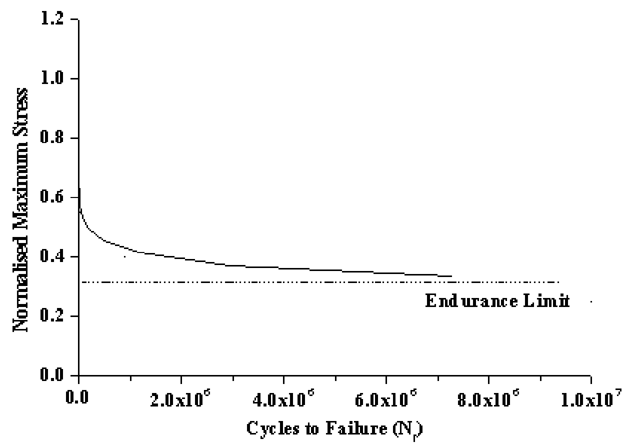
Typical k Values for Bonded Joints (R = 0.1 and f = 5 Hz)

Alternatively, fatigue data may be represented using the following relationship:

$$N = N_0 \left(\frac{S}{S_0} \right)^{-m}$$

N_0 and S_0 are coordinates of a given point on the S-N curve and m is the slope of the curve.

S-N curves can be used to determine the fatigue or endurance limit (i.e. maximum fluctuating stress a material can endure for an infinite number of cycles without causing failure) of a material or structure. Under constant amplitude loading conditions, most materials or structures exhibit a plateau in the stress-cycle curve (see figure right), which typically occurs at N > 106 cycles. The plateau level corresponds to the fatigue or endurance limit. Below this limit, the material or structure can be cycled indefinitely without causing failure. In most engineering applications, designers aim to ensure that no fatigue cracks develop during the service life of the component; S-N approach works well in these cases.



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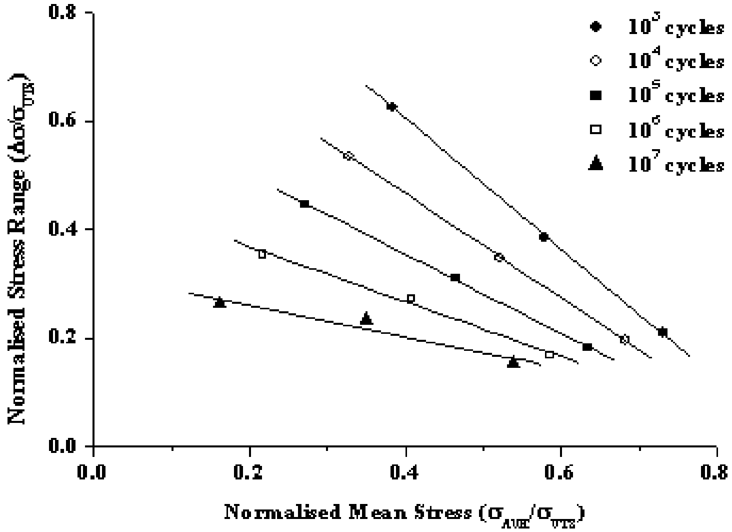
S-N Curves Continued

Constant-life Diagrams

Constant-life diagrams can be used to represent the effects of mean stress and stress amplitude on fatigue performance of adhesive joints and composites. Different combinations of normalised stress amplitude, $\Delta\sigma/\sigma_{ULT}$, and the normalised mean stress, $\sigma_{MEAN}/\sigma_{ULT}$, are plotted to give constant fatigue life curves.

The results shown in the figure (right) have been normalised with respect to the ultimate tensile strength, σ_{UTS}, of the material. In principle, the curves should converge to the static strength of the composite on the mean stress axis (i.e. when the mean load is increased to the static strength then no amplitude is required to cause failure).



Stress amplitude-life plots for different mean stress values

A Goodman type curve can also be used to represent the effect of mean stress and stress amplitude on fatigue performance of bonded joints. The Goodman relation is given below:

$$\sigma_A = \sigma_{FS}\left(1 - \frac{\sigma_{MEAN}}{\sigma_{ULT}}\right)$$

where σ_A is the stress amplitude (for a non-zero mean stress), σ_{FS} is the fatigue strength (for a fixed life), σ_{MEAN} is the mean stress and σ_{ULT} is the ultimate strength of the material.

Creep>>

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Fatigue and Creep

Creep

Creep deformation usually occurs over a period of time when a material (or structure) is subjected to constant load (or stress) (i.e. time-dependent deformation). Strain (or deformation) increases with load, temperature, relative humidity and time. Polymeric materials, such as adhesives can undergo creep deformation at room temperature (referred to as cold flow).

Creep data is usually presented as a plot of creep versus time with stress and temperature remaining constant (see figure right). As creep is defined as time-dependent deformation of a material (or structure) under a constant load, the design process should involve substituting creep modulus for stiffness (or Young's modulus). The creep modulus is the apparent stiffness as determined by the total deformation to the time defined.

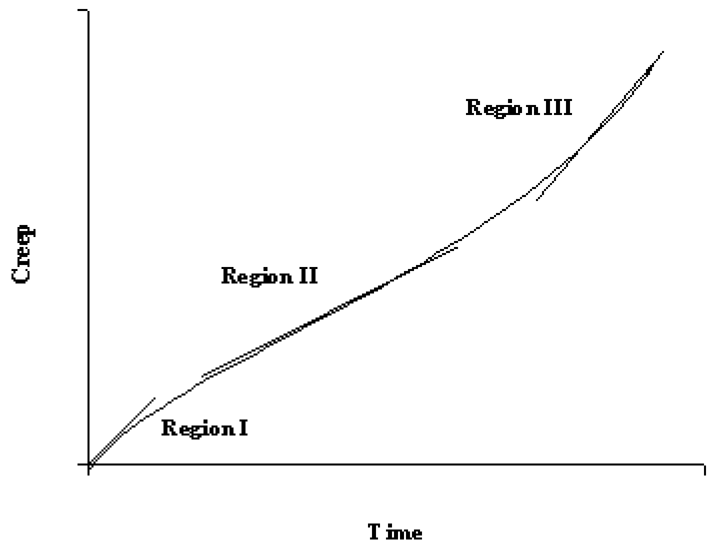
When the applied loads are approximately constant for the duration of loading, a "pseudo-elastic" design method may be used. Creep or time-dependent modulus:

$$E(t) = \sigma/\varepsilon(t)$$

may be modelled by the following relationship:

$$E(t) = E_o t^{-n}$$

where $\varepsilon(t)$ is the strain-time function, E_O is initial (or 1 second) modulus and n is the creep index (an experimentally derived constant). The value of E_O is obtained by extrapolation. This approach can be used for different loading modes and elastic properties.



Creep versus time plot

Regions

Region I - First stage, or primary creep, starts at a rapid rate and slows with time.
Region II - Second stage (secondary) creep has a relatively uniform rate (minimum gradient).
Region III - Third stage (tertiary) creep has an accelerating creep rate and terminates by failure of material at time for rupture.

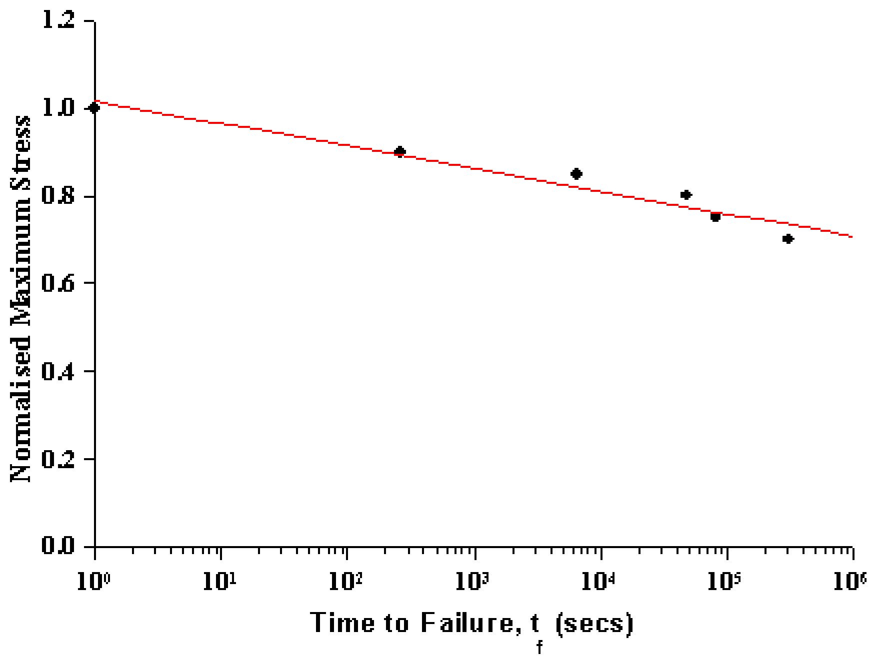
Creep Continued….

Fatigue and Creep

Creep Continued

The creep index n is a measure of viscoelastic behaviour and is dependent on the resin type and degree of cure, interfacial bonding, fibre format, orientation of the fibres with respect to the applied load, fibre volume content, loading regime and environmental effects (i.e. temperature, moisture and aggressive chemicals). Creep index can be obtained from the gradient of E(t) versus log t. For design purposes, creep modulus and creep index should be obtained from direct experiments on the bonded joint.

In addition to loss of stiffness as a consequence of creep, it is possible that strength reductions will occur. Creep rupture can occur at stress levels below the monotonic strength of the joined system. Tests need to be carried out to verify that the joined system will not fail as a result of stress rupture. The figure below shows the time-to-failure for a 5251 aluminium alloy strap joint bonded with AF126-2 epoxy adhesive



Creep rupture of 5251 aluminium alloy/AF126-2 epoxy tapered strap joints

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Design Software >>

Design Software

FEA Programmes| Analytical Software

Techniques for stress analysis of a joint generally fall into two main categories: analytical, closed-form methods and finite element methods. Analytical methods are generally quick and easy-to-use, but are only suitable for simple geometries. These methods cannot accurately predict stresses and strains as the analytical equations, by their simple nature, cannot fully account for the complete stress and strain conditions within the joint. Finite element methods have an advantage in that almost any geometrical shape can be analysed and are capable of more accurate analysis of stress and strain distributions. The disadvantages of these methods are that analyses are expensive and specialist knowledge is required. An ideal method would be an accessible yet accurate stress analysis technique. This section reviews finite element analysis (FEA) and analytical based software developed for the analysis and design of bolted and bonded structures, and materials selection.

FEA Programs

Numerical analysis techniques, such as FEA, are used extensively in the design and stress analysis of adhesively bonded and bolted structures. These techniques offer solutions to complex problems that are too difficult or impossible to resolve using analytical, closed-form solutions. Numerous FEA codes are available (see table below). These codes provide in-built constitutive models for simulating the behaviour of most adhesives, allowing for non-uniform stress-strain distributions, geometric non-linearity, hygrothermal effects, elastic-plastic and visco-elastic behaviour, static and dynamic analysis, and strain rate dependence. Orthotropic element types include two-dimensional (2-D) solid plane-stress or plain-strain elements, axisymmetric shell or solid elements, three-dimensional (3-D) solid or "brick" elements and crack-tip elements. A number of automatic mesh (element) generators are available with post-processing capabilities (e.g. PATRAN and FEMGV).

Although numerical methods are able to accommodate complex geometries, loadings, material properties and boundary conditions, the solutions generated are only approximations to the actual solution. It is important that the designer/analyst is aware of the limitations of the numerical techniques being applied and has a fundamental understanding of the mechanics of bonded and bolted joints (i.e. stresses and failure mechanisms).

Stress analyses (especially FE methods) are often used to compare stress/strain distributions obtained from different joint configurations (e.g. lap, scarf and butt joints) or geometries (varying adhesive and adherend thickness, overlap lengths, fillet shapes). Hence, finite element stress analysis can be used as a tool for optimising the design of a joint. Evolutionary optimisation method **EVOLVE** has been used to optimise the shape of adhesive fillets [64]. This process allows selected properties to drive the optimisation process (e.g. minimising the maximum principal stress in the adhesive). **EVOLVE** relies on an iterative FE analysis and the progressive removal of elements using a rejection criterion. This takes the guesswork out of the design process.

Design Software Continued….

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Finite Element Packages (see also [57])

Name	Supplier	Application	Features
ABAQUS	Hibbit, Karlsson & Sorenson, Inc.	• General purpose FE program • Linear, non-linear and coupled analysis • Large materials model library	• Anisotropic material models in all elements • 2-D and 3-D plate/shell and solid elements • Temperature and strain-rate dependence of properties • Fracture mechanic/ crack propagation analysis • Maximum stress and strain, Tsai-Hill, Tsai-Wu, Azzi-Tsai-Hill and user defined failure criteria
ANSYS	Swanson Analysis System Inc.	• General purpose FE program • Non-linear analysis (non-composite applicable) • Pre- and post-processing	• Isotropic and orthotropic material properties • 2-D and 3-D plate/shell and solid elements • Laminated shell elements • Crack-tip solid and thick-shell elements • Maximum stress and strain, Tsai-Wu and user defined failure criteria
LUSAS	FEA Ltd.	• General purpose FE program • Linear and non-linear analysis • Static, creep, fatigue and dynamic analysis	• Plate/shell and solid elements • 2-D and 3-D interface elements • Laminate analysis/Hashin damage model • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill, Hoffmann, Hashin, Tsai-Wu failure criteria
COSMOS	Structural Research& Analysis Corp.	• General purpose FE program • Linear and non-linear analysisStatic and dynamic analysis • 2-D and 3-D structural and thermal models	• Plate/shell and solid elements • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill and Tsai-Wu failure criteria
NASTRAN	MacNeal-SchwendlerCorp.	• General purpose FE program • Static and dynamic analysis • Linear and non-linear analysis • Pre- and post-processing by PATRAN	• Plate/shell elements - allowance for orthotropic temperature-dependent properties • Ply stresses • Tsai-Hill, Hoffmann and Tsai-Wu failure criteria
NISA	Engineering Mechanics Research Corp.	• General purpose FE program • Linear, non-linear and coupled analysis	• Solid and thick-shell elements • Laminated shell elements • In-plane and interlaminar stresses • Maximum stress and strain, von Mises, Tsai-Wu, delamination failure criteria

Design Software

FEA Programmes| Analytical Software

Analytical Software

In the literature and design guides there are many examples of analytical, closed-form solutions for obtaining stress and strain distributions. These analytical models are generally based on modified shear-lag equations. Aside from the shear-lag analysis technique, other workers have carried out stress analyses using a variety of other methods, such as those based on Hashin's variational analysis using the principle of minimum complementary energy [65]. Reviews of these analytical theories and their assumptions have been published [66-67].

As the analytical equations have become more complex (including factors such as stress variation through the adhesive thickness, plasticity, thermal effects, etc.), there is a greater requirement to use computing power to solve for the stresses. Hart-Smith [52, 68-70] has had a great influence on the methods used for stress analysis of adhesive joints and much of his work is evident in the **P**rimary **A**dhesively **B**onded **S**tructure **T**echnology (**PABST**) programme. Versions of this method (e.g. A4EG, A4EH, A4EI) have been prepared as FORTRAN programmes and have been used extensively in the aerospace industry. Other analyses have been implemented in spreadsheets or as a programme for personal computers (e.g. **JOINT** [71]).

Although simplified analytical procedures for designing adhesively bonded joints are available in the form of PC compatible software [72], these packages are limited in number and scope. As with all design tools, the effectiveness of the analysis is directly related to the users knowledge, and therefore it is advisable that the user has a good understanding of engineering design and material behaviour. The software packages are there to assist in the design of efficient joints. A brief overview of commercial PC based analysis/design software packages is given in the table below. The main features of each software package are identified.

Engineering Sciences Data Unit (ESDU) provide a comprehensive range of data sheets and software for use in structural design, including analysis of bonded and bolted metallic structures (see table below). **ESDU** data sheets also cover circular holes in orthotropic plates, laminated composite materials and structures (including pipes, beams and sandwich panels), and fatigue endurance of metallic structures. The information is provided primarily for use in aerospace structures, but has wider application to other areas of engineering. The information is accepted by the Federal Aviation Administration (FAA) in the United States and by the Civil Aviation Authority (CAA) in the United Kingdom as a basis for submissions. In most cases, PC software is available with computer listings.

The production of the design data and software is monitored and guided by expert committees of professionally qualified engineers from industry, research laboratories and universities. Items are continuously checked and updated to include the latest amendments, which are available to software leasers on request. Each program has a main menu enabling the user access to different input screens (e.g. material properties, loading conditions and results). The **ESDU** software is relatively sophisticated compared with most analytical packages. To maximise the software benefits requires a good understanding of engineering design.

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PC Based Software Packages (see also [73])

Name	Supplier	Application	Features
BOLT	G.S. SpringerStanford University	Design of pin-loaded holes in composites	• Prediction of failure strength and failure mode • Three types of bolted joints: Joints with a single hole, Joints with two identical holes in a row Joints with two identical holes in tandem • Applicable to uniform tensile loads and symmetric laminates
BISEPS-LOCO	AEA Technology,UK	Closed form computer code for predicting stresses and strains in adhesively bonded single-lap joints	• Tensile/shear/bending moment loading • Adhesive peel and shear stress predictions • Allowance for plasticity in adhesive layer • Thermal stress analysis
BISEPS-TUG	AEA Technology, UK	Closed form computer code for predicting stresses and strains in adhesively bonded coaxial joints	• Stepped and profiled joints • Orthotropic adherends • Torsional and axial loading • Allowance for plasticity in adhesive layer • Thermal stress analysis
CoDA	National Physical Laboratory, UK.	Preliminary design of composite beams and panels, and bolted joints	• Synthesis of composite material properties (lamina and laminates for a range of fibre formats • Parametric analyses • Panel and beam design • Bonded and bolted double shear joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction
DLR	DLR-Mitteilung,Germany	Preliminary design of composite joints	• Adhesively bonded and bolted joints • Linear-elastic and linear-elastic/plastic behaviour • Tension and shear loading • Symmetric and asymmetric lap joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction. (washers and bolt tightening)
FELOCO	AEA Technology, UK	Finite element module computer code for predicting stresses and strains in adhesively bonded lap shear joints	• Stepped and profiled joints • Tensile/shear/bending moment/pressure loading • Linear and non-linear analysis • Peel, shear and longitudinal stress predictions in adhesive layer and adherends • Thermal stress analysis for adherend and adhesive
PAL	Permabond, UK	"Expert" system for adhesive selection	• Joined systems include: Lap and butt joints, Sandwich structures, Bushes/gears/bearings/shafts/pipes/threaded fittings • Elastic analysis • Creep/fatigue effects on joint stiffness (graphical)
RETCALC	Loctite, UK	Interactive windows based software General purpose	• Joint strength • Correction factors (temperature and fatigue)

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ESDU Data Sheets (Engineering Sciences Data Services)

ESDUpac Number	Application	Features
ESDU 78042	Shear stresses in the adhesives in bonded joints. Single step double lap joints loaded in tension	• Single-step double lap joints • Experimentally derived stress-strain curves for 3 adhesives • Adhesive behaviour modelled using shear-lag analysis • Average shear stress corresponding to peak strain
ESDU 79016	Inelastic shear stresses and strains in the adhesives bonding lap joints loaded in tension or shear	• Computer program • Stress and strain distributions in multi-step joints • Shear-lag analysis • Balanced (symmetric), rigid joints
ESDU 80011	Elastic stresses in the adhesive in single step double lap bonded joints	• Single-step double bonded lap joints • Applicable to thin bonded joints • Tension and compression loading • Peak elastic shear and direct tensile stresses in adhesive
ESDU 80039	Elastic adhesive stresses in multi-step lap joints loaded in tension	• Computer program • Single and double lap joints • Tension loading • Allowance for dissimilar adherends • Assumes adhesive and adherend elastic behaviour • Elastic shear and normal stress distributions
ESDU 81022	Guide to the use of data items in the design of bolted joints.	• Adhesive material property requirements and use in design of bonded joints
ESDU 85034	Flexibility of a single bolt shear joint	• Single and double bolted lap joints • Aluminium, steel or titanium alloy adherends • Steel or aluminium bolt • Stiffness predictions
ESDU 85035	Computer program for the flexibility of single and double lap thin plate joints loaded in tension	• Multi-bolt, single-row, single lap joints • Tension loading • Aluminium, steel or titanium alloy adherends • Steel or aluminium bolts • Bolt loads, joint extension and joint stiffness predictions

Design Requirements for Bolted Joints

This section covers the various aspects that need to be considered in the design of bolted joints for metallic and fibre-reinforced plastic (i.e. composite) materials. References to existing procedures and industrial codes of practice for design of mechanically fastened (bolted) joints are included. Consideration is given to design for static, fatigue and creep loading regimes. Relevant design guides are listed in the table below, although a complete review of all guidelines/standards was not possible as some documents, such as Naval Engineering Standards were restricted. Mechanical fastening using screws and/or rivets is not considered.

