# Design Requirements for Bonded and Bolted Composite Structures

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**ABSTRACT**

This report provides an overview of different approaches and industrial codes of practice used for the design and analysis (including data requirements and methods of data generation) of bolted and bonded composite structures for use under monotonic, fatigue and creep (i.e. static) loading conditions.  The report is concerned with structural applications where adhesives or bolts are used to join primary load-bearing multi-component assemblies.  It considers numerical and analytical solutions for the design and analysis of metallic and composite joints, and sandwich structures.  Finite element analysis and closed-form analysis based design software are reviewed.  The report provides recommendations as to the preferred design procedures currently practiced by industry to be used for the initial design of a generic joint configuration to be evaluated within the DTI funded Measurements for Materials project MMS11 "Design for Fatigue and Creep in Joined Systems".

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# 1    INTRODUCTION

Bolted and bonded joints are frequently expected to sustain static or cyclic fatigue loads for considerable periods of time without any adverse effect on the load-bearing capacity of the structure (rail road bridges and automotive parts are expected to last at least 120 and 10 years, respectively).  A major concern is that under dynamic fluctuating loads, joints will often fail at stress levels much lower than the static strength of the joint.  Fatigue critical structures containing bolted and/or bonded metallic assemblies include aircraft, ships, trucks, railroad cars, machinery, and highway and railroad bridges).  It is estimated that 90% of all structural failures that occur in service are caused by fatigue.  Hence, the importance of designing structures to avoid fatigue failure.

At present, there is insufficient information available as to the reliability of methodologies for predicting the fatigue and creep behaviour of bolted and bonded joints and their constituent components.  A lack of suitable material models and failure criteria has resulted in a tendency to "overdesign" composite structures.  Safety authorities will often require that adhesively bonded structures, particularly those employed in primary load-bearing applications, include mechanical fasteners (e.g. bolts) as an additional safety precaution.  Conservative design and engineering practices result in heavier and more costly components.  The development of reliable design and predictive methodologies for bolted and bonded structures can be expected to result in more efficient use of composites and adhesives, lower structural weight, and lower material and manufacturing costs.  The design and manufacture of safe and cost-effective, structurally efficient joints is a major challenge not only to the aerospace industry, but also for non-aerospace applications where most of the expansion in the use of adhesives and composites is expected to occur.

This report provides an overview of different approaches and industrial codes of practice used for the design and analysis (including data requirements and methods of data generation) of bolted and bonded composite structures for use under monotonic, fatigue and creep (i.e. static) loading conditions.  It includes a review of finite element analysis (FEA) and analytical based design software and a section dedicated to design requirements for composite sandwich constructions (i.e. panels).  The report concludes with recommendations as to the preferred design procedures currently practiced by industry to be used for the initial design of a generic joint configuration to be evaluated within the DTI funded Measurements for Materials project MMS11 "Design for Fatigue and Creep in Joined Systems".

The report consists of seven sections (including the Introduction).  Section 2 provides an overview of approaches to design, traditionally adopted for adhesively bonded and bolted metallic structures (e.g. steel and aluminium frames) for use under monotonic, fatigue and creep loading conditions.  Design methodologies for bolted and bonded composite structures are covered in Sections 3 and 4.  Industrial codes of practice are covered in these two sections. FEA and analytical design/analysis software for bolted and bonded composite structures are reviewed in Section 5.  Design requirements for composite sandwich panels are covered in Section 6.  Section 7 includes a general discussion, conclusions and recommendations as to the preferable design procedures currently practiced by industry to be used for the initial design of a generic joint configuration to be evaluated within MMS11.  Test methods for adhesives, adherends, and bonded and bolted joints are summarised in Appendix 1.  Appendix 2 describes a simple analytical procedure that can be used for producing satisfactory adhesively bonded single-lap joints.  Appendix 3 provides a list of test methods/standards for sandwich panels.

## 2. OVERVIEW OF BONDED AND BOLTED JOINT DESIGN

There are many advantages for bonding composites, particularly from a fatigue performance perspective. Adhesively bonded joints have excellent fatigue properties. Adhesives are used extensively for secondary (not main load-bearing frame) aircraft and automotive parts, and in some cases primary parts of aircraft. This method of joining is particularly attractive for joining relatively thin skinned or walled structures, particularly where fatigue is a problem. The success of adhesive bonding depends strongly on the surface treatment of the adherends, which needs to be optimised to ensure that structural integrity is maintained under service conditions for the required life of the component.

Mechanical fastening (i.e. bolting) is usually considered to be less costly than adhesive bonding because of its simplicity and low-cost tooling and inspection requirements [1]. However, drilling holes in composites is not straightforward. It is highly labour intensive (unless automated) with special care required to avoid damaging the composite. In addition, bolted structures often need numerous mechanical fasteners with washers/shims to prevent damaging the composite structure during bolt clamp-up. Adhesively bonding a structure may in many cases be the more economic option, despite the high tooling and process costs. Table 1 compares the advantages and disadvantages of adhesively bonded and bolted composite and metal joints.