Design Considerations

Failure

Design Requirements for Bolted Joints

Summary of Design Guidelines for Bonded and Bolted Joints

Guideline	Summary
European Space Agency (ESA) Structural Materials Handbook [57]	Design and application guidance for polymer-based composites used for space structures. Data provided for materials appropriate for space applications.
EUROCOMP Design Code and Handbook [8]	Provides design rules and guidance for the structural design of buildings and civil engineering applications specifically using glass-fibre reinforced polymer composites. Covers the requirements for resistance, serviceability and durability of structures.
MIL-HDBK-17-1E - Volume 3 - Materials Usage, Design and Analysis [7]	Provides guidelines and material properties for polymer matrix composite materials mainly focussing on aerospace applications. Represents a compilation of relevant composites design, manufacture and analysis experience of engineers in industry, government and academia.
Composites Engineering Handbook [11]	State-of-the-art information, data and procedures for a wide range of topics on fibre-reinforced composites.
Joint Aviation Requirements (JAR) 25 - Large Aeroplanes [58]	Detailed and comprehensive aviation requirements aimed at minimising Type Certification problems. Details an acceptable basis for showing compliance with airworthiness codes. Does not include design guidance.
Aluminium Design Manual (The Aluminium Association, Inc.) [59]	Provides design information for determining the strength of aluminium structural components, safety and resistance factors for aluminium building and bridge structures, fatigue resistance (especially mechanically fastened connections), adhesive bonded joints, sandwich panels and beams, extrusion design, corrosion prevention, fire protection, references, and other design codes for aluminium structural components.

Design Considerations for Bolted Joints

Introduction | Material Parameters | Bolt Parameters | Joint Configuration

Design Considerations for Bolted Joints

The use of bolts is considered an effective means of fastening load-carrying members only when careful consideration is given to:

- Tensile and bending stresses of components
- Strength and stiffness of bolts
- Loss of tensile strength in the component due to machining of holes
- Shear distribution in the joint
- Friction between parts
- Residual stresses
- Allowable bearing stresses
- Type of bolt
- Fatigue behaviour
- T-T force applied to adherends through the bolt

The following sections provide more in depth information as to a variety of factors effecting the effectiveness and efficiency of bolted joints.

Material Parameters>>

Design Considerations for Bolted Joints

Introduction | Material Parameters | Bolt Parameters | Joint Configuration

Material Parameters

When designing bolted joints a variety of material parameters must be considered. These are listed below.

- Material type (e.g. metals and composites)
- Fibre type (e.g. carbon, glass and, aramid)
- Fibre format (e.g. unidirectional, woven and non-crimp fabric)
- Fibre orientation
- Lay-up or laminate stacking sequence
- Fibre volume fraction
- Form of construction (e.g. solid laminate and sandwich construction)

Fibre Type: For composite materials, the main fibre types considered are carbon and glass. Aramid fibres are not generally used in bolted joint configurations due to their low compression strength (inherent due to the fibrillated structure of the fibre) leading to poor bearing performance.

Lay-Up: Several of the guidelines reviewed provided recommendations on the lay-up of composite materials used in load carrying members in joints.

Guideline	Guidance
European Space Agency (ESA) Structural Materials Handbook [57]	• Inclusion of ± 45° plies to reduce stress concentration factors around holes in CFRP. • Optimum tensile properties obtained with a ratio of 0° and ± 45° plies of 2:1. Optimum shear strength is achieved with ratio of 1:1. • Recommends placement of 90° plies on surface of GFRP laminates
EUROCOMP Design Code and Handbook [8]	• Ideally, balanced symmetrical lay-up to be used with different orientation plies distributed throughout the thickness. • Preferably there should be 25% plies in 0°, 25% plies in 90° and 50% plies in ± 45°. • There should be at least 12.5% of plies in each of the four directions.
Composites Engineering Handbook [11]	• Lay-up chosen should be approximately quasi-isotropic (i.e. based on 0°, ± 45° and 90° plies). • Non-zero plies to reduce stress concentrations and avoid shear-out or cleavage failures. • 0° plies to carry main bearing and tensile loads.

Material Parameters Continued….

Design Considerations for Bolted Joints

Introduction | Material Parameters | Bolt Parameters | Joint Configuration

Material Parameters Continued

Some general points on laminate lay-up:

- Optimum performance is generally obtained with a quasi-isotropic (0°/90°/±45°) lay-up; in any one direction fibre proportions should be between 12.5 and 37.5%. For (0°/±45°) lay-ups, the proportion of ± 45° fibre plies should be between 37.5 and 75%.
- A composite laminate will most probably fail as a result of shear out if the lay-up is dominated by 0° fibre layers with only a few transverse (i.e. 90°) fibre layers.
- A minimum of 40% of ±45° plies should be used with a minimum of 10% of 90 plies to achieve highest bearing strength. For these lay-ups, w > 8d.
- Net-tension failure is influenced by the tensile strength of the fibres at the joint. Resistance to this mode of failure for multi-array bolted joints is maximised when the fastener spacing p > 4d along the direction of applied load and p > 5 across the width of the joint.
- For multi-array bolted composite joints, most of the load transfer occurs through the outer bolts.

Stacking Sequence is of minor importance for fully tightened bolts. It is recommended that grouping of plies of similar orientation be kept to a minimum in order to reduce interlaminar (through-thickness) tensile and shear stresses, which could reduce joint strength and fatigue resistance. Through-Thickness tensile and shear properties of composite materials are low in comparison with in-plane properties, and therefore structures may fail as a result of interlaminar stresses.

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Bolt Parameters>>

Design Considerations for Bolted Joints

Introduction | Material Parameters | Bolt Parameters | Joint Configuration

Bolt Parameters

Care should be taken when selecting the appropriate bolt material for use with composites, as galvanic corrosion can be problematic (i.e. carbon fibre-reinforced systems). Specially designed fasteners have been developed for use with composites in order to ensure that galvanic corrosion does not occur and that the full bearing capability of the composite is achieved.

Bolt/Washer Size: Recommendations on bolt, hole and washer sizes in order to achieve the maximum strength of a bolted joint are listed in the table below.

Design Guidance on Fastener Parameters

Guideline	Guidance
European Space Agency (ESA) Structural Materials Handbook [57]	• Holes should be reamed and bolts should be a good fit. • Washer hole size should be close to the size of the bolt shaft.
EUROCOMP Design Code and Handbook [[8]	• Clearance of hole should be large enough so that the bolt can be easily inserted into the hole even when all other fasteners have been tightened, but no more than 5% of fastener diameter. • Bolts should be self-locking or fitted with locknuts. • Washers fitted under head and nut of bolt should have a similar internal diameter as that of the diameter of the holes. Least external diameter of the washer shall not be less than twice the larger or largest diameter of the holes in the laminates through which the bolt passes. • Thickness of the washer shall not be less than 20% of the thickness of the outermost laminate through which the fastener passes.
Joint Aviation Requirements (JAR) 25 - Large Aeroplanes [58]	• The handbook provides a brief guidance on the type of fasteners to be used.

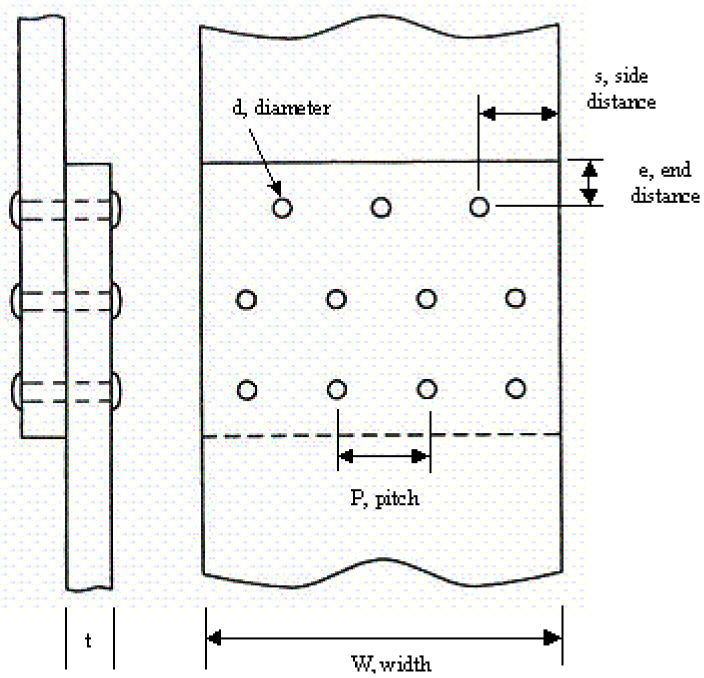
Bolt Parameters Continued….

Design Considerations for Bolted Joints

Bolt Parameters Continued

Clamping Force: The through-thickness force exerted on joined members is critical to the performance of the joint, especially for composite laminates. The clamping force exerted by torqued bolts, if sufficient, will suppress delamination driven failure modes. Increasing the clamping force will increase the bearing strength up to an optimum level. However, for joints under fatigue or creep loading, stress relaxation can lead to a reduction in clamping force. Hence for the purposes of design, most guidelines recommend that only a finger tight bolt torque value should be used (NB. For finger-tight bolts, a reduction of ~15% in bearing strength should be expected for quasi-isotropic lay-ups).

For metallic joints, the clamping force is also important, as joints can be designed as slip-critical connections, which resist shear by friction between the surfaces of the joined members. Slip critical connections are used when it is desirable to prevent movement of connected parts relative to each other. These types of connection are useful for joints that are subjected to fatigue loading.



Multi-array bolted joint with commonly used terminology. (rows 1 and 2 are staggered, rows 2 and 3 are uniform rectangular pattern)

Hole Patterns in Multi-Array Bolted Joints: A tandem row should be used in preference to a parallel row (see figure below). Unless the pitch is sufficiently large, the strength per fastener in a row joint will be less than that of a single bolt joint. Generally a pitch $p > 5d$ should be sufficient for joints with large safety factors. The pitch on outer rows should be greater than for the inner rows, to enable the joint to tolerate higher tensile loads. Multiple row joints minimise damage due to bending, but minimise axial load carrying capacity of the joint. It is advised to use double lap joint configurations in preference to single-lap joints in order to minimise bending forces.

Note: Generally, most efficient multi-row bolted joints are approximately 50% as strong as the unnotched parent material.

Diameter/Thickness Effect: It is advised that extremely thin laminates (< 1 mm thick) be reinforced and that $d < 4t$. A ratio of $d < 3t$ is generally satisfactory for most carbon fibre-reinforced laminates.

Design Considerations for Bolted Joints

Joint Configuration

A number of the guidelines provide details on the stacking sequence and configuration of bolted joints in order to minimise the possibility of the occurrence of undesirable failure modes. A summary of guidance given in the documents is presented in the table below.

Design Guidance on Joint Configuration

Guideline	Guidance
European Space Agency (ESA) Structural Materials Handbook [57]	Provides guidance on the effects of end-distance, width, row and pitch distances, hole patterns and multiple rows of bolts.Recommends values of ratios for e/d, w/d and d/t.
EUROCOMP Design Code and Handbook [8]	Guidance given on minimum distances between holes and minimum end-distances.Recommends values of d/t, e/d ratios.
Composites Engineering Handbook [11]	Provides general discussion on the effects of parameters such as w, d, t and e.
Joint Aviation Requirements (JAR) 25 - Large Aeroplanes [58]	No guidance provided.
Aluminium Design Manual (The Aluminium Association, Inc.) [39]	Guidance on spacing and edge distances for bolts.
Lloyd's Register of Shipping - Rules and Regulations for the Classification of Ships [60]	No guidance provided. Requires bolt configuration to be specified for certification.

Failure>>

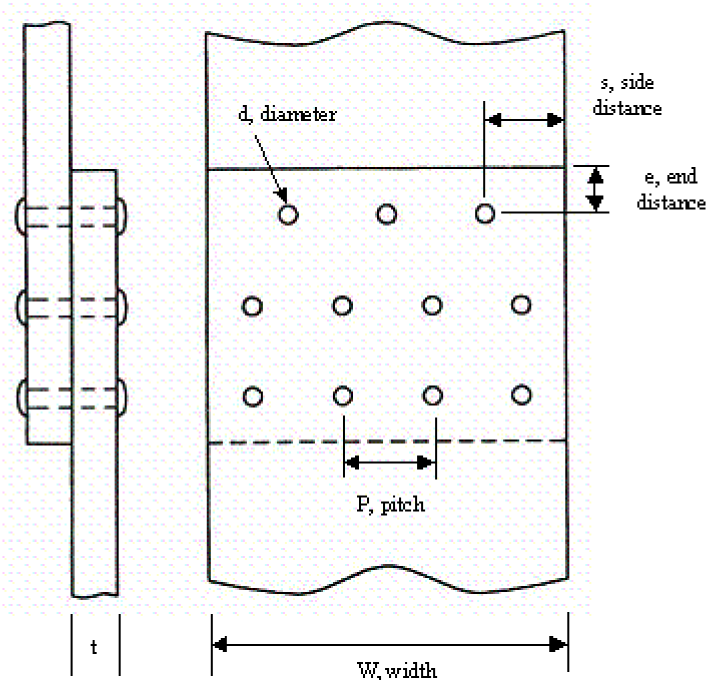
Failure

Failure Modes | Failure Criteria | Analysis for Different Failure Modes

Failure Modes

The four main failures modes (see table below) that are encountered are:

- Shear out
- Tension (or net-tension/section)
- Bearing
- Cleavage



However, mixed mode failures frequently occur. Failure is usually gradual with progressive damage eventually resulting in loss of load bearing capacity.

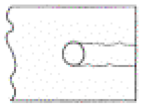
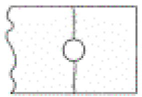
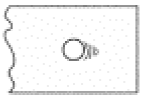
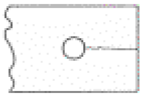
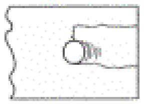
The following points should assist in either avoiding or minimising the occurrence of various failure modes (see terminology in figure right).

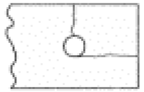
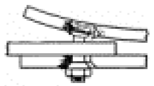
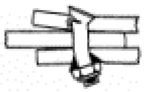
- Shear out can be avoided by having adequate end distance (e > 4d).
- Cleavage failure can be avoided by having adequate width and end distance (w > d and e > 4d).
- Pull through is more likely to occur with countersunk bolt heads and thin substrates. Use the maximum possible head angle (120° and 130°).
- Fastener shear failure can be avoided by selecting the correct bolt diameter and material. Checks need to be made to ensure the bolt has sufficient stiffness and strength to meet the design requirements. Bending failure of the bolt is usually avoided provided d/t > 1.

Tension and bearing failure are essentially dependent on material properties of the substrate (NB. Bearing failure is usually benign (non-catastrophic) and tension failure is generally catastrophic).

Failure Criteria>>

Typical Failure Modes for Bolted Joints

Failure Mode		Comment
Shear out		Caused by shear stresses and occurs along shear out planes on hole edge, typical failure mode when end distance is short.
Tension (net-section)		Caused by tangential tensile or compressive stresses at the edge of the hole. For uniaxial loading conditions, failure occurs when bypass/bearing stress ratio is high (or d/w is high).
Bearing		Occurs in area adjacent to contact area due to compressive stresses, likely when bypass/bearing stress ratio is low (or d/w is low), strongly effected by through-thickness clamping force.
Cleavage		
Bearing/shear out		Mixed-mode

Failure Mode		Comment
Bearing/tension/shear out		
Tension/shear out		
Bolt pull-through		Due to low through-thickness strength of composite material.
Bolt shear failure		Cause by high shear stresses in the bolt.

Tension/Shear Out

Bearing Failure

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National Physical Laboratory

Failure

Failure Criteria

When designing a bolted joint, all possible failure modes must be considered, evaluated and their chances of occurring minimised (through selection of appropriate edge distances, widths, etc.). In most cases, joint design will be such that the net-tension or bearing failure modes will be most probable as net tension provides the highest strength whilst the bearing mode offers a less catastrophic, more progressive failure.

The definition of failure varies widely, which confuses the issue of defining allowable design strength. An approach frequently used is based on the degree of hole deformation under load, however there is a lack of agreement over the allowable level of hole deformation. Values vary between 0.5% to 4% of the un-deformed hole diameter. A safety factor of 2 is commonly applied to the ultimate strength, which is close to the stress at which damage is initiated. Other definitions of failure criteria are based on:

- Onset of damage
- Degree of damage
- Load bearing capacity of the joint

Analytical Methods

Designers/engineers in the aerospace industry tend to use a two-stage approach to structural analysis, whereby the loads in the individual bolts are calculated and then stress analysis is performed on the load transfer for each bolt deemed critical. The second stage of this analysis approach effectively takes the form of modelling a single bolt in a plate.

Analytical methods for modelling the behaviour of bolted joints can be categorised in either of the following groups:

- Two-dimensional classical elasticity methods with complex variables; or
- Full, three-dimensional finite or boundary element analyses (FEA/BEA).

Failure Criteria Continued….

Failure

Failure Criteria Continued

Two-dimensional stress analysis techniques developed for calculating the strength of bolted joints in isotropic and orthotropic plates do not account for the following factors:

- Through-thickness clamping effects
- Free edge effects at open holes
- Influence of delaminations in redistributing the load.

Having calculated the stress distribution in a bolted joint, the next step is to predict failure. Several failure criteria have been formulated. The main approaches used are listed below.

Average Stress Criterion: Failure is considered to have occurred when the average tensile stress over a certain distance from the hole reaches the un-notched strength of the laminate.

Point Stress Criterion: Failure is considered to have occurred when the local value of tensile stress reaches the un-notched tensile strength of the laminate at a certain distance from the hole.

Yamada Failure Criterion: This approach assumes failure occurs when every ply has failed due to fibre cracking. The condition for failure in any ply is given by the following quadratic relationship [57]:

$$\left(\frac{\sigma_{xx}}{X}\right)^2 + \left(\frac{\sigma_{xy}}{S_c}\right)^2 = e^2 \quad \text{(if } e < 1 \text{ no failure occurs and if } e \geq 1 \text{ failure occurs)}$$

where σ_{xx} and σ_{xy} are the longitudinal and shear stresses in a ply, X is the longitudinal tensile strength of the ply, Sc is the shear strength of a symmetric, cross-ply laminate having the same number of plies as the laminate under consideration.

Failure Criteria Continued….

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Failure

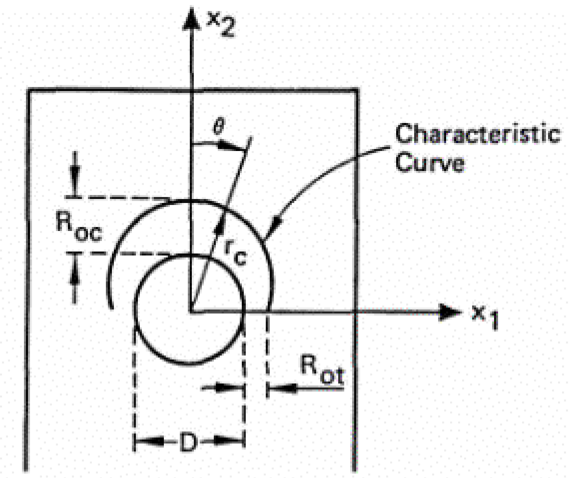
Failure Modes | Failure Criteria | Analysis for Different Failure Modes

Failure Criteria Continued

The Yamada criterion is generally used with a failure hypothesis, such as the Whitney/Nuismer point stress criterion [61]. The hypothesis assumes that failure will occur when the stresses in any ply of a laminate satisfy the Yamada criterion on a characteristic curve around a loaded hole. The characteristiccurve shown in the figure below is described by the following relationship:

$$r_c(\theta) = \frac{D}{2} + R_{ot} + (R_{oc} - R_{ot})\cos\theta$$

where $-\pi/2 \le \theta \le \pi/2$, and R_{ot} and R_{oc} are the characteristic lengths for tension and compression



Whitney/Nuismer failure hypothesis [57].

The parameters R_{ot} and R_{oc} depend on the material only and are determined experimentally. When coupling the Yamada failure criterion with the Whitney/Nuismer hypothesis, failure will occur at any point on the characteristic curve (i.e. $r = r_c$) when $e > 1$.

For a full theoretical description of the bolted joint problem, factors such as friction between the bolt and hole, contact areas (modelled using slide lines in FEA packages such as LUSAS to allow intermittent contact) and through thickness clamping effects must be included. Three-dimensional FEA models have been used for bolted joint analysis and good agreement with experimental results have been achieved. FEA modelling (particularly 3-D analysis), however, requires significant effort and time for meshing the geometry and considerable computer time to perform the analysis. Despite these factors, 3-D FEA is recommended for bolted joint configurations.

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Analysis>>

Failure

Analysis for Different Failure Modes

TENSILE FAILURE

The maximum tensile stress σ_{MAX} at the edges of a loaded hole in a joint under an arbitrary load P is given by [11]:

$$\sigma_{MAX} = \frac{K_{tc}P}{t(w-d)}$$

where K_{tc} is the effective stress concentration factor, based on net-section.

The joint strength efficiency, expressed as the ratio of the load capacity of the joint P, to the no-hole load capacity is given by:

$$\frac{\sigma_u}{\sigma_{tu}} = \frac{P}{\sigma_{tu}wt} = \frac{1-d/w}{K_{tc}}$$

The following empirical relationship can be used to determine elastic stress concentration:

$$K_{te} = \frac{w}{d} + 1 - 1.5\left(\frac{w/d-1}{w/d+1}\right)$$

For an elastic isotropic material, the ratio of joint strength to basic strength is given by:

$$\frac{P}{\sigma_{tu}tw} = \frac{1}{\frac{2}{1-d/w} + \frac{1}{d/w} - \frac{1.5}{1+d/w}}$$

For composites:

$$K_{tc} - 1 = C(K_{te} - 1)$$

The value of C in the above relationship is 0 for ductile metals (e.g. aluminium) and " 0.27 for quasi-isotropic laminates. For orthotropic laminates connected with 6.5 mm diameter bolts, the value of C can be estimated as shown below. This relationship can be used provided that the mode of failure does not change.

$$C \approx (\%0^\circ \text{plies})/100$$

Continued…..

Failure

Analysis for Different Failure Modes Continued

BEARING FAILURE

The bearing capacity of a metallic joint is generally based on the nominal bearing strength σ_b using the relationship [11]:

$$P = \sigma_b dt$$

The joint efficiency ratio for bearing failure is given by:

$$\frac{P}{\sigma_b wt} = \frac{d}{w}$$

SHEAR FAILURE

For metallic and glass fibre-reinforced materials, the maximum load capacity in shear for a can be estimated using the following expression [11]:

$$P = \tau_{MAX} st$$

CLAMPING PRESSURE

For un-lubricated steel bolts loaded in tension by means of an applied torque, the tensile load in the bolt Pt is given by [11]:

$$P_t = \frac{T}{Kd}$$

where T is the applied torque, d is the hole diameter and K is a torque coefficient. K has been measured to have a constant value of 0.2 for all bolt diameters. This equation can be rewritten as:

$$P_t = \frac{5T}{d}$$

Transverse compressive stress or clamping stress σ_z (i.e. lateral constraint) is a function of washer diameter D and hole diameter d as follows:

$$\sigma_Z = \frac{P_t}{\pi/4\left(D^2 - d^2\right)}$$

The recommended washer diameter D is given by:

$$D = 2.2d$$

Substituting the equations for P_t and σ_z into the second equation for P_t, this equation can be rewritten in terms of applied torque T and hole diameter as shown below:

$$\sigma_z = 1.658\frac{T}{d^3}$$

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Continued…..

Failure

Analysis for Different Failure Modes Continued

SINGLE ROW BOLTED JOINTS

The following analysis applies to plates of the same material with equal thickness and width and connected together with a single row of bolts.

BOLT SHEAR FAILURE

Shear failure will occur when the load P on a single bolt results in a shear stress that exceeds the shear strength τ_{MAX} of the bolt. The maximum shear force P_{MAX} that the bolt can withstand is defined by the following relationship:

$$P_{MAX} = \frac{\pi d^2 \tau_{MAX}}{4p}$$

where d is the diameter of the bolt and p is the pitch (i.e. distance between the hole centres).

BEARING PRESSURE

The maximum bearing pressure that a bolted joint can withstand is given by:

$$P_{MAX} = \frac{P_b td}{p}$$

where P_b is the pressure at which either the bolt or the hole fails in bearing.

MULTI-ROW LAP JOINTS

In design, it is customary to allow for higher stress concentration at a loaded hole than at an unloaded hole. Also, a distinction must be made between bearing and non-bearing or by-pass loads, to be able to characterise the internal loads in multi-row joints. The following problems need to be considered when designing multi-row bolted joints (see Tate and Rosenfeld [63]):

- Distribution of load in the pins, plate and straps (by-pass loads)
- Prediction of the stress concentration in a pin-loaded hole
- Prediction of the failure load for the joint

The reader is referred to NPL CODA (Composite Design and Analysis) software, which enables the user to determine the load sharing between individual bolts in a line of bolts in a symmetrical lap joint.

Cyclic Fatigue>>

Cyclic Fatigue

The fatigue performance of bolted composite joints is generally superior to that of metal joints, although occasional overloads are known to compromise the fatigue life of composite joints. CFRP bolted joints can be expected to exhibit considerable fatigue life under high stress fatigue loads with minimal reduction in residual strength. The fatigue lives of the bolts play a significant role in determining the fatigue lives of bolted joints, as the fasteners will frequently fatigue before the substrate materials.

Key Points

- Bolted joints perform best under compression-compression fatigue (R = -¥), less well under tension-tension ((R = 0.1) and worst under fully reversible loading tension-compression (R = -1).
- The predominant failure mode under cyclic loading is usually bearing failure, which is in the form of hole elongation (bolt hole is elongated by wear during fatigue). Other forms of failure include bolt failure, net-section and shear-out (see table below).
- Rate of hole elongation is dependent on clamping pressure. Increasing the clamping pressure will reduce hole elongation and increase fatigue life. A higher clamping pressure will increase the amount of load transferred between the plates through friction and thereby reducing the pressure on the hole surface from the bolt shank.
- Increasing the clamping pressure will increase bearing strength, although improvement in fatigue performance in real terms will only be small (i.e. when fatigue load is normalised with respect to the ultimate bearing strength).
- Avoid using countersunk fasteners - fatigue threshold load will be reduced compared with protruding head fasteners.
- Conditions required for maximising joint strength under static loading conditions usually also apply to fatigue performance. However, there are limited data available as to the influence of key variables to be confident that the two are inseparable (i.e. coupled).
- Fretting (i.e. wear of the bearing surface) and stress relaxation may pose particular problems, which may not necessarily improve when optimising joint strength under static loading conditions. Wear of the bearing surface will lead to an increase in clearance and a modification in bolt stiffness and clamping pressure. Considerable heat can also be generated through fretting, which can cause thermal damage to composite laminates (including softening and charring of the matrix).

Cyclic Fatigue Continued….

Cyclic Fatigue

Cyclic Fatigue Continued

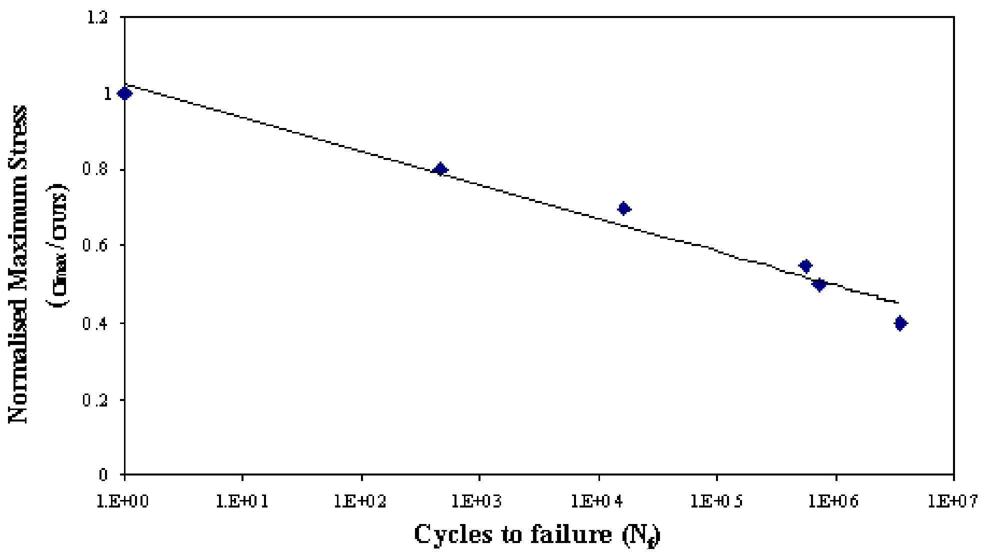
Normalised S-N curves for tension-tension (R = 0.1 and f = 5Hz) fatigue of undamaged, pin-bearing and open-hole tension coupon specimens can be approximated by the following relationship:

$$\sigma_{MAX} / \sigma_{ULT} = 1 - k \log N_f$$

where σ_{MAX} is the maximum load applied to the specimen, σ_{ULT} is the ultimate strength of the joint, Nf is the number of cycles to failure and k is the fractional loss in strength per decade of cycles. The value of k is a measure of fatigue resistance and is similar for the three different specimen types (see table below).

Values of k for Different Structural Elements and Composite Materials [62]

Material	Pin bearing	OHT	Tension
Pultruded glass fibre/ polyester	0.10	0.09	0.10
Quasi-isotropic carbon-fibre/epoxy	0.07	0.06	0.06
Glass fibre/polypropylene	0.08	-	0.08
Woven glass fibre/epoxy	0.09	0.10	0.10



S-N curve for pin bearing woven glass-fibre/epoxy [62].

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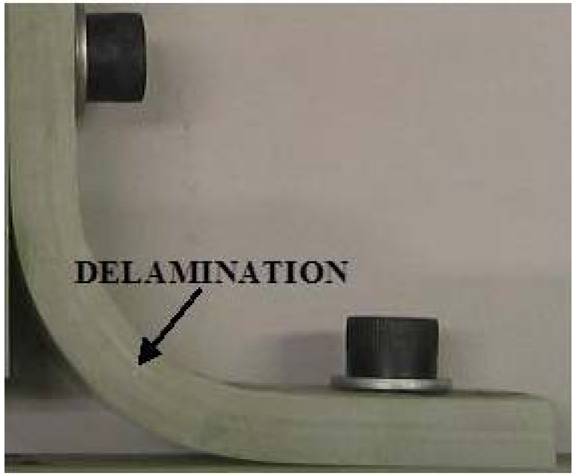
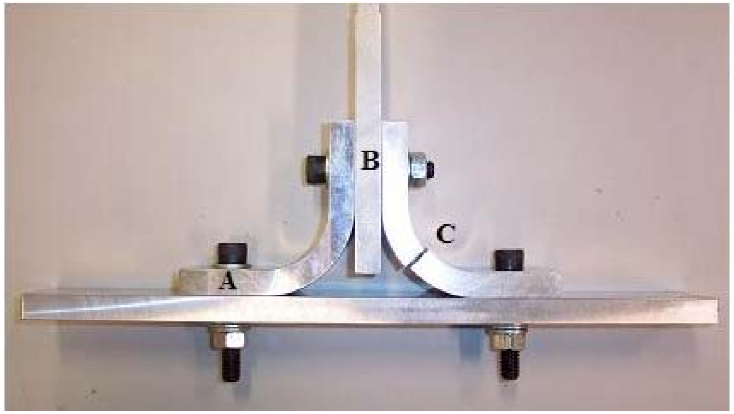
T-Joint Example>>

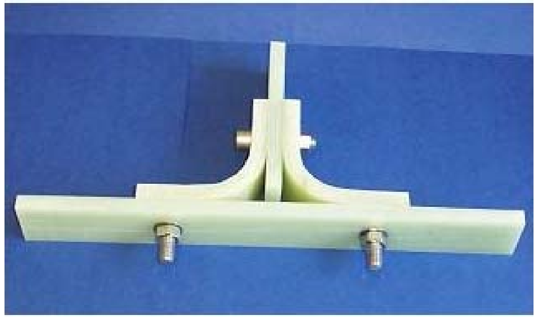
Bolted T-Joint Analysis

As with most joints, multiple modes of damage can occur in bolted T-joints. In the case of composite laminates, delaminations may occur in the vicinity of the bolts securing the flanges to the base plate as a result of combined through-thickness shear and tensile stresses. Crack growth is rapid, propagating around the flange circumference resulting in catastrophic failure of the structure. Although other forms of localised damage (e.g. longitudinal splitting and bearing failure) occur around bolt holes prior to delamination formation, these modes of failure are relatively benign. This applies equally to quasi-static and cyclic fatigue loading conditions. Care needs to be taken to account for the possibility of premature interlaminar failure of the composite laminate. These materials have low through-thickness tensile and shear strength (see Yamada Failure Criterion).

Metallic T-joints may fail as a result of tensile cracking at the apex of the flange (location C). The onset of bearing failure may occur in the vicinity of the bolts connecting the flanges to the base plate (location A) and to the central web (location B) prior to tensile failure of the flange. Again bearing failure will have occurred at lower loads, but with minimal effect on joint properties.







Design Software>>

Design Software

FEA Programmes| Analytical Software

Techniques for stress analysis of a joint generally fall into two main categories: analytical, closed-form methods and finite element methods. Analytical methods are generally quick and easy-to-use, but are only suitable for simple geometries. These methods cannot accurately predict stresses and strains as the analytical equations, by their simple nature, cannot fully account for the complete stress and strain conditions within the joint. Finite element methods have an advantage in that almost any geometrical shape can be analysed and are capable of more accurate analysis of stress and strain distributions. The disadvantages of these methods are that analyses are expensive and specialist knowledge is required. An ideal method would be an accessible yet accurate stress analysis technique. This section reviews finite element analysis (FEA) and analytical based software developed for the analysis and design of bolted and bonded structures, and materials selection.

FEA Programs

Numerical analysis techniques, such as FEA, are used extensively in the design and stress analysis of adhesively bonded and bolted structures. These techniques offer solutions to complex problems that are too difficult or impossible to resolve using analytical, closed-form solutions. Numerous FEA codes are available (see table below). These codes provide in-built constitutive models for simulating the behaviour of most adhesives, allowing for non-uniform stress-strain distributions, geometric non-linearity, hygrothermal effects, elastic-plastic and visco-elastic behaviour, static and dynamic analysis, and strain rate dependence. Orthotropic element types include two-dimensional (2-D) solid plane-stress or plain-strain elements, axisymmetric shell or solid elements, three-dimensional (3-D) solid or "brick" elements and crack-tip elements. A number of automatic mesh (element) generators are available with post-processing capabilities (e.g. PATRAN and FEMGV).

Although numerical methods are able to accommodate complex geometries, loadings, material properties and boundary conditions, the solutions generated are only approximations to the actual solution. It is important that the designer/analyst is aware of the limitations of the numerical techniques being applied and has a fundamental understanding of the mechanics of bonded and bolted joints (i.e. stresses and failure mechanisms).

Stress analyses (especially FE methods) are often used to compare stress/strain distributions obtained from different joint configurations (e.g. lap, scarf and butt joints) or geometries (varying adhesive and adherend thickness, overlap lengths, fillet shapes). Hence, finite element stress analysis can be used as a tool for optimising the design of a joint. Evolutionary optimisation method **EVOLVE** has been used to optimise the shape of adhesive fillets [64]. This process allows selected properties to drive the optimisation process (e.g. minimising the maximum principal stress in the adhesive). **EVOLVE** relies on an iterative FE analysis and the progressive removal of elements using a rejection criterion. This takes the guesswork out of the design process.

Design Software Continued….

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Finite Element Packages (see also [57])

Name	Supplier	Application	Features
ABAQUS	Hibbit, Karlsson & Sorenson, Inc.	• General purpose FE program • Linear, non-linear and coupled analysis • Large materials model library	• Anisotropic material models in all elements • 2-D and 3-D plate/shell and solid elements • Temperature and strain-rate dependence of properties • Fracture mechanic/ crack propagation analysis • Maximum stress and strain, Tsai-Hill, Tsai-Wu, Azzi-Tsai-Hill and user defined failure criteria
ANSYS	Swanson Analysis System Inc.	• General purpose FE program • Non-linear analysis (non-composite applicable) • Pre- and post-processing	• Isotropic and orthotropic material properties • 2-D and 3-D plate/shell and solid elements • Laminated shell elements • Crack-tip solid and thick-shell elements • Maximum stress and strain, Tsai-Wu and user defined failure criteria
LUSAS	FEA Ltd.	• General purpose FE program • Linear and non-linear analysis • Static, creep, fatigue and dynamic analysis	• Plate/shell and solid elements • 2-D and 3-D interface elements • Laminate analysis/Hashin damage model • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill, Hoffmann, Hashin, Tsai-Wu failure criteria
COSMOS	Structural Research& Analysis Corp.	• General purpose FE program • Linear and non-linear analysisStatic and dynamic analysis • 2-D and 3-D structural and thermal models	• Plate/shell and solid elements • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill and Tsai-Wu failure criteria
NASTRAN	MacNeal-SchwendlerCorp.	• General purpose FE program • Static and dynamic analysis • Linear and non-linear analysis • Pre- and post-processing by PATRAN	• Plate/shell elements - allowance for orthotropic temperature-dependent properties • Ply stresses • Tsai-Hill, Hoffmann and Tsai-Wu failure criteria
NISA	Engineering Mechanics Research Corp.	• General purpose FE program • Linear, non-linear and coupled analysis	• Solid and thick-shell elements • Laminated shell elements • In-plane and interlaminar stresses • Maximum stress and strain, von Mises, Tsai-Wu, delamination failure criteria

Design Software

FEA Programmes| Analytical Software

Analytical Software

In the literature and design guides there are many examples of analytical, closed-form solutions for obtaining stress and strain distributions. These analytical models are generally based on modified shear-lag equations. Aside from the shear-lag analysis technique, other workers have carried out stress analyses using a variety of other methods, such as those based on Hashin's variational analysis using the principle of minimum complementary energy [65]. Reviews of these analytical theories and their assumptions have been published [66-67].

As the analytical equations have become more complex (including factors such as stress variation through the adhesive thickness, plasticity, thermal effects, etc.), there is a greater requirement to use computing power to solve for the stresses. Hart-Smith [52, 68-70] has had a great influence on the methods used for stress analysis of adhesive joints and much of his work is evident in the **P**rimary **A**dhesively **B**onded **S**tructure **T**echnology (**PABST**) programme. Versions of this method (e.g. A4EG, A4EH, A4EI) have been prepared as FORTRAN programmes and have been used extensively in the aerospace industry. Other analyses have been implemented in spreadsheets or as a programme for personal computers (e.g. **JOINT** [71]).

Although simplified analytical procedures for designing adhesively bonded joints are available in the form of PC compatible software [72], these packages are limited in number and scope. As with all design tools, the effectiveness of the analysis is directly related to the users knowledge, and therefore it is advisable that the user has a good understanding of engineering design and material behaviour. The software packages are there to assist in the design of efficient joints. A brief overview of commercial PC based analysis/design software packages is given in the table below. The main features of each software package are identified.

Engineering Sciences Data Unit (ESDU) provide a comprehensive range of data sheets and software for use in structural design, including analysis of bonded and bolted metallic structures (see table below). **ESDU** data sheets also cover circular holes in orthotropic plates, laminated composite materials and structures (including pipes, beams and sandwich panels), and fatigue endurance of metallic structures. The information is provided primarily for use in aerospace structures, but has wider application to other areas of engineering. The information is accepted by the Federal Aviation Administration (FAA) in the United States and by the Civil Aviation Authority (CAA) in the United Kingdom as a basis for submissions. In most cases, PC software is available with computer listings.

The production of the design data and software is monitored and guided by expert committees of professionally qualified engineers from industry, research laboratories and universities. Items are continuously checked and updated to include the latest amendments, which are available to software leasers on request. Each program has a main menu enabling the user access to different input screens (e.g. material properties, loading conditions and results). The **ESDU** software is relatively sophisticated compared with most analytical packages. To maximise the software benefits requires a good understanding of engineering design.

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PC Based Software Packages (see also [73])

Name	Supplier	Application	Features
BOLT	G.S. SpringerStanford University	Design of pin-loaded holes in composites	• Prediction of failure strength and failure mode • Three types of bolted joints: Joints with a single hole, Joints with two identical holes in a row Joints with two identical holes in tandem • Applicable to uniform tensile loads and symmetric laminates
BISEPS-LOCO	AEA Technology,UK	Closed form computer code for predicting stresses and strains in adhesively bonded single-lap joints	• Tensile/shear/bending moment loading • Adhesive peel and shear stress predictions • Allowance for plasticity in adhesive layer • Thermal stress analysis
BISEPS-TUG	AEA Technology, UK	Closed form computer code for predicting stresses and strains in adhesively bonded coaxial joints	• Stepped and profiled joints • Orthotropic adherends • Torsional and axial loading • Allowance for plasticity in adhesive layer • Thermal stress analysis
CoDA	National Physical Laboratory, UK.	Preliminary design of composite beams and panels, and bolted joints	• Synthesis of composite material properties (lamina and laminates for a range of fibre formats • Parametric analyses • Panel and beam design • Bonded and bolted double shear joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction
DLR	DLR-Mitteilung,Germany	Preliminary design of composite joints	• Adhesively bonded and bolted joints • Linear-elastic and linear-elastic/plastic behaviour • Tension and shear loading • Symmetric and asymmetric lap joints • Bearing, shear-out, pin shear and by-pass tensile failure prediction. (washers and bolt tightening)
FELOCO	AEA Technology, UK	Finite element module computer code for predicting stresses and strains in adhesively bonded lap shear joints	• Stepped and profiled joints • Tensile/shear/bending moment/pressure loading • Linear and non-linear analysis • Peel, shear and longitudinal stress predictions in adhesive layer and adherends • Thermal stress analysis for adherend and adhesive
PAL	Permabond, UK	"Expert" system for adhesive selection	• Joined systems include: Lap and butt joints, Sandwich structures, Bushes/gears/bearings/shafts/pipes/threaded fittings • Elastic analysis • Creep/fatigue effects on joint stiffness (graphical)
RETCALC	Loctite, UK	Interactive windows based software General purpose	• Joint strength • Correction factors (temperature and fatigue)

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ESDU Data Sheets (Engineering Sciences Data Services)

ESDUpac Number	Application	Features
ESDU 78042	Shear stresses in the adhesives in bonded joints. Single step double lap joints loaded in tension	• Single-step double lap joints • Experimentally derived stress-strain curves for 3 adhesives • Adhesive behaviour modelled using shear-lag analysis • Average shear stress corresponding to peak strain
ESDU 79016	Inelastic shear stresses and strains in the adhesives bonding lap joints loaded in tension or shear	• Computer program • Stress and strain distributions in multi-step joints • Shear-lag analysis • Balanced (symmetric), rigid joints
ESDU 80011	Elastic stresses in the adhesive in single step double lap bonded joints	• Single-step double bonded lap joints • Applicable to thin bonded joints • Tension and compression loading • Peak elastic shear and direct tensile stresses in adhesive
ESDU 80039	Elastic adhesive stresses in multi-step lap joints loaded in tension	• Computer program • Single and double lap joints • Tension loading • Allowance for dissimilar adherends • Assumes adhesive and adherend elastic behaviour • Elastic shear and normal stress distributions
ESDU 81022	Guide to the use of data items in the design of bolted joints.	• Adhesive material property requirements and use in design of bonded joints
ESDU 85034	Flexibility of a single bolt shear joint	• Single and double bolted lap joints • Aluminium, steel or titanium alloy adherends • Steel or aluminium bolt • Stiffness predictions
ESDU 85035	Computer program for the flexibility of single and double lap thin plate joints loaded in tension	• Multi-bolt, single-row, single lap joints • Tension loading • Aluminium, steel or titanium alloy adherends • Steel or aluminium bolts • Bolt loads, joint extension and joint stiffness predictions

National Physical Laboratory

Manufacture and Assembly of Bonded Joints

Manufacture and Assembly of Bonded Joints

The reliability of a bonded joint depends not only on selecting the correct adhesive, but also on the preparation of the adherends, mixing of the adhesive, joint assembly and the curing process. It is worth noting that a high percentage of failures can be attributed to poor joint manufacture or a lack of understanding of that factors that influence joint performance. These problems can be minimised or eliminated through proper training and education. This section examines the key issues relating to the preparation and assembly of adhesive joints.

Manufacture and Assembly of Bonded Joints

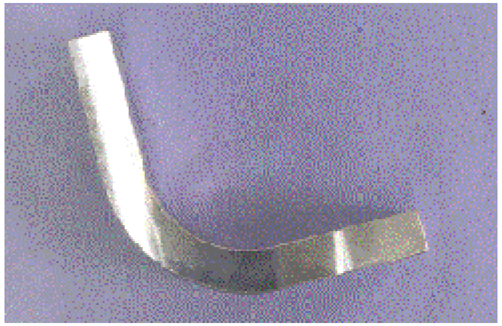
Manufacture and Machining of Adherends

Adherends should be manufactured and/or machined accurately to ensure specimen dimensions meet design specifications. It is important to ensure that the adherends are free of any edge or surface damage. The surfaces to be bonded must be parallel and flat to ensure uniform, intimate contact, across the entire bond area when the two surfaces are clamped or pressed together. The operator should ensure that during the machining process, no nicks, cuts or scratches are introduced at the edges or surfaces of the adherends. Surface or edge defects can cause premature failure of the adhesive joint and/or the adherend. It is advisable before preparing the surface to ensure that the adherend sections to be bonded fit together well with the bonded surfaces closely matching (i.e. intimate contact between the two surfaces).

Note: Background information on all aspects of machining of fibre-reinforced plastic composites is given in Measurement Good Practice Guide 38 "Fibre Reinforced Plastic Composites – Machining of Composites and Specimen Preparation" [20].

Thin Metallic Sections: Guillotining thin metal sheets is a rapid and low cost method for producing large quantities of thin adherend sections, however the cutting operation can result in bending of the adherends and operators will therefore need to be ruthless by discarding those specimens that fail to meet the specification.

Thick Curved Metallic Sections: Bending metal sections to shape frequently result in the metal being stressed beyond its yield point. It is advised that thick aluminium or steel curved sections, such as right angle flanges (see figure right), should be milled from a solid block of material. Although corner damage can be eliminated by either milling or spark eroding the adherend, springback can occur due to recovery of in-built residual elastic strains. After the deforming forces have been relieved, the metal part has a permanent set, which is less than the angle that was machined. The difference between the permanent angle of bend and the maximum angle, to which the metal was forced, is commonly known as springback. The effect of springback can be allowed for by first machining the flange so that it is oversized. The oversized section will undergo springback, but as there is sufficient excess material remaining it enables the machinist to finish milling the section to the final shape and dimensions. Spark eroding the adherends to shape will also result in springback, however it is generally quicker and more convenient to mill curved sections.



Aluminium right angle flange

Adherends Continued…..

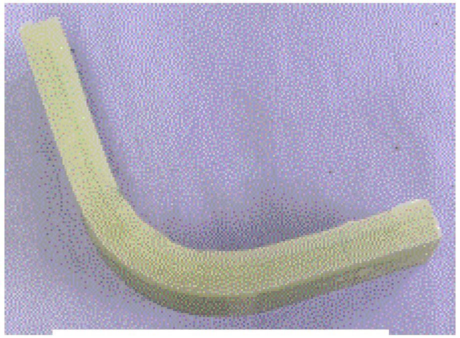
Manufacture and Assembly of Bonded Joints

Manufacture and Machining of Adherends Continued

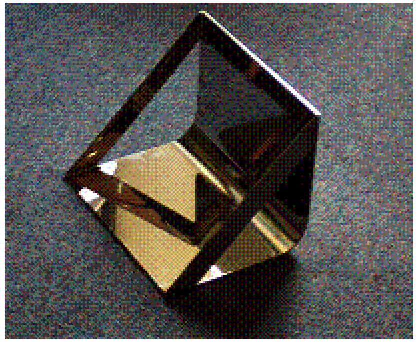
Curved Composite Laminated Sections:
In order to manufacture curved laminated sections a mould is required (see figures right). The shape of the mould needs to be determined in advance allow for springback that occurs on cure of the composite component. Springback angle $\Delta\theta$ can be estimated using the following relationship:

$$\Delta\theta \cong [\alpha_{11}(T) - \alpha_{33}(T)]\Delta T$$

where
$\alpha 11$ = in-plane coefficient of thermal expansion
$\alpha 33$ = through-thickness coefficient of thermal expansion
ΔT = stress free temperature – ambient temperature



GRP right angle flange



Polished stainless steel autoclave mould

The first approach requires measurement of the in-plane and through-thickness coefficients of expansion using either dilatometry or strain gauges from room temperature to the cure temperature. As expansion coefficient is temperature dependent and highly non-linear for polymeric materials, it is necessary to calculate the cumulative value of $\Delta\theta$. More comprehensive analyses are available. FEA can also be used to predict $\Delta\theta$.

Alternatively, a right angle section can be manufactured with an identical lay-up and dimensions as the final product. The springback angle is then measured directly from the component. The angle of the mould is then adjusted so that it equals 90° + $\Delta\theta$. This is approach is more reliable than computing the springback angle.

- Springback is very sensitive to the method of manufacture. Important parameters are:
- Difference between Tg and room temperature (i.e. ΔT)
- Amount of cure or crystallisation shrinkage below Tg
- Gradients in fibre volume fraction or the development of resin rich layers (often related to the type of mould)
- Gradients in through-thickness temperature
- Rates of heating/cooling during processing (effects resin flow, final degree of cure, visco-elastic effects)

Adherends Continued….

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Manufacture and Assembly of Bonded Joints

Manufacture and Machining of Adherends Continued

Moisture Effects: Fibre-reinforced thermoset composites are known to absorb moisture in relatively benign environments. For example, epoxy based composite systems can absorb 0.2 wt % moisture in a laboratory environment (i.e. 23°C and 50% RH) within 2 to 4 weeks. The presence of moisture in the composite can adversely affect the properties of the adhesive during the cure process, and as a result the joint strength may be compromised. Moisture released from the composite substrate during cure will enter the adhesive, and has been known to reduce the glass transition temperature Tg by as much as 20°C, and lower the fracture toughness Gc of a rubber toughened epoxy adhesive by a factor of 10 [27]. With toughened adhesive formulations, the presence of moisture may inhibit phase separation of the rubber-toughening agent, thus preventing the formation of rubber-toughened particles.

Adherends (pre-dried) should therefore be stored in a dry area (i.e. dessicator or sealed container with a suitable dessicant). It is recommended that polymer composites be pre-dried in an oven maintained at 50 ± 2°C (unless otherwise specified) until the specimen weight reaches a constant value. The temperature of the drying oven should not exceed the maximum operating temperature of the polymeric-matrix. The SAE Aerospace Recommended Practice (ARP) ARP4977 [28] describes standard methods for drying commercial aircraft composite materials structures.

Surface Preparation>>>

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Manufacture and Assembly of Bonded Joints

Surface Preparation of Adherends

Surface preparation is recognised as the most critical step in the adhesive bonding process and considerable adhesive joint testing is performed to optimise surface treatment. The selection of surface treatment is largely dependent on the required strength and durability of the joint, although economic considerations, such as costs and time involved in preparation, also play a role in the selection process. Correct surface preparation is essential for good joint strength and maintaining long-term structural integrity of bonded joints. Unsatisfactory surface preparation will result in the bond failing adhesively and unpredictably at the adhesive/adherend interface.

The role of surface preparation is to remove surface contaminants (grease and dust), increase surface area for bonding, promote micro-mechanical interlocking, and/or chemically modify a surface. It is important that the process of surface preparation only affects the chemistry and morphology of thin surface layer of the adherend(s) and does not alter the mechanical and physical properties of the underlying substrate. NPL Measurement Good Practice Guide No 47 "Preparation and Testing of Adhesive Joints" [22] provides a brief description of general procedures required for preparing different substrates for adhesive bonding. More detailed information can be obtained from the Adhesives Tookit Website www.adhesivestoolkit.com and "Guide to The Structural Use of Adhesives" produced by The Institution of Structural Engineers [28]. Specific treatments can be found in BS 7079, BS EN 12768, ASTM D 2651, ASTM D 2093, BS EN 1840 and SAE ARP4916 [29-34] – see also [35-38]. Advice should be sought on surface preparation from the adhesive manufacturer. **Surface preparation procedures often require potentially hazardous or environmentally damaging chemicals. All preparation should be carried out to COSHH specifications (see also [22]).**

Note: After completion of the surface preparation process, the adherends must not be exposed to physical handling or uncontrolled atmospheric environments in order to prevent surface contamination prior to bonding. It is advisable that bonding be performed immediately following surface treatment to maximise performance. Table 1 provides a guide as to the relative quality and cost of various surface treatments. Environmental impact of the process may also need to be assessed when selecting a surface treatment. Clean grit, clean solvent and clean cloths must be used to avoid spreading contamination.

Table 1: Relative Cost and Quality of Various Surface Treatments [9]

Surface Treatment	Cost	Quality
None Solvent Degrease Vapour Degrease Mechanical Abrasion Plasma Chemical Etch Anodising	Low ↓ Expensive	Poor ↓ Excellent

Surface Preparation Continued…..

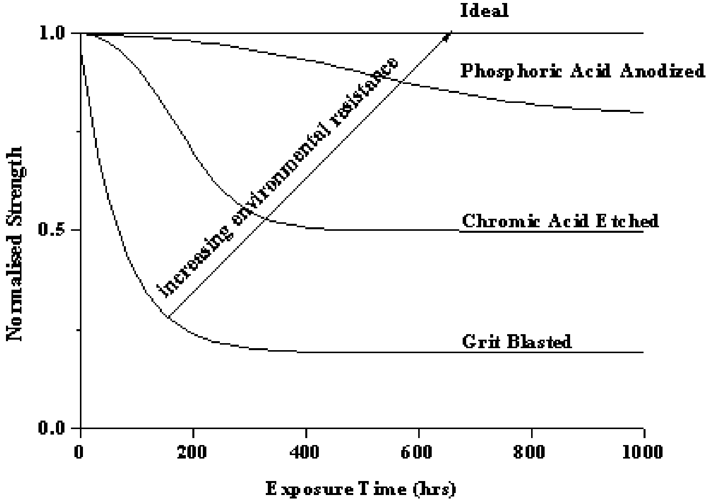
Manufacture and Assembly of Bonded Joints

Surface Preparation of Adherends Continued

The maximum allowable time between surface preparation and bonding or priming metal and composite substrates is dependent on the substrate and the surface treatment (see Table 2). The results presented in Table 2 refer to unconditioned material that has been tested within a short period (e.g. 1-2 weeks) of joint fabrication. The variations in joint strength quoted in Table 2 are optimal values that can be expected under controlled conditions. The uncertainty (or variation) in joint strength can be expected in many cases to increase following exposure to a hostile environment and/or through poor workmanship. Simple surface treatments (e.g. grit blasting or vapour degreasing) are less prone to human error and therefore the variation in joint strength is unlikely to exceed ± 20%. Bonded joints with chemical surface treatments are more prone to large variations in joint strength, particularly as there are often numerous controlling variables, which need to be strictly controlled. However, environmental durability is far better than mechanical surface treatments (see figure right).

Table 2: Maximum Exposure Time Between Surface Preparation or Priming Metal Substrates, and Associated Variation in Tensile Shear Strength

Surface Treatment	Max Exposure Time	Strength Variation (%)
None	1-2 hrs	± 20
Solvent Degrease	1-2 hrs	± 20
Vapour Degrease	1-2 hrs	± 20
Dry Grit-Blasting (Steel)	4 hrs	± 20
Wet Grit-Blasting (Steel)	8 hrs	± 20
Wet Grit-Blasting (Aluminium)	72 hrs	± 20
Chromic Acid Etch (Aluminium)	6 days	± 10
Sulphuric Acid Etch (Stainless Steel)	30 days	± 10
Anodising (Aluminium)	30 days	± 10
Dry Grit + Organosilanes (Aluminium)	2-5 hrs	± 5



Primers such as aminopropyltriethoxysilane (or γ-APS) are often applied to: (i) protect the substrate surface prior to bonding; (ii) increase surface wettability; and (iii) and inhibit corrosion [9]. Primers also act as a coupling agent, forming chemical bonds with the adherend and adhesive, thus improving joint strength and environmental durability. Silane coupling agents (e.g. γ-APS) [36] are known to improve durability in the presence of moisture by increasing the water resistance of the oxide layer on the adherend surface. Joint strengths can be very low, however in cases where the silane treatment is poorly controlled. Incorrect application can result in interfacial failure or cohesive failure within the silane coating. Care is needed to ensure a uniform coating, a monomolecular layer thick, is produced across the entire surface area of the adherends to be bonded.

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[Joint Assembly>>>](#)

Manufacture and Assembly of Bonded Joints

Joint Assembly

This section is concerned with issues relating to joint assembly prior to curing the adhesive (i.e. control of bond-line thickness and adhesive fillet and removal of adhesive spew).

Bonding Fixture

A bonding fixture is recommended to ensure correct bond length, accurate alignment and uniform bondline thickness. Checks should always be made to ensure that there is no mechanical damage due to machining or handling (i.e. adherend bending) or that excessive adhesive is forced from the joint due to clamping forces applied to the bonded joint. It may be necessary to check the clamping force applied by the fixture to the joint during the curing process to ensure that clamping force remains constant and has not relaxed through adhesive flow. Mould release agent or thin polytetrafluorene (PTFE) film will need to be used to guarantee easy release of bonded components from the clamping fixture.

Care needs to taken to ensure good alignment during specimen preparation (i.e. bonding of adherends). Small misalignment can detrimentally affect joint strength and fatigue performance.

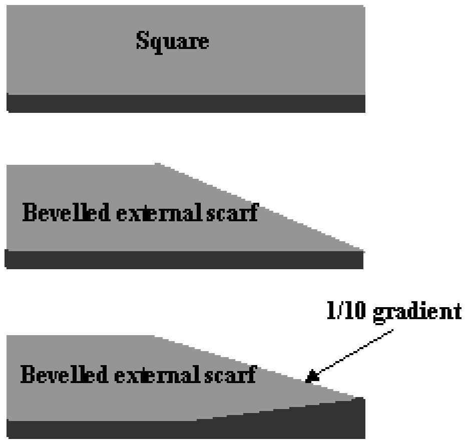
Joint Assembly Continued…..

Manufacture and Assembly of Bonded Joints

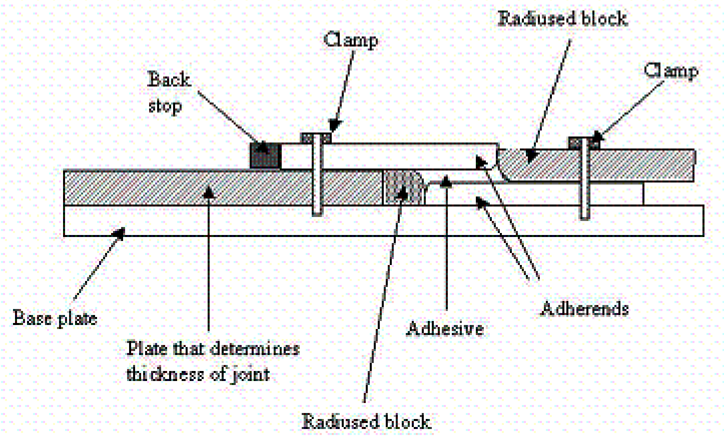
Joint Assembly Continued

Adhesive Fillet

The use of tapered or bevelled external scarf and radius fillets at the bond-line ends will reduce peel and shear stresses induced by eccentricity in the loading path (see figure below left). These additional features will add considerably to the costs of specimen manufacture. The use of absolutely rigid adherends will not eliminate stress concentrations at the bond-line. Significant increases in shear strength of lap joints, compared with square-ended bond-lines, can be achieved through the formation of a fillet or spew at the overlap ends. Ideally the gradient should be 1/10. Further increases in strength may be achieved by rounding the ends of the adherends (see figure below right). The spew also acts as a barrier to water and chemical ingress from the surrounding environment.



Adhesive fillets



Fixture for controlling single-lap joint fillet

Joint Assembly Continued…..

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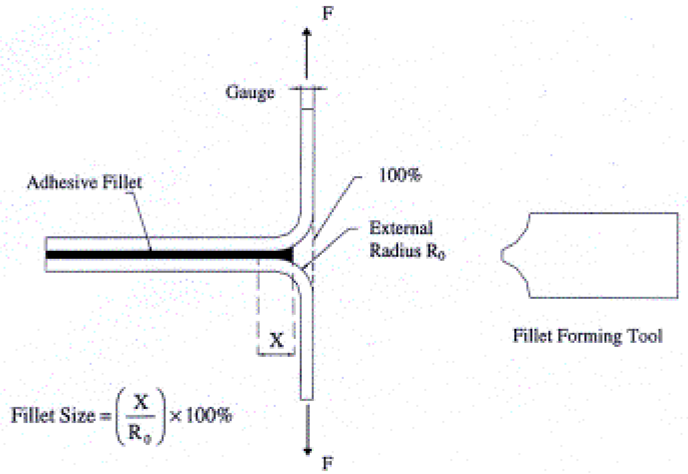
Manufacture Home | Manufacture of Adherends | Surface Preparation | Joint Assembly | Curing Adhesive

Manufacture and Assembly of Bonded Joints

Adhesive Fillet Continued

In the case of T-joints, the size of external fillets has minimal affect on joint stiffness and strength. In contrast, the size of the internal fillets between the central web and adjacent flanges, and between the flanges and base plate can have a major affect on joint properties; particularly fatigue performance. A small increase in the size of the internal fillet can result in a ten-fold increase in fatigue life.



Definition of fillet size in T-peel joint (including fillet forming tool)

A number of points are worth noting:

- Fillet size and shape should be controlled throughout the bonding process. This can be achieved using either a specially designed bonding fixture as shown in the figure right or a special tool shaped to fit within the bonded joint. The tool can be held in place using heat resistant tape. The figure below shows a tool that was used to produce a consistent fillet for T-peel joints bonded with a paste adhesive. The tool can be fabricated from either aluminium or stainless steel coated with release agent.
- Controlling fillet or spew geometry is not always possible as a number of adhesives undergo either minimal flow during cure (e.g. flexible adhesives) or excessive flow. The high viscosity associated with flexible adhesives prevents adhesive flow, thus making it difficult to control the fillet geometry. Low viscosity adhesives may require a dam to be constructed to restrict adhesive flow.
- Care needs to be taken to ensure no adhesive is removed from inside the bond area when removing excess adhesive from the joint prior to cure. Removing adhesive from inside the joint will result in localised debonding and poor joint performance.
- Avoid removal of adhesive spew from the ends of joints after cure, as there is the possibility of damaging the joint. It may be convenient to remove spew from the specimen sides to provide a straight edge for aligning in a test machine. This can be achieved using emery paper.

[Joint Assembly Continued…..](#)

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Manufacture and Assembly of Bonded Joints

Joint Assembly Continued

Bond-line Thickness

Bond-line thickness needs to be accurately controlled (i.e. uniform adhesive layer thickness across the entire bonded area) in order to obtain consistent and reliable joint strength. Also, the method used to control bond-line thickness must not introduce voids or promote void formation in the adhesive otherwise the joint performance will be compromised. It should be noted that the thicker the bond-line the higher the risk of incorporating a high level of voids.

In addition, stresses at the corners of the joint tend to be larger as it is difficult to maintain axial loading with a very thick bond-line. Thick adhesive layers can change the cure properties producing internal stresses, thereby reducing short and long-term performance. Conversely too thin a bond-line can result in adhesive starvation and debonding. Optimum bond thickness will depend on the type of adhesive used.

Control of bond-line thickness can be achieved by mechanical means (i.e. separation of adherends physically controlled by the bonding fixture), through the use of thin wire spacers (e.g. stainless steel) inserted between the adherends or by ballontini glass balls, which can be mixed with single- and two-part adhesive pastes (typically 1% by mass). Film adhesives are available with carriers (e.g. nylon mat or mesh), which control bond-line thickness. It is essential that wire spacers used to control bond-line thickness are located well within the bonded area away from the specimen edges and regions of high stress concentrations (i.e. ends of joints).

When using glass beads to control bond-line thickness, the distribution of glass beads in the adhesive must be uniform and therefore glass beads should be thoroughly mixed into the adhesive. Mixing should take place before applying the adhesive to the adherend surfaces. Controlling the bond-line thickness of flexible adhesives joints is difficult due to the highly viscous nature of flexible adhesives. The preferred method is to use thin wire spacers, however work carried out at NPL showed that this method of bond-line control was not always reliable.

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Curing Adhesive>>

Manufacture and Assembly of Bonded Joints

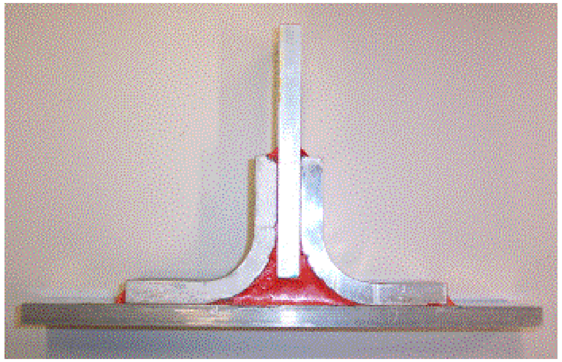
Curing Adhesive

There are a number of key points that should be considered when curing adhesive joint specimens [18, 21, 40-44]:

- Porosity, in the form of entrapped air and volatiles, is a common cause of premature failure. In many cases it is virtually impossible to produce void free specimens, particularly for materials with a high viscosity. Specimens should be prepared using methods that minimise the inclusion of air in the test specimens. Visual inspection should be carried out to ensure there is no air entrapment.
- The cure state of the adhesive layer in the adhesive joint should be similar to that of bulk adhesive specimens [40-41]. Failure to achieve similar thermal histories can result in significant differences in material properties. Differences between thermal histories will lead to differences in mechanical properties.
- Temperatures in the adhesive should be monitored throughout the cure cycle. It is recommended that trials be carried out on the adhesive joint using a thermocouple embedded in the adhesive in order to ensure that the temperature within the adhesive layer actually reaches the specified cure temperature.
- Due to differences in thermal mass, joint specimens may heat at different rates than bulk test specimens and therefore the final temperature of the adhesive joint at the end of the cure period can be significantly different to that of the bulk adhesive.
- For heat curing systems, the temperature of the specimen will lag behind the oven temperature, and it may therefore be necessary to elevate the oven temperature when curing joint specimens.
- Adhesives should be fully cured prior to conditioning and testing otherwise an adhesive will continue to cure, thus invalidating the test data.
- Handling adhesives can be hazardous to human health, thus COSHH procedures should be followed to minimise operator exposure. Ovens and work areas should be suitably ventilated, ensuring minimal levels of hazardous vapours/gases in the work area.

Curing Adhesive Continued…..

Manufacture and Assembly of Bonded Joints

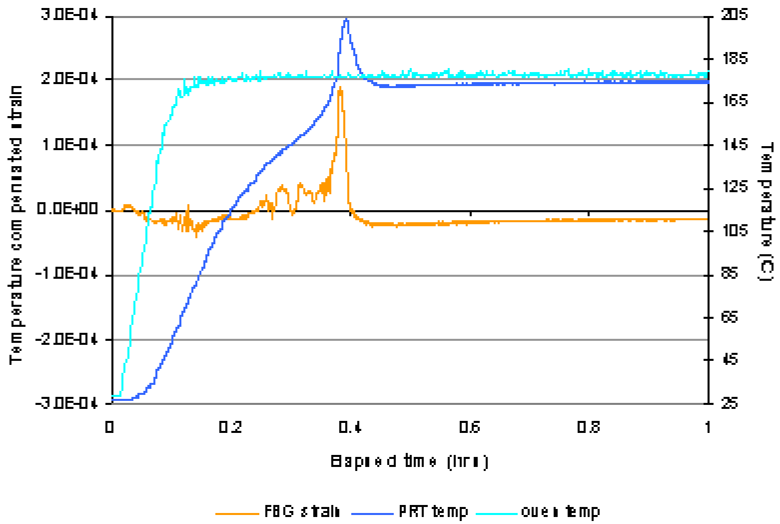


Fully filled aluminium T-joint

Curing Adhesive Continued

- Adhesives have a low thermal conductivity. This may prevent dissipation of heat generated by exothermic cure reactions, thus causing heat damage (i.e. charring of the adhesive). This is a particular problem when large volumes of adhesive are used to fill internal cavities in joints (see figure below left). The figure below right shows an exothermic response of a high temperature curing single-part epoxy adhesive in which the cure temperature is exceeded by 40°C.
- Residual thermal stresses may be generated as a result of non-uniform (rapid) cooling, resin shrinkage and thermal expansion coefficient mismatch between the adhesive and adherend. As the joint is cooled down from the cure temperature, residual stresses are frozen in the material.

It is recommended that the quality documentation should include details on the cure variables (i.e. temperature, pressure, heating and cooling rates and dwell times), and a record of equipment used for curing the adhesive joints and monitoring the temperature within the oven and adhesive joint (i.e. oven type and thermocouples). Real-time monitoring of material property development in adhesives can be achieved using oscillatory rheometry, Fibre Bragg Grating (FBG) or ultrasonic methods. Thermal analytical techniques, such as differential scanning calorimetry (DSC) and dynamic mechanical thermal analysis (DMTA), can provide useful information relating to adhesive composition and final state of cure [18,19,22, 44-46].



Exothermic response of a high temperature curing epoxy adhesive

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Mechanical Testing >>>

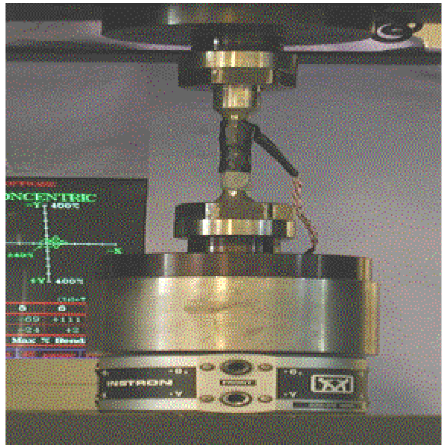
Mechanical Testing of Bonded and Bolted Joints

This section considers the affect of test parameters (i.e. test machine alignment, load train stiffness, methods of gripping test machines, accuracy of load and displacement transducer) on the accuracy and reliability of strength and long-term performance of bonded and bolted joints. Guidance is provided on the main factors that need to be controlled when carrying out mechanical testing. Consideration is given to various loading modes (i.e. static, cyclic fatigue and creep) and environmental conditions (i.e. elevated humidity and temperature). Appendix 1 provides a summary of commonly used adhesive joint test methods and related standards.

Mechanical Testing of Bonded and Bolted Joints

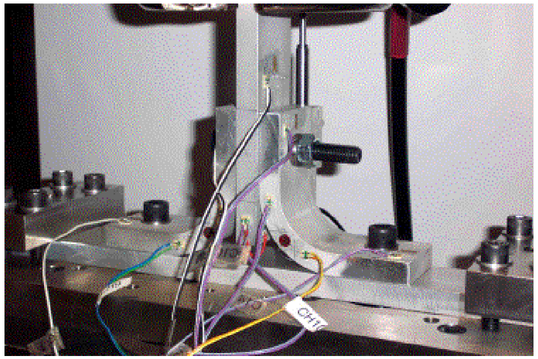
Test Machine and Specimen Alignment

The test machine should have high lateral rigidity and accurate alignment between the upper and lower gripping faces. The load train should be as short and as stiff as possible (i.e. no universal joints included). If the grips are articulated, as in the case of universal joints, then the specimen may be subjected to large bending and twisting loads, resulting in reduced joint strength. Avoid eccentric acting forces. Small lateral (1 to 2 mm) or angular (1 to 2 degrees) offsets in the loading train can lead to additional shear and bending stresses, resulting in premature joint failure. It is worth noting that the slope of the load-displacement response can be similar for poor and well-aligned specimens.



Alignment specimen with strain gauges

It is recommended that the alignment of the test machine and the test specimen be checked at the centre of the gauge length using a strain gauged coupon specimen [47-48]. Alignment specimens can be in the form of a rectangular or circular bar (see figure above left). These specimens need to be accurately machined to ensure errors in parallelism are < 0.2 mm/m and in concentricity (lateral offset) of 0.03 mm [47]. Strain gauges are bonded to the surface of the alignment specimen in order to monitor alignment and bending strains. Bending strains should be less than 3 to 5% of the average axial strain.



Bolted aluminium T-joint with strain gauges

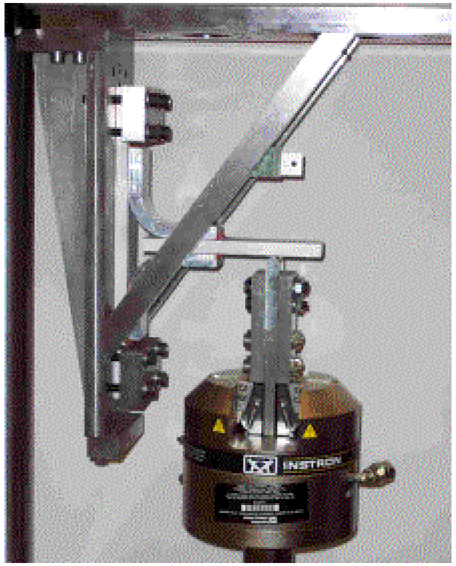
Use a device to ensure that the specimens are positioned in the grips in a repeatable manner. An alignment fixture can also be included in the loading train to minimise angular and lateral offset between the upper and lower machine grips or loading rods. The alignment cell is attached to the upper or lower crosshead of the test frame; whichever is the most convenient. Commercial alignment cells are available that allow lateral movement, tilt and rotation of the machine grip or loading rod (see figure top right). Strain gauges attached to the test piece can also be used to check alignment (see figure left).

Gripping Specimens>>

Mechanical Testing of Bonded and Bolted Joints

Gripping of Specimens

Grips for holding test specimens to be loaded in tension should be attached to the test frame so that the major axis of the test specimen coincides with the direction of pull through the centreline of the gripping assembly. The centre line of the specimen should be aligned with the axis of the loading fixtures to avoid bending and asymmetric loading. It is important than when loading test specimens in the grips that no lateral or angular offset is introduced to the specimen. The figure right shows a rigid frame for applying lateral loads to a bonded T-joint. The diagonal struts provide additional rigidity, thus preventing side-ways movement of the test fixture.



Side loading fixture with bonded aluminium T-joint

Avoid rotating the grips during gripping operation. If one of the grips is articulated, this should be tightened first to prevent the specimen being subjected to large bending and twisting loads during tightening. Care should be taken to avoid axially stressing the specimen whilst the grips are being tightened. Any pre-stressing of the specimen should be kept to a minimum. Grips should be slowly tightened with any induced loads removed by progressively adjusting the crosshead position. The applied load on the specimen should be zero at the onset of testing. It may be necessary to use a device (i.e. metal spacer) during the test set-up to ensure good alignment and repeatable test results, as often the specimen width is less than the width of the mechanical grips.

Manual or servo-hydraulic grips can be used to hold specimens during testing. Wedge-action grips are recommended as the lateral force (i.e. pressure) applied to the test specimen in the gripping regions increases as the axial load applied to the specimen increases. Servo-hydraulic grips provide uniform pressure in a controllable manner. Gripping pressure should be sufficient to prevent specimen slippage throughout the duration of the test, but not excessive to initiate failure of the specimen at the grips. For cyclic loading, it is essential that fretting in the gripped region be prevented to avoid the possibility of premature failure.

Strain Measurements>>

Mechanical Testing of Bonded and Bolted Joints

Contact Extensometer | Non-contact Extensometer | Strain Gauges | Crosshead Displacement | ESPI

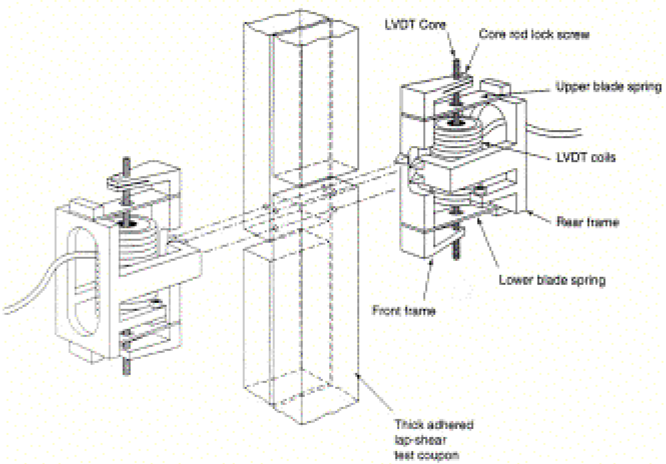
Strain and Displacement Measurement Techniques

A number of contact and non-contact techniques are available for measuring strain and displacement. This section considers the use of contact extensometers, linear voltage displacement transducers, video extensometers, electronic speckle pattern interferometry (ESPI), strain gauges and crosshead movement for measuring strain and displacement under ambient and hostile environments, and static and cyclic fatigue loading conditions [49, 22]. With the exception of strain mapping techniques (i.e. electronic speckle pattern interferometry (ESPI) and digital image correlation), the measured strain will be an average strain in the bond-line. Also, strain gauges only measure strain at the location of the gauge.

Contact Extensometers

Contact extensometers are the preferred method for measuring strain and displacement, and hence stiffness of bonded and bolted joints. It is recommended that two extensometers, attached to opposite faces of the specimen, be used to measure displacement [18]. Any bending of the specimen will be apparent from diverging displacement readings. It is recommended that the individual transducer readings be recorded so that the quality of the test data can be checked. Errors due to minor bending are minimised by taking the average measurement of the two displacement transducers. To minimise inclusion of adherend deflection in the measurement the contact points should be as close to the bond layer as possible.

The figure right shows two extensometers attached to a thick adherend shear test (TAST) specimen for measuring shear deformation. The three-point contact minimises rotation of the extensometers. Knife-edged tensile extensometers, as described in reference [18], can be used provided that extensometer straddles the bondline. The deformation of the adherends needs to be accounted for when analysing the data, but where the stiffness of the adherends is very much greater than that of the adhesive layer then corrections may be minimal.



TAST specimen with extensometers for measuring shear deformation

Measurement Techniques Continued…..

Mechanical Testing of Bonded and Bolted Joints

Contact Extensometer | Non-contact Extensometer | Strain Gauges | Crosshead Displacement | ESPI

Contact Extensometers Continued

Where adherends are flexible, it is advisable to support the weight of the extensometer because allowing the extensometer to hang unsupported from the specimen may cause bending and introduce contact stresses. The contact forces should be sufficient to prevent slippage between the extensometer and the specimen, but not large enough to cut or nick the specimen surface causing the specimen to fail prematurely. It may be necessary to remove extensometers attached to a specimen prior to failure in order to prevent the possibility of the extensometer sustaining damage during failure. Failure can be a violent event, releasing considerable energy, thereby damaging or even destroying the extensometer.

An extensometer should be capable of measuring the change in gauge length with an accuracy of 1% of the applied displacement or better (i.e. equivalent to ± 0.5 mm for 10 % strain over a typical bond thickness of 0.5 mm). It is important that the extensometers are able to operate satisfactorily within the test environment (i.e. temperature and humidity), and that these devices are resistant to chemical attack when used in hostile environments. Precautions may need to be taken to insulate the leads to prevent moisture ingress.

Non-Contact Extensometers

Non-contact or optical extensometers (e.g. video extensometers) are available, which avoid contact damage and can be used up to failure, since there is no possibility of damage to the extensometer. Video extensometers are not particularly suited to measuring small strains (e.g. movements of a few μm), which limits their applicability to structural adhesive joints. Furthermore, measurement is normally only possible at one side of the joint so that bending cannot be evaluated. However, video extensometers have been used in tests on joints bonded with flexible adhesives where deflections are larger. Measurements of joint stress-strain curves have been in reasonably good agreement with contact extensometer results. Some modern systems provide capabilities for dot location measurements, which allows a limited strain mapping capability.

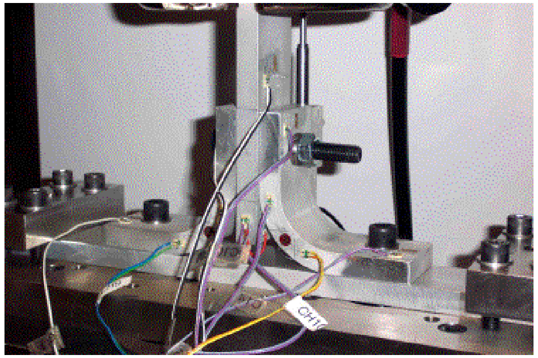
The technique relies on a remote camera monitoring the separation of two marks or lines inscribed on the test specimen, which define the gauge length. The change in separation of the two lines is recorded throughout the test. The gauge marks should be approximately equidistant from the mid-point, and the measured distance between the marks should be measured to an accuracy of 1%, or better. Gauge marks should not be scratched, punched or impressed on the specimen in any way that may cause damage to the specimen. It is advisable to ensure that there is a sharp contrast in colour between the specimen surface and the gauge marks. The lines should be as narrow as possible. There are no temperature restrictions as video extensometers can be located outside the test chamber.

Measurement Techniques Continued…..

Mechanical Testing of Bonded and Bolted Joints

Strain Gauges

Currently there are no standard tests using strain gauges to monitor strain in an adhesive layer. However, structural monitoring capabilities, where strain-sensing devices (e.g. Fibre Bragg Grating) are embedded in materials, are the subject of research in many organisations. Strain gauges can be attached to adherends and will measure the strain in these adherends. The usefulness of such measurements may be limited except in the cases where changes in joint performance are manifested in measurable changes in the adherend strain. One such application is back-face strain gauging of thin lap-shear joints where crack growth in the adhesive layer can be monitored through strains measured by gauges bonded to the external surface of the adherends at the overlap. Strain gauges are occasionally used for monitoring strain in bonded and bolted structures (see figure right) and have proved useful for determining the onset of localised damage, such as cracking in composites and gross yielding in aluminium [17].



Bolted aluminium T-joint with strain gauges

Strain gauges are generally limited to the measurement of strains less than 10%. Biaxial rosettes are available for measuring longitudinal and lateral strains. Large strain gauges are preferable as alignment and handling is easier, and they average out local strain variations. Local strain variations can cause premature failure of the strain gauges. Correct alignment of strain gauges is important, as significant errors can be caused by careless application of the strain gauges to the specimen. Errors of 15% can occur from a 2° misalignment [49].

The adhesive used to bond the strain gauge should be capable of withstanding the test environment for the complete duration of the test [22]. Most adhesives are sensitive to moisture (and other chemicals), which can often preclude bonding prior to specimen conditioning. Moisture attack of an adhesive and strain gauges will occur from the top, edges and in the case of polymeric materials through the substrate beneath the gauge. The situation is exacerbated at elevated temperatures. It is therefore important to ensure that the adhesive selected for bonding the strain gauge and associated electrical wiring is suitably encapsulated.

Strain Gauges Continued…..

Mechanical Testing of Bonded and Bolted Joints

Strain Gauges Continued

The strain gauges are usually bonded to the specimen following moisture conditioning (i.e. immersion in water or exposure to humid environments). However, bonding the strain gauge to the specimen may require heat and pressure, which will induce drying out of the conditioned specimen. To avoid drying out, room temperature curing anaerobic adhesives have been used and have proved satisfactory for bonding strain gauges to moisture conditioned specimens. For hot/wet conditions, a high temperature anaerobic adhesive can be used provided the application temperature does not result in thermal damage to the adhesive joint (i.e. adhesive and adherend). Although anaerobic adhesives have good moisture, solvent and temperature degradation resistance, these adhesives are known to attack certain plastics. Hence, precautions need to be taken when selecting these materials for use with plastics or fibre-reinforced polymer composites. Cyanoacrylates (or super glues), which are sensitive to surface moisture and low pH levels, are unsuitable for environmental testing. Strain gauge manufacturers can provide information on adhesive selection and procedures for protecting strain gauges.

For cyclic loading, it is essential that the fatigue life of the strain gauges, over the operating strain levels, should be well in excess of the life expectancy of the test component. Autogenous (self-generated) heating can degrade the mechanical properties of the adhesive bond between strain gauges and the specimen. This can result in small errors in strain measurement, thus requiring correction of the data to account for the temperature rise. Measurements should also be carried out to determine the magnitude of creep within the strain gauge adhesive.

Recent developments have seen the use of instrumented bolts in which strain gauges oriented along the axis and at ±45° to the axis of the bolt are bonded to the inside of the bolt. A small diameter hole is drilled in the bolt to accommodate the strain gauges. A precision jig is required to ensure that the hole is accurately located at the centre of the bolt. This technique is considerably more expensive than using strain gauges bonded directly to the adherends, however the instrumented bolt technique is well suited to those applications where strain gauges cannot be used (e.g. single-lap joints). As a research tool, instrumented bolts offer designers/engineers a means of accurately monitoring multi-bolt array joints, but reservations must be expressed as to the strength and fatigue life of modified bolts particularly under service conditions.

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Measurement Techniques Continued…..

Mechanical Testing of Bonded and Bolted Joints

Strain and Displacement Measurement Techniques

Crosshead Displacement

An approximate measurement of strain and hence stiffness can be obtained from measuring the crosshead displacement of the test frame [18]. The strain is the ratio of crosshead displacement and the initial grip separation. Hence, any slippage within the loading train will produce errors in the strain measurement. The strain values obtained from crosshead measurements will differ from the actual strain in the central region of the specimen.

Stiffness measurements directly obtained from the crosshead movement need to be corrected to take into account the stiffness of the loading train. This can be a difficult task as the specimen size and geometry, and the deformation behaviour of the specimen need to be taken into account [14]. Given the small adhesive layer deflections that occur even at large strains owing to the thin bondlines, the accuracy of strains determined using crosshead displacements must be considered suspect and used only for qualitative purposes.

Linear voltage displacement transducers (LVDTs) are recommended in preference to monitoring crosshead movement. LVDTs provide a direct reading of the moving part and can be attached at any point on the structure as required. These devices tend to be used to monitor global rather than localised deformation. Accurate alignment is essential otherwise measurement errors will occur and the movement of the device can be restricted

Measurement Techniques Continued…..

Mechanical Testing of Bonded and Bolted Joints

Strain and Displacement Measurement Techniques

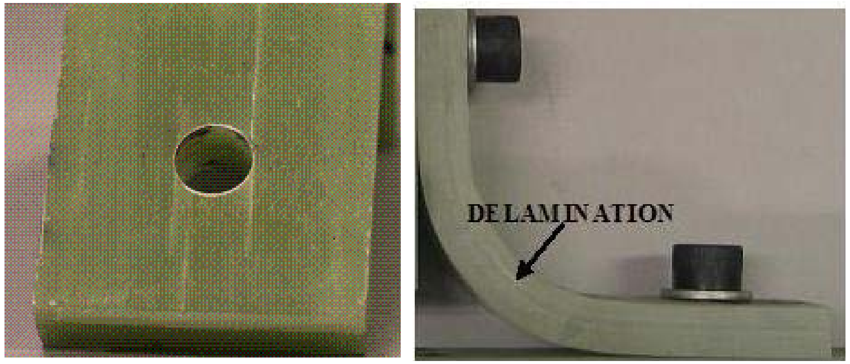
Electronic Speckle Pattern Interferometry (ESPI)

Electronic speckle pattern interferometry (ESPI) is a non-contact technique capable of measuring and monitoring non-uniform strain fields at high resolution. The system can measure the deformation and thus the strain under mechanical and/or thermal loads along the three material axes (i.e. 3-D strain measurement). ESPI systems are capable of measuring local deformation with a resolution of 0.1 mm, equivalent to 200 microstrain for a 0.5 mm thick bond. The technique needs minimal specimen preparation and is capable of inspecting areas ranging from 25 mm2 to 600 mm2, but capital outlay for equipment is generally prohibitive for most test facilities. The technique can be used to measure strain distributions in complex geometries, and for checking finite element analysis. Further details of the technique with illustrated case studies are given in reference [50]. Interferometry techniques are not routine and are thus unlikely to be suitable for mass screening programmes. Similarly the technique may not be suitable for cyclic testing.

Mechanical Testing>>

Mechanical Testing of Bonded and Bolted Joints

Mechanical Testing



Typical damage induced in a bolted GRP T-joint. (left: longitudinal splitting around bolt hole; right: delamination in flange)

The strength and stiffness measured in mechanical tests form only part of the useful data that can be obtained. The modes of failure (i.e. cohesive failure in the adhesive or delamination failure in a composite) and in the case of bonded joints the degree of cohesive and adhesive failure should be recorded. Optical microscopy or scanning electron microscopy (SEM) may be required to analyse the fracture morphology, particularly if failure is close to the adhesive-adherend interface. The figure below shows a flange from a bolted T-joint fabricated from glass fibre-reinforced epoxy that has undergone delamination. The SEM investigation revealed that failure originated in the vicinity of the bolt as a result of combined shear and through-thickness tensile stresses.

The effects of test environment need to be considered. The operator should ensure that the equipment used to load and monitor (i.e. extensometers and load cell) the specimen are unaffected by the test environment. It may be necessary to thermally insulate load cells and use molybdenum grease to ensure moving parts in test fixtures do not seize whilst testing. It is recommended that the loading fixtures be fabricated from stainless steel to avoid environmental attack.

Number of Test Specimens

Ideally, a minimum number of five specimens should be tested for each batch of specimens. If a greater precision of the mean value is required then the number of specimens tested should be increased (see ISO 2602 [51]).

Mech Testing Continued…..

Mech Testing Home | Alignment | Gripping Specimens | Strain & Displacement Measurement | Mechanical Testing | Cyclic Fatigue | Creep

Mechanical Testing of Bonded and Bolted Joints

Specimen Dimensions

Joint dimensions, bolt hole diameters and spacing need to be accurately measured, as small measurement errors can translate into large variations in strength or stiffness, particularly if the calculation includes squares or cubes terms of the measured parameter (see table below). The uncertainty in strength or fracture toughness calculation is compounded where there is more than one term (i.e. width, thickness and crack length, etc), each with an associated uncertainty. Bondline thickness, being a very small dimension, tends to be the dimension where accuracy and precision of dimensional measurement are most critical. Measurements at different locations should be carried out to check the uniformity of bondline thickness.

Vernier callipers or travelling microscope are recommended for measuring specimen width and bond length, and a micrometer or travelling microscope for measuring specimen thickness. A travelling microscope should be used to measure crack length.

Associated Uncertainty with Measurement Error			
Dimensional Error (%)	Linear Error (%)	Squared Error (%)	Cubed Error (%)
± 1 ± 5 ± 10	± 1 ± 5 ± 10	± 2 ± 10 ± 21	± 3 ± 16 ± 33

Speed of Testing

Polymeric adhesives and composites are visco-elastic materials; that is their mechanical properties (strength and stiffness) are sensitive to the rate at which they are loaded (or more accurately the strain rate). Standards relating to testing of bonded or bolted joints infrequently specify the speed or rate of testing required, but instead specify that the test joint be loaded at a constant stress or strain rate and to ensure that failure is achieved in a prescribed period of time (typically 60 to 90 seconds). This introduces a degree of subjectivity into the selection of test conditions. Where adhesive joints have different bondline thickness then strain rates may vary within and between specimens leading to greater uncertainties in the results. For comparative measurements, it is recommended that all joints be tested at comparable strain rates. This can be achieved by ensuring the ratio of test speed over bond thickness is approximately the same for each test specimen. The standard requirement to fail the joint in the prescribed time is convenient for testing but may not impose strain rates relevant to the design requirement.

A series of trials to failure are recommended in order to ascertain the test speed required to meet the strain rate or time limit specified in the standard. It is therefore advisable that additional specimens be prepared for this purpose. It should be noted that the small gauge length, due to the thin bond line, leads to relatively high rates of strain in the adhesive at moderate test speeds. This needs to be considered when comparing joint specimen tests with bulk specimen properties.

Mech Testing Continued…..

Mechanical Testing of Bonded and Bolted Joints

Adherend Property and Geometric Effects

Altering the geometry of a bonded joint will invariably cause changes to occur in the stress and strain distribution within the adhesive layer. These differences can have a profound effect on the stress concentrations and consequently the load-capacity and long-term performance of the joint. Currently, there are no well-established design procedures for predicting failure behaviour or relating changes in material and geometric parameters to joint strength. Finite element analysis enables the prediction of the effects of changing joint geometry parameters on stress/strain levels in the structure. Thus joints can be designed to minimise stress concentrations. With accurate material properties data, relevant materials models and reliable failure criteria the strength of any joint under any stress state could be predicted. However, the state of the art is not at this stage yet. Research on this is continuing.

The results presented in the table right clearly indicate that the strength of bonded joints is dependent on the adherend material, adherend thickness, bondline thickness and bond length. The table compares the failure load per unit width (N/mm) for various single-lap joint configurations and materials bonded with an epoxy adhesive. The structural properties of bolted joints are also sensitive to geometric parameters in addition to bolt parameters (i.e. location, spacing hole diameter and clamping force).

Adherend Thickness/Overlap Length	Load/Width (N/mm)
CR1 Mild Rolled Steel	
1.5 mm/12.5 mm	334 ± 11
2.5 mm/12.5 mm	354 ± 10
2.5 mm/25.0 mm	428 ± 38
2.5 mm/50.0 mm	633 ± 63
5251 Aluminium Alloy	
1.6 mm/12.5 mm	191 ± 14
3.0 mm/12.5 mm	325 ± 28
6Al-4V Titanium Alloy	
2.0 mm/12.5 mm	457 ± 52
Unidirectional T300/924 Carbon/Epoxy	
2.0 mm/12.5 mm	369 ± 41
Plain Woven Fabric (Tufnol 10G/40)	
2.5 mm/12.5 mm	275 ± 28
2.5 mm/25.0 mm	454 ± 27
2.5 mm/50.0 mm	511 ± 32
5.1 mm/12.5 mm	327 ± 27

For the purpose of the measurement of the adhesive properties, steel adherends are recommended because of the materials high stiffness. For ambient tests, suitable steels are XC18 and E24 grade 1 or 2. However, corrosion-resisting steel (e.g. A167, Type 302) or titanium alloy (e.g. Ti-6Al-4V) are preferable for environmental testing.

Cyclic Fatigue>>

Mechanical Testing of Bonded and Bolted Joints

Cyclic Fatigue

The fatigue properties of a bonded joint are a function of the joint geometry and adhesives, and therefore cannot be determined from the intrinsic properties of the adhesive. For joint characterisation purposes it is recommended that specimens are mechanically loaded at each of five stress levels (i.e. 80%, 70%, 55%, 40% and 25% of the short-term strength of the joint). Fatigue data are normally obtained at the highest frequency possible in order to minimise the duration of tests. The uncertainty in life expectancy at any stress level is typically an order of magnitude.

Restrictions on test frequency can arise from test equipment limitations (response time), time dependent processes and hysteretic (self-generated) heating. Hysteretic heating, which increases with increasing load and frequency, can result in thermal softening of the adhesive, adversely affecting the fatigue performance of composite joint. Reliable data can be obtained at high frequencies provided the stress levels are low. Test frequencies of the order of 10 to 30 Hz can result in substantial heating, particularly in the grip regions. The upper frequency limit will be dependent upon the thermal conductivity of the adherend/adhesive system, mode of loading and specimen size. Trials may be necessary to determine the upper frequency limit.

It is recommended that the temperature rise of the material surface be kept to a minimum. It may be necessary to stop testing to allow the specimen to cool. Alternatively, the test could be carried out in an environmental cabinet with a thermocouple attached to the specimen surface for monitoring and controlling the temperature of the test specimen, although the cooling rate may be too slow to be practical. Thermal imaging equipment can be used to monitor surface temperature, although the latter is beyond the budget of most industrial facilities. The temperature resolution is ~1 °C for the two methods.

Creep>>

Mechanical Testing of Bonded and Bolted Joints

Creep

Creep tests are performed to assess the extension of joints under load to predict long-term behaviour or to assess the long-term strength of joints under load. The first requirement needs high precision extensometry to monitor joint extension and the tests must be performed under stable environmental conditions (temperature and humidity) to avoid artefacts in the measurement. These tests could, in theory, be performed using any of the loading options outlined below, although the highest accuracy is achieved using either option (1) or (2).

- Servo-hydraulic test machines;
- Dead-weight and lever creep testing machines;
- A screw jack in series with a load cell;
- Self-stressing fixture where specimens are placed in either a tube equipped with a pre-calibrated spring system for loading specimens or a circular ring.

The use of a servo-hydraulic test machine is not always an economic option in most cases. A bank of small creep machines can be assembled at a considerably lower cost compared with the capital outlay involved with purchasing and operating servo-hydraulic units.

The large uncertainty associated with creep test results, implies that the current approach of conducting three tests per stress level is inadequate and that considerably more data points are required for generating reliable creep rupture curves for engineering design purposes. Five specimens per stress level with five stress levels per condition should provide a reasonable number of data points. For joint characterisation purposes it is recommended that specimens are mechanically loaded at each of five stress levels (i.e. 80%, 70%, 55%, 40% and 25% of the short-term strength of the joint).

Two approaches have been adopted for assessing the degree of degradation under combined static load and environment:

- **Rate of strength loss with time (i.e. residual strength):** This approach determines the time taken for the strength of the joint to decline to a design stress limit, below which the joint is no longer considered safe. Specimens are removed at regular intervals to assess strength reduction.
- **Time-to-failure:** This approach attempts to determine the probable average life expectancy of a bonded joint at a prescribed stress level or to determine the percentage of failures that can be expected to occur within a given exposure period.

FE Analysis>>>

Finite Element Analysis

Finite element analysis (FEA) is a computational tool that can be used for calculating forces, deformations, stresses and strains throughout a bonded or bolted structure [22]. FEA can be used to model complex multi-component systems. The structure is represented by a series of nodes that define the corners of a two- or three-dimensional array of elements. The mesh used to represent the geometry of the structure can affect the results predicted. Applying constraints and forces or displacements at the boundary of the structure simulates a loading situation. The resulting forces, displacements, stresses and strains throughout the structure are calculated by solving the equations (constitutive laws) that describe the deformation behaviour of the materials constituting the structure whilst maintaining continuity of displacement at all the nodes. The equations are solved for each incremental increase in the applied force or displacement. As a result of the analysis, the following quantities can be calculated [23]:

- Deflection of any point (node) in the structure as a function of the applied force (load-displacement or force-extension curve)
- Stress and strain components in any element at particular levels of applied force or displacement (often displayed in the form of contour plots).

The load-displacement curves can be compared easily with experimental data. The FEA outputs, such as stress and strain, can be used with failure criteria to predict failure.

Creating Meshes>>

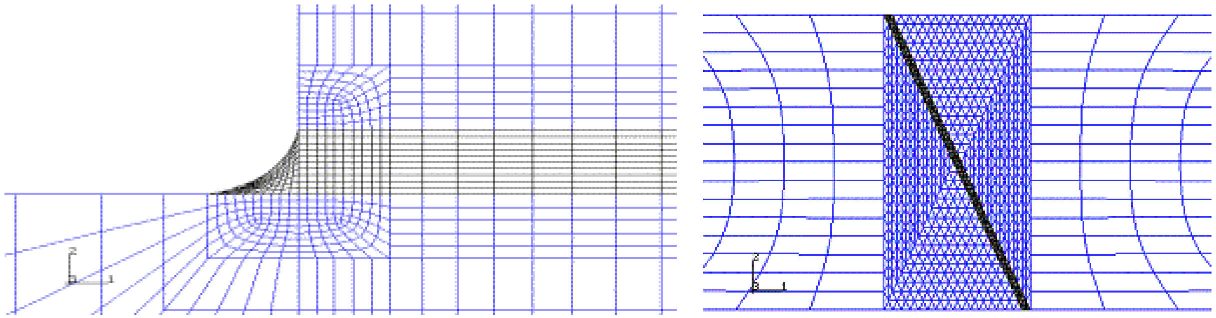
NPL National Physical Laboratory

Finite Element Analysis

Creating Meshes

An element mesh is used to model the required geometry of the structure, see examples below. The geometry needs to be accurately modelled, taking into account any symmetry, which could reduce the size of the model and the number of calculations required. The FE mesh is locally refined in the vicinity of geometric singularities (i.e. sharp corners). This is an area of concern as geometric singularities effect stress-strain predictions. Using elastic analysis, the stresses will go to infinity at the singularity, which is unrealistic. The preferred approach is to model the large stress concentrations using elastic-plastic analysis. The stresses under these conditions will only reach the plastic limit, however predicted strains in the vicinity of the singularity will be unreliable, appearing much higher than expected.

A pre-processor, such as FEMGV [25] is used to generate the geometry and mesh. It is advisable to apply boundary conditions and loads to features in the geometry so that the mesh can be changed without altering the restraints, thus reducing meshing and computational times. The boundary conditions and loads need to accurately represent the real situation. Incorrectly modelling boundary and loading conditions will inevitably result in inaccurate predictions.



Single-lap joint mesh **Scarf joint mesh**

Finite element meshes for bonded joints

There are several factors that need to be considered when meshing a joint geometry, such as element type and mesh density. The choice of elements (e.g. simple beam and solid (continuum) elements) can have a profound affect upon the analysis. Each element type has advantages and disadvantages depending on the application. Solid (continuum) elements, used to model bonded joints, are suitable for linear analysis and also for complex non-linear analyses involving plasticity and large deformations.

Creating Meshes Continued…..

Finite Element Analysis

Creating Meshes Continued

An adhesive joint can be modelled as either a two- or three-dimensional geometry. Two-dimensional continuum elements include plane stress, plane strain and generalised strain elements. These are briefly described below [23].

- Plane stress elements are used when the out-of-plane dimension of a body is small relative to its in-plane dimension (out-of-plane stress assumed zero). Used to model thin flat bodies.
- Plane strain elements are used when the out-of-plane dimension of a body is much larger than its in-plane dimension (out-of-plane strain assumed zero).
- Generalised plane strain element - model is placed between two rigid planes, which can only move closer or further apart. Deformation of the model is assumed to be independent of the axial position so that relative movement of the two planes causes a direct strain in the axial direction only (transverse shear strains are zero).

Three-dimensional continuum elements avoid the artificial distinction between plane stress and plane strain. Although, a converged three-dimensional analysis may provide a more accurate solution to the problem of structural assessment of bonded joints than two-dimensional analysis, the time and effort required for mesh generation and analysis of results is substantially increased. For the bonded joints two-dimensional analysis is generally preferred for comparative studies where a series of finite element models are required. However, the assumptions of plane stress or plane strain will not be valid at all locations within the joint.

The mesh should be refined at regions of high stress gradients. The general approach is to run an initial analysis with a coarse mesh then to progressively increase the mesh density (i.e. increase number (and therefore decrease size) of elements within the mesh). Checks should be carried out to ensure that the boundary conditions and loads are behaving as expected and that the force-extension data obtained is physically reasonable. It is impractical to refine the whole mesh, particularly if the only information required is the force-extension response. A coarse mesh is used in areas where stress and strain values are relatively low and uniform.

Calculation of localised stresses and strains require refinement of the mesh in regions of high stress gradients. A reduction in element size generally leads to higher maximum stress and strain values, although at very small element sizes the effect of further dimension changes has minimal effect on stress and strain predictions. This is known as mesh convergence (i.e. stable maximum value reached). Mesh refinement need only be applied to regions of high strain (or stress) concentrations - highly refined mesh with a high mesh density inevitably leads to longer process times. Once the mesh is refined satisfactorily and the analysis run, the post-processor should be used to check the continuity of the stress contours. It is important to ensure that any discontinuity is kept to a minimum otherwise stress localisation studies are difficult (if not impossible).

Back Creating Meshes Continued…..

Finite Element Analysis

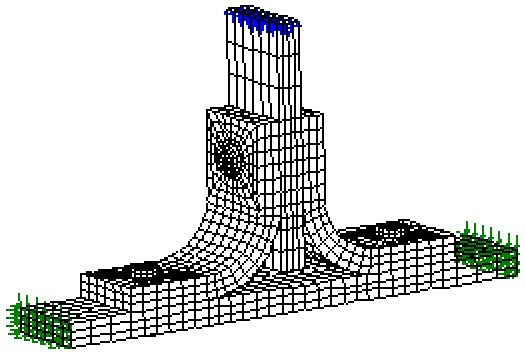
Creating Meshes Continued

Bolted Joints

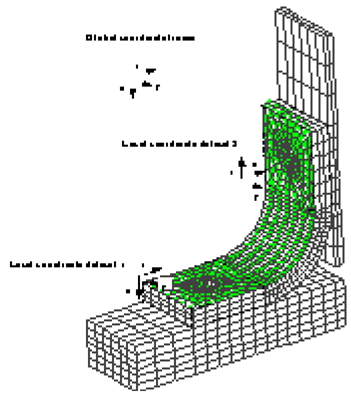
In the case of bolted joints, three-dimensional analysis is essential for modelling the behaviour of the bolted region (see below). Solid continuum elements are used to model both the bolt and the adherend section around the bolt. In order to model the clamping force of torqued bolts, a pre-stress is applied to elements in the shank of the bolt. An important feature of modelling bolted configurations is the analysis of contact between component parts. Parts of the model that are in contact before a load is applied or would come into contact during loading need to be identified. These regions are

- Initial contact between adherend sections,
- Initial contact between the underside of the bolt heads to adherend sections,
- Loss of, or increased contact between adherend sections,
- Potential contact between the bolt shanks and holes.

It should be noted that the introduction of washers will add to the complexity of the analysis, and thus to the meshing and processing time. For simplicity, bolts are generally considered to behave elastically.



Full mesh for bolted T-joint



Local co-ordinate system for bolted T-joint

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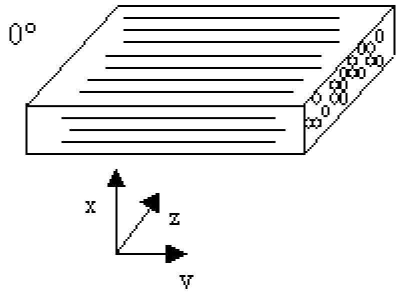
[Creating Meshes Continued…..](#)

Finite Element Analysis

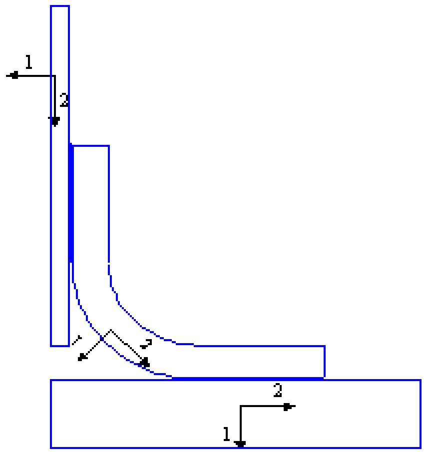
Creating Meshes Continued

Composite Joints

When modelling composite laminates local coordinate datasets need to be carefully defined and assigned to the relevant sections in the model in order to ensure that the ply orientations of the laminate coincide with the correct orientations of the model. For curved sections, such as flanges (where it was not possible to define an appropriate local coordinate dataset), the volume orientation axes are positioned such that they coincide with the respective orientations of the laminate (see below).



Global coordinate system



Local coordinate system

FE Solvers>>

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Finite Element Analysis

Finite Element Solvers

The numerical problem can be solved using either standard (implicit) or explicit mathematical code (see below) [23]:

- Implicit code can be used to solve a wide range of linear and non-linear problems
- Explicit code is suitable for short, transient dynamic events, such as impact, and is also very efficient for highly non-linear problems. Explicit solver is particularly useful for modelling tough adhesives that exhibit high strain to failure.

The term 'convergence' is used to indicate that the process for solving the equation system converges. The solution at the end of an increment is, by definition, converged. Failure to find a solution for a given increment results in the solver terminating the analysis - the model is deemed to have failed to converge. Failure to converge is generally associated with unstable material behaviour, and occurs more often with non-linear analyses. Convergence problems are less likely to occur if the model is loaded with applied displacements rather than applied loads.

Materials Modelling>>

Finite Element Analysis

Materials Modelling

A number of approaches have been adopted by engineers/designers for predicting the deformation and static strength (failure load) of adhesives and adhesively bonded structures. A number of these materials models are briefly described below.

VON MISES YIELD CRITERION

The most simple yield criterion interprets yielding as a purely shear deformation process which occurs when the effective shear stress σ_e reaches a critical value. This effective stress is defined in terms of principal stress components σ_i (i = 1, 2 or 3) by [23, 26]:

$$\sigma_e = \left\{ \frac{1}{2} \left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right] \right\}^{1/2}$$

The von Mises criterion then relates σ_e to the yield stress in tension σ_T by

$$\sigma_e = \sigma_T$$

Materials Modelling Continued…..

Finite Element Analysis

Materials Modelling Continued

LINEAR DRUCKER-PRAGER YIELD CRITERION

A simple modification of the von Mises criterion that includes hydrostatic stress sensitivity, known as the linear Drucker-Prager model, follows from the equation above [23, 26]:

$$\sigma_e = \sigma_o - \mu\sigma_m$$

Here σ_o is a material parameter that is related to the shear yield stress σ_S by:

$$\sigma_o = \sqrt{3}\,\sigma_S$$

and σ_m is the hydrostatic stress given in terms of principal stresses by:

$$\sigma_m = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3)$$

The parameter μ depends on the adhesive material and characterises the sensitivity of yielding to hydrostatic stress. A value for μ is determined from tests under two different stress states.

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Materials Modelling Continued…..

Finite Element Analysis

Materials Modelling Continued

EXPONENT DRUCKER-PRAGER YIELD CRITERION

The exponent Drucker-Prager criterion is significantly more accurate for predicting deformation than the linear version under stress conditions where there is a high hydrostatic stress component (e.g. adhesive joints). This model can be written in the form [23, 26]:

$$\sigma_e^2 = \lambda\sigma_T^2 - 3(\lambda - 1)\sigma_T\sigma_m$$

where λ is another hydrostatic stress sensitivity parameter. The exponent Drucker-Prager criterion is implemented in ABAQUS with an exponent parameter of 2.

This equation can then be expressed in the form:

$$aq^2 = p + p_1$$

where $q = \sigma_e$ and $p = -\sigma_m$. Comparison of the above two equations then gives the following relationships between model parameters:

$$a = \frac{1}{3\sigma_T(\lambda - 1)} \quad \text{and} \quad p_1 = a\lambda\sigma_T^2$$

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Materials Modelling Continued…..

Finite Element Analysis

Materials Modelling Continued

CAVITATION MODEL

The linear Drucker-Prager yield criterion $\sigma_e = \sigma_o - \mu\sigma_m$ is modified as follows to give a yield function Φ that includes the effect of cavitation of the rubber on yield stresses for the adhesive [26]:

$$\Phi = \frac{\sigma_e^2}{\sigma_M^2} - (qf)^2 + 2qf \cosh \frac{3\sigma_m}{2\sigma_M} - \left(1 - \frac{\mu\sigma_m}{\sigma_M}\right)^2 = 0$$

Here f is the effective volume fraction of cavities, which at small strains is zero but increases rapidly over some characteristic strain region responsible for cavity nucleation. The parameter q has been included to account for the effect of void interactions on the stress distribution in the matrix between cavities. The yield stress σ_M is the effective yield stress of the matrix polymer between cavities and is equal to σ_o in the absence of cavities. . The effective yield stresses rises with increasing f as the volume fraction of the rubber in the matrix decreases. The increase in σ_M as cavities nucleate is given by the expression [26]:

$$\sigma_M = \frac{\sigma_o}{1 - kv_{Ro}}\left[1 - k\frac{(v_{Ro} - f_n)}{1 - f_n}\right]$$

where v_{Ro} is the volume fraction of rubber and f_n is the effective volume fraction of cavities created by cavity nucleation in rubber particles. At large strains, the quantity f may be larger than f_n as a result of cavity growth [26] subsequent to nucleation. The parameter k relates the shear yield stress of an uncavitated rubber-toughened adhesive to the volume fraction of rubber v_{Ro} by the equation [26]:

$$\sigma_o = \sigma_{o1}(1 - kv_{Ro})$$

where σ_{o1} is the effective shear yield stress of the untoughened adhesive. It can now be seen that, for uncavitated material, f = 0 and σ_e reduces to $\sigma_e = \sigma_o - \mu\sigma_m$. In the cavitation model, $\sigma_o = \sqrt{3}\,\sigma_s$ is derived from experimental data obtained in shear and replaces σ_T as the basic hardening function for the polymer.

Cavitation Model Continued…..

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Finite Element Analysis

CAVITATION MODEL CONTINUED

The nucleation of a cavity in a rubber particle is assumed to occur at some critical volumetric strain that decreases with increasing particle diameter. For a distribution of particle sizes, the cavity nucleation should then occur over a range of total volumetric strain ε_V related to the critical strain range for the rubber particles. Through comparisons with experimental data for a number of plastics and adhesives materials, it has been shown that the increase in the volume fraction of nucleated cavities f_n with volumetric strain is given with satisfactory accuracy by the expressions [26]:

$$f = 0 \qquad \text{for } \varepsilon_v \le \varepsilon_{1v}$$

$$f = v_{Ro}\left\{1 - \exp\left[-\left(\frac{\varepsilon_v - \varepsilon_{1v}}{\varepsilon_{2v}}\right)^{\beta}\right]\right\} \quad \text{for } \varepsilon_v > \varepsilon_{1v}$$

The parameters ε_{1V}, ε_{2V} and β determine the location and breadth of the volumetric strain range over which cavity nucleation occurs.

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Materials Modelling Continued…..

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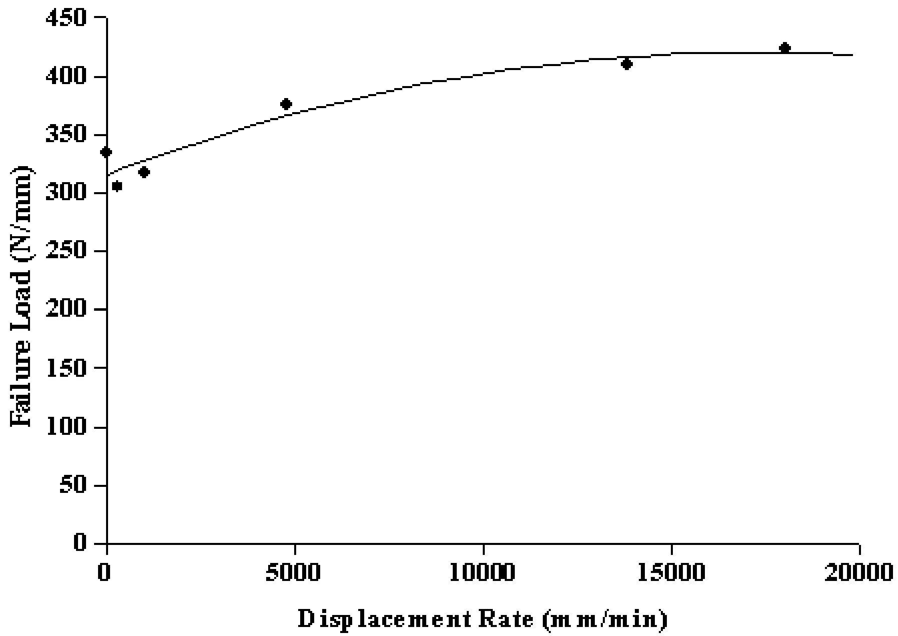
Finite Element Analysis

Materials Modelling Continued

RATE DEPENDENCE

Joint strength and stiffness (see table and figure below) are generally strain rate dependent, thus designers are advised to use engineering data generated at the strain rate to be experienced by the adhesive layer in the bonded structure.



Failure load versus displacement rate for CR1/AV119 single-lap joint

Shear Strength Rate Dependence for AV119/CR1 Single-Lap Joint

Displacement Rate(mm/min)	Load/Width(N/mm)
1	334 ± 11
300	305 ± 10
1,020	317 ± 5
4,800	376 ± 4
13,800	410 ± 8
18,000	423 ± 8

Predicting Failure>>

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Finite Element Analysis

Materials Modelling Continued

Predicting Failure

Outputs required generally include the load-displacement (force-extension) curve and predicted loads corresponding to the appropriate failure criteria for the adhesive (in the case of bonded joints) and each layer in the case of composite laminates. There are numerous failure criteria for adhesives. Comparisons of maximum values of selected components of stress and strain in the adhesive, at extensions where failure is considered to initiate in different types of joints, reveal that the maximum principle stress and hydrostatic stress are approximately constant at failure [26]. This applies to bulk adhesives and adhesive joints. Furthermore, the location of failure initiation has been shown to occur in adhesive joints at the location predicted by the cavitation model in combination with the hydrostatic stress criterion.

For composite materials, the failure strengths of the different plies are used. The stress components required are S11, S22 and S12 (local orientations). The failure strengths in these directions for the individual plies are either measured or predicted using the combination of micromechanics and classical laminate analysis. For the failure criteria to work correctly, the 1- and 2-directions of the material constants defined for the composite sections must align with the fibre and the transverse to the fibre directions, respectively. Further information on the use of FEA for design with adhesives can be found in NPL Measurement Good Practice Guide No. 48 [23].

Design Software>>

Design Software

Techniques for stress analysis of a joint generally fall into two main categories: analytical, closed-form methods and finite element methods. Analytical methods are generally quick and easy-to-use, but are only suitable for simple geometries. These methods cannot accurately predict stresses and strains as the analytical equations, by their simple nature, cannot fully account for the complete stress and strain conditions within the joint. Finite element methods have an advantage in that almost any geometrical shape can be analysed and are capable of more accurate analysis of stress and strain distributions. The disadvantages of these methods are that analyses are expensive and specialist knowledge is required. An ideal method would be an accessible yet accurate stress analysis technique. This section reviews finite element analysis (FEA) and analytical based software developed for the analysis and design of bolted and bonded structures, and materials selection.

FEA Programs

Numerical analysis techniques, such as FEA, are used extensively in the design and stress analysis of adhesively bonded and bolted structures. These techniques offer solutions to complex problems that are too difficult or impossible to resolve using analytical, closed-form solutions. Numerous FEA codes are available (see table below). These codes provide in-built constitutive models for simulating the behaviour of most adhesives, allowing for non-uniform stress-strain distributions, geometric non-linearity, hygrothermal effects, elastic-plastic and visco-elastic behaviour, static and dynamic analysis, and strain rate dependence. Orthotropic element types include two-dimensional (2-D) solid plane-stress or plain-strain elements, axisymmetric shell or solid elements, three-dimensional (3-D) solid or "brick" elements and crack-tip elements. A number of automatic mesh (element) generators are available with post-processing capabilities (e.g. PATRAN and FEMGV).

Although numerical methods are able to accommodate complex geometries, loadings, material properties and boundary conditions, the solutions generated are only approximations to the actual solution. It is important that the designer/analyst is aware of the limitations of the numerical techniques being applied and has a fundamental understanding of the mechanics of bonded and bolted joints (i.e. stresses and failure mechanisms).

Stress analyses (especially FE methods) are often used to compare stress/strain distributions obtained from different joint configurations (e.g. lap, scarf and butt joints) or geometries (varying adhesive and adherend thickness, overlap lengths, fillet shapes). Hence, finite element stress analysis can be used as a tool for optimising the design of a joint. Evolutionary optimisation method **EVOLVE** has been used to optimise the shape of adhesive fillets [64]. This process allows selected properties to drive the optimisation process (e.g. minimising the maximum principal stress in the adhesive). **EVOLVE** relies on an iterative FE analysis and the progressive removal of elements using a rejection criterion. This takes the guesswork out of the design process.

Finite Element Packages (see also [57])

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Name	Supplier	Application	Features
ABAQUS	Hibbit, Karlsson & Sorenson, Inc.	• General purpose FE program • Linear, non-linear and coupled analysis • Large materials model library	• Anisotropic material models in all elements • 2-D and 3-D plate/shell and solid elements • Temperature and strain-rate dependence of properties • Fracture mechanic/ crack propagation analysis • Maximum stress and strain, Tsai-Hill, Tsai-Wu, Azzi-Tsai-Hill and user defined failure criteria
ANSYS	Swanson Analysis System Inc.	• General purpose FE program • Non-linear analysis (non-composite applicable) • Pre- and post-processing	• Isotropic and orthotropic material properties • 2-D and 3-D plate/shell and solid elements • Laminated shell elements • Crack-tip solid and thick-shell elements • Maximum stress and strain, Tsai-Wu and user defined failure criteria
LUSAS	FEA Ltd.	• General purpose FE program • Linear and non-linear analysis • Static, creep, fatigue and dynamic analysis	• Plate/shell and solid elements • 2-D and 3-D interface elements • Laminate analysis/Hashin damage model • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill, Hoffmann, Hashin, Tsai-Wu failure criteria
COSMOS	Structural Research& Analysis Corp.	• General purpose FE program • Linear and non-linear analysisStatic and dynamic analysis • 2-D and 3-D structural and thermal models	• Plate/shell and solid elements • Delamination elements for fracture mechanics • Fatigue analysis of structural components • Tsai-Hill and Tsai-Wu failure criteria
NASTRAN	MacNeal-SchwendlerCorp.	• General purpose FE program • Static and dynamic analysis • Linear and non-linear analysis • Pre- and post-processing by PATRAN	• Plate/shell elements - allowance for orthotropic temperature-dependent properties • Ply stresses • Tsai-Hill, Hoffmann and Tsai-Wu failure criteria
NISA	Engineering Mechanics Research Corp.	• General purpose FE program • Linear, non-linear and coupled analysis	• Solid and thick-shell elements • Laminated shell elements • In-plane and interlaminar stresses • Maximum stress and strain, von Mises, Tsai-Wu, delamination failure criteria

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AEA Harwell Didcot Oxfordshire, OX11 0QJ UK Tel: 0870 1901900	ISE Institute of Structural Engineers 11 Upper Belgrave Street London, SW1X 8BH UK Tel: 020 72354535	QinetiQ Cody Technology Park Farnborough Hampshire, GU14 0LX UK Tel: 01252 392000
ASTM American Society for Testing and Materials 100 Barr Harbor Drive West Conshohocken Pennsylvania USA Tel: 001 610 8329500	ISO International Standards Organisation 1, ch. de la Voie-Creuse Case Postale 56 CH-1211 Genève20 Switzerland Tel: +41 22 7490111	SATRA SATRA Footwear Technology Centre SATRA House Rockingham Road Kettering Northants, NN16 9JH UK Tel: 01536 410000
BSI British Standards Institution British Standards House 389 Chiswick High Road London, W4 4AL UK Tel: 020 89969001	MERL Ltd Materials Engineering Research Laboratory Ltd Wilbury Way Hitchen Hertfordshire, SG4 0TW UK Tel: 01462 427850	TWI (Formerly The Welding Institute) Granta Park Great Abington Cambridge, CB21 6AL UK Tel: 01223 899000

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Appendices

Appendix 1: TEST METHODS

Appendix 1 describes test methods for determining input design/analysis data

Please click here for Appendix 1

Appendix 2: DESIGN PROCEDURES FOR A SINGLE-LAP JOINT

Appendix 2 describes a simple analytical procedure, which can be used as a preliminary tool to produce satisfactory single-lap joints. The procedure does not take into account thermal and moisture effects which would be needed for the final design.

Please click here for Appendix 2

Appendix 3: CLOSED FORM SOLUTIONS FOR BONDED T-JOINTS

Appendix 3 contains the closed form solutions for bonded T-joints. The solutions are given for vertical, horizontal and tangential deflections. To demonstrate the accuracy of the closed form solutions a comparison of measured and predicted stiffness is provided.

Please click here for Appendix 3

Appendix 1: Test Methods

Table A1.1: Test Methods for Determining Input Design/Analysis Data

Material Property	Standard/Test Method
Elastic Properties - Adherends Metals E, G, ν Composites In-plane (E_{XX}, $E_{YY,}$ ν_{XY}) Through-thickness (E_{ZZ}, ν_{XZ}, ν_{YZ}) In-plane shear (G_{XY}) Through-thickness shear (G_{XZ}, G_{YZ})	 Tensile test of plastics - BS EN ISO 527-2 **m.d = multidirectional, u.d = unidirectional** Tension - BS EN ISO 527-4 (m.d)/BS EN ISO 527-5 (u.d) T-T tension and compression-NPL draft procedures ±45° tension method - BS EN ISO 14129 (u.d)* V-notched beam test - ASTM D 5379
Strength Properties - Adherends Metals Tension Compression Shear Composites In-plane tension (S_{XX}, S_{YY}) Through-thickness tension (S_{ZZ}) In-plane compression (S_{XX}, S_{YY}) Through-thickness compression (S_{ZZ}) In-plane shear (S_{XY}) Through-thickness (S_{XZ})	 Tensile testing of metallic materials - BS EN 10002-1 Compression testing of metallic materials - ASTM E9 Shear modulus - BS EN 10002-1* **m.d = multidirectional, u.d = unidirectional** Tensile - BS EN ISO 527-4 (m.d)/BS EN ISO 527-5 (u.d) Through-thickness tension - NPL draft procedure Compression - BS EN ISO 14126 Through-thickness compression - NPL draft ±45° tension method - BS EN ISO 14129 (u.d) V-notched beam method - ASTM D 5379 (u.d)
Elastic Properties - Adhesives E, G, ν	 Tensile test of plastics - ISO 527-2 V-notched beam method - ASTM D 5379
Strength Properties - Adhesives Tension Compression Shear Maximum principal strain	 Tensile test of plastics - BS EN ISO 527-2 Compressive testing of rigid plastics - IS0 604/ ASTM D695 V-notched beam method - ASTM D 5379 Tensile test of plastics - BS EN ISO 527-2
Fracture Toughness Mode I – composites Mode I – adhesive joints Mode II – composites Mode II – adhesive joints	 Double cantilever beam (DCB) test - ISO 15024/prEN 6033 As above – draft BSI under review End notched flexure (ENF) test - prEN 6034 As above – no national or international standards

Table continued…..

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Appendix 1: Test Methods Continued

Table A1.1: Test Methods for Determining Input Design/Analysis Data

Material Property	Standard/Test Method
Joint Coupon Tests	
Tension shear strength and modulus	Butt joint
Shear strength and modulus	Thick adherend shear test (tension) - ISO 11003-2
Compression strength and modulus	Butt joint
Additional Tests	
Tensile strength of lap joint	BS EN 1465
Tension-tension fatigue	BS EN ISO 9664
Moisture absorption/conditioning	BS EN ISO 62
Effect of water/moisture	ISO 62/ISO 175
Effect of chemicals	ISO 175
Effect of hear ageing	ISO 216
Test and conditioning atmospheres	ISO 291
Tensile creep behaviour of plastics	ISO 899-1
Failure patterns	EN 923
Dynamic Mechanical Analysis	ISO 6721-4
Differential Scanning Calorimetry	ISO 11357

Symbols: E = modulus of elasticity, G - shear modulus, ν = Poisson's ratio, S = strength
Subscripts: XX, YY and XY denote in-plane properties, XZ, YZ and ZZ denote through-thickness properties
* Plate twist method - ISO 15310 (simple test for measuring shear modulus only)

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Appendix 2: Design Procedures For a Single-Lap Joint

Single lap joints create bending loads in the adherends and tensile stresses in the adhesive with the result that the joint becomes very inefficient. Designing with single-lap joints should be avoided unless the overlap to thickness ratio is greater than 10, such that the transverse deflections under tensile load can relieve the eccentricity in the load path, thus producing acceptable structural efficiencies. Stress is transferred from one adherend through the adhesive to the second adherend. These stresses are highly non-linear (Figure A2.1), increasing rapidly near the ends.

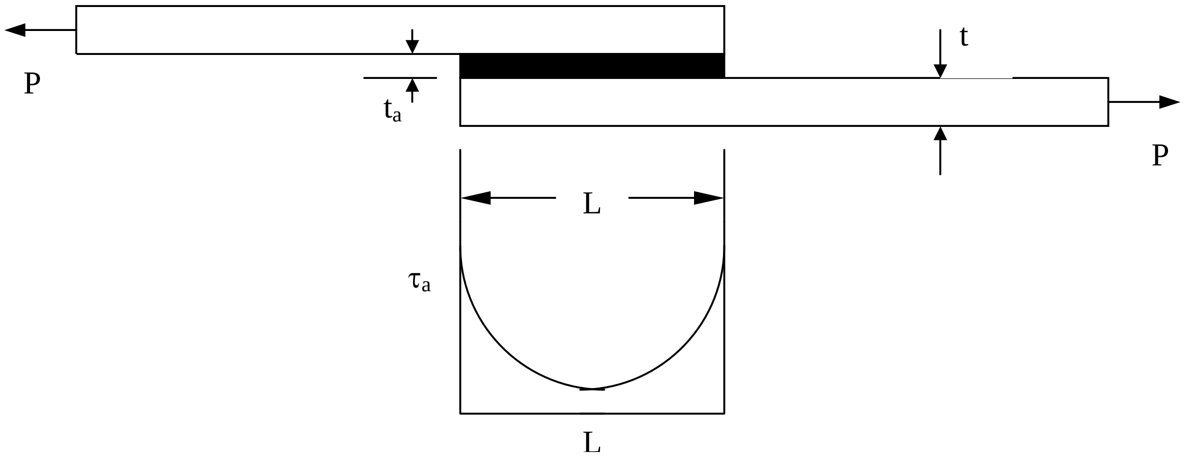

Figure A2.1: Schematic of single-lap joint.

The design procedure for a single lap joint assumes a perfect bond between the adhesive and adherend. The following criteria need to be satisfied:

1. The maximum adhesive shear stress is to be less than or equal to the maximum allowable adhesive shear stress.

2. The maximum adhesive peel stress is to be less than or equal to the maximum allowable adhesive tensile stress.

3. The maximum adherend tensile stress in the through thickness direction is to be less than or equal to the maximum allowable adherend through thickness tensile stress.

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Appendix 2: Design Procedures For a Single-Lap Joint Continued

The following design procedure is used in the EUROCOMP design code and handbook [8].

Step 1: The parameter β**/t** is calculated as follows:

$$\frac{\beta}{t} = \sqrt{8\frac{G_a}{Et_a t}} \qquad \text{(A2.1)}$$

where: G_a = adhesive shear modulus E = adherend tensile modulus t = adherend thickness t_a = adhesive layer thickness

This is used to obtain the lap length, **L** (= **2c**), by reading the appropriate value for **c** from a plot of **c** as a function of β**/t** (see Figure 5.3.13 in EUROCOMP design code). There is no mention of where this curve originates. β is related to the load transfer length (the distance to transfer 95% of the load).

Step 2: Calculate the maximum adhesive shear stress at the ends of the joint:

$$\tau_{max} = \frac{\sigma}{8}(1+3k)\sqrt{8\frac{G_a t}{Et_a}} \qquad \text{(A2.2)}$$

where:

$$\sigma = \frac{P_d \gamma_f}{t} \qquad \text{(A2.3)}$$

$$k = \frac{\cosh(u_2 c)\sinh(u_1 L)}{\sinh(u_1 L)\cosh(u_2 c) + 2\sqrt{2}\cosh(u_1 L)\sinh(u_2 c)} \qquad \text{(A2.4)}$$

$$u_1 = 2\sqrt{2}\, u_2 \qquad \text{(A2.5)}$$

$$u_2 = \frac{1}{\sqrt{2}\, t}\sqrt{\frac{3\sigma(1-\nu^2)}{E}} \qquad \text{(A2.6)}$$

and P_d = design load per unit width ν= adherend's Poisson's ratio γ_f = partial safety factor

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Appendix 2: Design Procedures For a Single-Lap Joint Continued

Step 3: The calculated value of τ_{max} should be checked against the maximum allowable adhesive shear stress, i.e.

$$\tau_{max} \leq \tau_{allowable} \qquad \text{(A2.7)}$$

Step 4: Calculate the value of λ as follows:

$$\lambda = \frac{c}{t}\left(\frac{6E_a t}{Et_a}\right)^{0.25} \qquad \text{(A2.7)}$$

where:

$$c = \frac{L}{2} \qquad \text{(A2.8)}$$

and $\mathbf{E_a}$ = adhesive tensile modulus.

Providing that λ is greater than 2.5, the maximum value of peel stress at the joint end for a long overlap is:

$$\sigma_{max} = \sigma\left[\frac{k}{2}\sqrt{6\frac{E_a t}{Et_a}} + \frac{k't}{c}\sqrt{6\frac{E_a t}{Et_a}}\right] \qquad \text{(A2.9)}$$

where

$$k' = \frac{kc}{t}\left[\frac{3(1-v^2)\sigma}{E}\right] \qquad \text{(A2.10)}$$

Step 5: Investigate the magnitude of the adhesive peel stress with respect to the allowable adhesive tensile strength and the allowable adherend through-thickness tensile strength as follows:

$$\sigma_{max} \leq \sigma_{allowable} \qquad \text{(A2.11)}$$

$$\sigma_{max} \leq \sigma_{z\ allowable} \qquad \text{(A2.12)}$$

If the design criteria are met, the design procedure is completed. If the criteria are not met, the lap length can be increased to try to reduce τ_{max} and σ_{max}.

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Appendix 2 Continued…..

Appendix 2: Design Procedures For a Single-Lap Joint Continued

Alternative Approach

The following summarises a simple model suggested by Adams [66] for predicting the ultimate strength for lap joints bonded with either toughened or flexible adhesives. The model assumes that the absolute maximum strength for a lap joint occurs when the whole of the adhesive layer is at the shear yield strength, and cohesive failure of the adhesive occurs. The shear yield stress of the adhesive layer needs to be determined separately using for example the Arcan or thick adherend shear tests. The yield shear stress used in the calculation needs to be representative of the test or service conditions to be experienced by the lap joint. Hence, the effects of temperature, moisture and load/displacement rate on the shear yield stress need to be included.

The average shear stress within the bondline can be calculated as follows [66]:

$$\tau_{AVE} = \frac{F}{A} \qquad \text{(A2.13)}$$

where **F** is the applied load and **A** is the area of the bondline.

An additional requirement is that the stresses in the connecting substrates $\sigma_{S;MAX}$ would not exceed the yield stress of the substrate material. For a single lap joint, the maximum stress in the substrate is equal to:

$$\sigma_{S;MAX} = \frac{N}{A_S} + \frac{M}{W_S} \qquad \text{(A2.14)}$$

A_S is the cross-sectional area of the substrate and W_S is section modulus of the substrate. The axial load and moment are equal to:

$$N = F \qquad \text{(A2.15)}$$

$$M = \frac{1}{2} t_S k F \qquad \text{(A2.16)}$$

where t_S is the substrate thickness, and **k** is the reduction factor [56]:

$$k = \frac{1}{1 + 2\sqrt{2}\tanh \dfrac{\sqrt{\dfrac{12(1-\nu^2)F}{E_S t_S^3}}\left(\dfrac{L}{2}\right)}{2\sqrt{2}}} \qquad \text{(A2.17)}$$

L is the overlap length. The reduction factor **k** is only valid for single lap joints constructed from adherends with equal thickness.
Yielding of the substrate is determined by comparing the maximum linear elastic stress with the design value of the yield stress.

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Appendix 3: Closed Form Solutions For Bonded T-Joints

Parameters

a = width of flange, web and base plate
b_1 = web thickness
b_2 = flange thickness
b_3 = base plate thickness
E = longitudinal stiffness
I = second moment of area = $ab^3/12$
L_{AB} = length AB
L_{BC} = length BC
L_{CD} = length CD
L_{DE} = length DE
L_{EF} = length EF (fully clamped)
M = bending moment
P = vertical load
R = inner radius of flange
s_1 = length AW
s_2 = length BX
s_3 = length CY
s_4 = length DZ
T = transverse load
δ = displacement
θ = angle

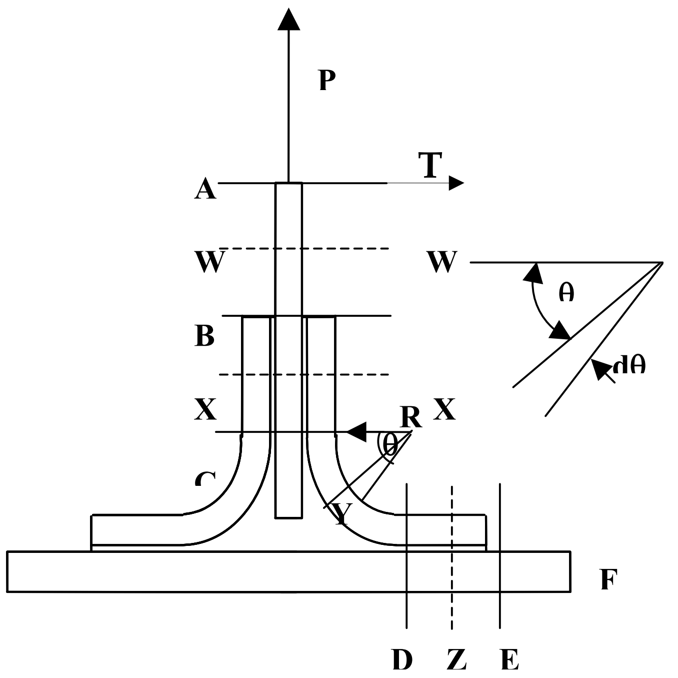

Figure A3.1: Schematic of bonded T-joint.

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Appendix 3: Closed Form Solutions For Bonded T-Joints Continued

A3.1 VERTICAL DEFLECTION

The following analysis uses Castigliano's energy method to analyse one-half of the T-joint, which is assumed to be symmetric about the vertical axis.

$$\delta = \int \frac{M}{EI}\frac{\partial M}{\partial P}\,ds \qquad \text{(A3.1)}$$

Deflection for AB

$$\delta_{AB} = \int \frac{M_{WW}}{EI}\frac{\partial M_{WW}}{\partial P}\,ds_1 = 0; \quad M_{WW} = 0 \qquad \text{(A3.2)}$$

Deflection BC

$$\delta_{BC} = \int \frac{M_{XX}}{EI}\frac{\partial M_{XX}}{\partial P}\,ds_2 = 0; \quad M_{XX} = 0 \qquad \text{(A3.3)}$$

Deflection for DE

$$\delta_{DE} = \int \frac{M_{ZZ}}{EI}\frac{\partial M_{ZZ}}{\partial P}\,ds_4 \qquad \text{(A3.4)}$$

$$M_{ZZ} = P(R + s_4); \quad \frac{\partial M_{ZZ}}{\partial P} = R + s_4 \qquad \text{(A3.5)}$$

$$\delta_{DE} = \frac{P}{EI}\int_0^{L_{DE}} (R + s_4)(R + s_4)\,ds_4 \qquad \text{(A3.6)}$$

$$\delta_{DE} = \frac{P}{EI}\int_0^{L_{DE}} \left[R^2 + 2Rs_4 + (s_4)^2\right] ds_4 \qquad \text{(A3.7)}$$

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Appendix 3: Closed Form Solutions For Bonded T-Joints Continued

A3.1 VERTICAL DEFLECTION CONTINUED

$$\delta_{DE} = \frac{P}{EI}\left[R^2 s_4 + R(s_4)^2 + \frac{(s_4)^3}{3}\right]_0^{L_{DE}} \tag{A3.8}$$

$$I = I_{DE} = \frac{a(b_2 + b_3)^3}{12} \tag{A3.9}$$

Deflection for CD

$$\delta_{CD} = \int \frac{M_{YY}}{EI}\frac{\partial M_{YY}}{\partial P} ds_3 \tag{A3.10}$$

$$M_{YY} = P(R - R\cos\theta); \quad \frac{\partial M_{YY}}{\partial P} = R - R\cos\theta; \quad ds_3 = Rd\theta \tag{A3.11}$$

$$\delta_{CD} = \frac{P}{EI}\int_0^{\pi/2}(R - R\cos\theta)(R - R\cos\theta)Rd\theta \tag{A3.12}$$

$$\delta_{CD} = \frac{PR^3}{EI}\int_0^{\pi/2}\left[1 - 2\cos\theta + \frac{1+\cos 2\theta}{2}\right]d\theta \tag{A3.13}$$

$$\delta_{CD} = \frac{PR^3}{EI}\left[\theta - 2\sin\theta + \frac{\theta}{2} + \frac{\sin 2\theta}{4}\right]_0^{\pi/2} d\theta \tag{A3.14}$$

$$\delta_{CD} = \frac{PR^3}{EI}\left[\frac{\pi}{2} - 2 + \frac{\pi}{4}\right] = \frac{PR^3}{EI}\left[\frac{3\pi}{4} - 2\right] \tag{A3.15}$$

$$I = I_{CD} = \frac{a(b_2)^3}{12} \tag{A3.16}$$

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Appendix 3: Closed Form Solutions For Bonded T-Joints Continued

A3.1 VERTICAL DEFLECTION CONTINUED

The total vertical deflection of the full T-joint at A is given by:

$$\delta_{TOTAL} = \frac{(\delta_{AB} + \delta_{BC} + \delta_{CD} + \delta_{DE})}{2} \tag{A3.17}$$

A3.2 HORIZONTAL OR LATERAL DEFLECTION

Deflection for DE

$$\delta_{DE} = \int \frac{M_{ZZ}}{EI}\frac{\partial M_{ZZ}}{\partial T} ds_4 = 0; \quad M_{ZZ} = 0 \tag{A3.18}$$

Deflection for AB

$$\delta_{AB} = \int \frac{M_{WW}}{EI}\frac{\partial M_{WW}}{\partial T} ds_1 \tag{A3.19}$$

$$M_{WW} = Ts_1; \quad \frac{\partial M_{WW}}{\partial T} = s_1 \tag{A3.20}$$

$$\delta_{AB} = \frac{T}{EI}\int_0^{L_{AB}} (s_1)^2 \, ds_1 \tag{A3.21}$$

$$\delta_{AB} = \frac{T}{EI}\left[\frac{(s_1)^3}{3}\right]_0^{L_{AB}} \tag{A3.22}$$

$$I = I_{AB} = \frac{a(b_1)^3}{12} \tag{A3.23}$$

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Appendix 3 Continued…..

Appendix 3: Closed Form Solutions For Bonded T-Joints Continued

A3.2 HORIZONTAL OR LATERAL DEFLECTION CONTINUED

Deflection for BC

$$\delta_{BC} = \int \frac{M_{XX}}{EI} \frac{\partial M_{XX}}{\partial T} ds_2 \tag{A3.24}$$

$$M_{XX} = Ts_2; \quad \frac{\partial M_{XX}}{\partial T} = s_2 \tag{A3.25}$$

$$\delta_{BC} = \frac{T}{EI} \int_0^{L_{BC}} (s_2)^2 \, ds_2 \tag{A3.26}$$

$$\delta_{BC} = \frac{T}{EI} \left[\frac{(s_2)^3}{3} \right]_0^{L_{BC}} \tag{A3.27}$$

$$I = I_{BC} = \frac{a(0.5b_1 + b_2)^3}{12} \tag{A3.28}$$

Deflection for CD

$$\delta_{CD} = \int \frac{M_{YY}}{EI} \frac{\partial M_{YY}}{\partial T} ds_3 \tag{A3.29}$$

$$M_{YY} = T(R + R\sin\theta); \quad \frac{\partial M_{YY}}{\partial T} = R + R\sin\theta; \quad ds_3 = Rd\theta \tag{A3.30}$$

$$\delta_{CD} = \frac{T}{EI} \int_0^{\pi/2} (R + R\sin\theta)(R + R\sin\theta) Rd\theta \tag{A3.31}$$

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A3.2 HORIZONTAL OR LATERAL DEFLECTION CONTINUED

$$\delta_{CD} = \frac{TR^3}{EI}\int_0^{\pi/2}\left[1 + 2\sin\theta + \frac{1-\cos 2\theta}{2}\right]d\theta \qquad \text{(A3.32)}$$

$$\delta_{CD} = \frac{TR^3}{EI}\left[\theta - 2\cos\theta + \frac{\theta}{2} - \frac{\sin 2\theta}{4}\right]_0^{\pi/2} d\theta \qquad \text{(A3.23)}$$

$$\delta_{CD} = \frac{TR^3}{EI}\left[\frac{\pi}{2} + \frac{\pi}{4}\right] = \frac{TR^3}{EI}\left[\frac{3\pi}{4}\right] \qquad \text{(A3.34)}$$

$$I = I_{CD} = \frac{a(b_2)^3}{12} \qquad \text{(A3.35)}$$

The total horizontal deflection of the full T-joint at A is given by:

$$\delta_{TOTAL} = \frac{(\delta_{AB} + \delta_{BC} + \delta_{CD} + \delta_{DE})}{2} \qquad \text{(A3.36)}$$

A3.3 TANGENTIAL LOADING (ϕ)

For the case where load **V** is applied at an angle ϕ, the load can be resolved into vertical and horizontal components. The vertical and horizontal load components **P** and **T** can be determined using the following relationships:

$$P = V\sin\phi; \quad T = V\cos\phi \qquad \text{(A3.37)}$$

The tangential displacement δ_V can be calculated as follows:

$$\delta_V = \sqrt{(\delta_P)^2 + (\delta_T)^2} \qquad \text{(A3.38)}$$

where the vertical deflection δ_P and horizontal deflection δ_T are calculated using the analyses in Sections A3.1 and A3.2, respectively.

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Appendix 3 Continued…..

Appendix 3: Closed Form Solutions For Bonded T-Joints Continued

A3.4 COMPARISON OF MEASURED AND PREDICTED STIFFNESS

Table A3.1: Comparison of Predicted and Measured T-Joint Stiffness [54]

Configuration	FEA	Analytical	Measured
Direct Tension			
Aluminium	56.80	56.80	58.58 ± 4.38
GRP	16.05	16.05	17.15 ± 2.38
Transverse Tension			
Aluminium	3.74	3.74	3.88 ± 0.38
GRP	1.06	1.06	1.25 ± 0.11

Table A3.1 compares predicted joint stiffness determined using the cavitation model (FEA) and closed form solution described above.

A3.5 PARAMETRIC ASSESSMENT

Using the closed form solutions for the two loading modes it is possible to determine the effect of geometric parameters on joint stiffness. Figure A3.2 shows the effect of varying base plate thickness and flange radius on joint stiffness of aluminium T-joints under direct tension loading.

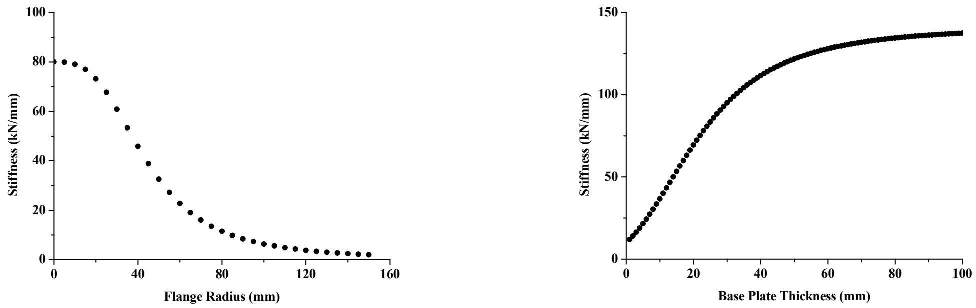

Figure A3.2: Effect of base plate thickness/flange radius on aluminium joint stiffness

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