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Joint types can be characterised in terms of those that are:

- Mechanically fastened (e.g. with bolts, rivets, screws, etc.)
- Adhesively bonded (see Section 3)
- Consist of a combination of mechanical fastening and adhesive bonding

The choice of joining technique is dependent on several factors including:

- Materials to be joined
- Design of joint region
- Joining process used
- Joint strength and/or stiffness – dependent on:
  - Arrangement of parts to be joined
  - Properties and geometry of materials to be joined
  - Load-transfer path between parts
  - Joining process and process control
- Need to disassemble
- Durability
  - Absences of crevices, traps for moisture + other chemicals
  - Contact between materials, galvanic corrosion
  - Sharp corners, holes/notches: stress raisers

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**Table 1:  Advantages and Disadvantages of Adhesively Bonded and Bolted Joints**

| Advantages | Disadvantages |
|---|---|
| **Bonded Joints** | |
| Low stress concentrations in adherends | Difficulty in bonding thick sections |
| Lightweight | Difficult to inspect |
| Stiff connection | Prone to environmental degradation |
| Excellent fatigue life | Sensitive to peel and cleavage stresses |
| No fretting problems | Cannot be disassembled |
| Smooth surface contour | High quality control required |
| Damage tolerant | Difficulties in recycling materials |
| **Bolted Joints** | |
| Easy to disassemble | Considerable stress concentration |
| No thickness limitations | Added weight of mechanical fasteners |
| Simple joint configuration | Metallic components are prone to fatigue |
| Manufacturing and inspection straightforward | Hole formation can damage composite |
| Environmentally insensitive | Composites have poor bearing properties |
| Insensitive to peel forces | Fretting a problem in metals |
| Residual stress is generally not a problem | Extensive shimming is often required for composites |

This section examines different approaches that are being used to design for fatigue and creep in joined metallic structures, and in many cases being adopted for use with glass and carbon fibre-reinforced plastics (i.e. GFRP and CFRP)

## 2.1    GENERAL APPROACHES TO FATIGUE DESIGN

Three approaches to designing adhesively bonded and bolted joints for cyclic fatigue loading are shown below:

- Stress-life approach (i.e. stress-cycle (**S-N**) curves of typical joints and assemblies)
- Fracture mechanics
- Strain-based approach to fatigue life

### 2.1.1   Stress-Life Approach

Stress-life approach is an empirical method, which uses stress-cycle (**S-N**) curves 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. This approach is the most widely used of the above-mentioned techniques.  Under constant amplitude loading conditions, most materials or structures exhibit a plateau in the stress-cycle curve (Figure 1), which typically occurs at $N > 10^6$ 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.

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 bolt holes and ends of adhesive joints).  The mean stress level and stress amplitude of the imposed fatigue cycle are known to play an important role in influencing the fatigue behaviour of engineered structures. Ideally, the range of stresses should be kept below the "endurance limit".  The stress levels should be sufficiently low for fatigue not to be a problem.
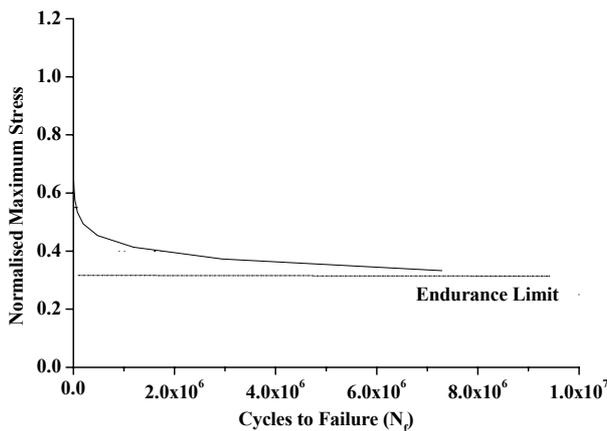
**Figure 1:  Typical S-N curve with fatigue or endurance limit**.

Cyclic fatigue loading can be in the form of either constant amplitude or variable amplitude spectrum loading.

**Constant Amplitude Loading** is defined by the following terms (see also Figure 2):

Minimum stress, $\sigma_{MIN}$
Maximum stress, $\sigma_{MAX}$
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.
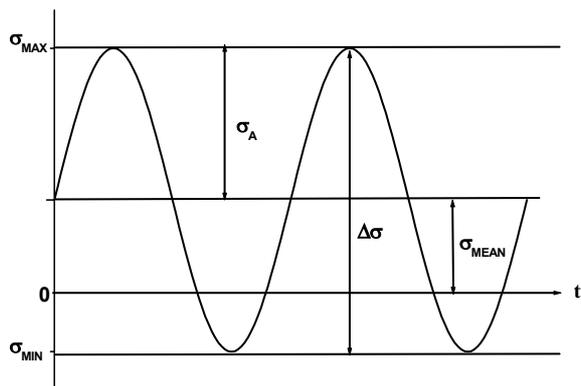


**Figure 2:  Stress parameters for constant amplitude sinusoidal stress-time cyclic loading.**

Figure 3 shows an **S-N** curve (**$R = 0.1$** and test frequency **$f = 5$ Hz**) that was obtained for an aluminium/epoxy tapered strap joint [2].  The **S-N** curve has been normalised with respect to the ultimate failure stress of specimens tested under monotonic loading at an equivalent loading rate to the fatigue cycling.  The gradient of the slope **k** is the fractional loss in strength per decade of cycles and is dependent on the joint geometry and loading conditions.

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Normalised **S-N** curves (see Figure 3) can be approximated by the following relationship [2]:

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

where $\sigma_{MAX}$ is the maximum load applied to the specimen, $\sigma_{ULT}$ is the ultimate strength of the joint and $N_f$ is the number of cycles to failure. The value of **k** was $\approx 0.09$ for cyclic fatigue tests carried out on 5251 aluminium alloy/AF126-2 epoxy tapered strap joints.
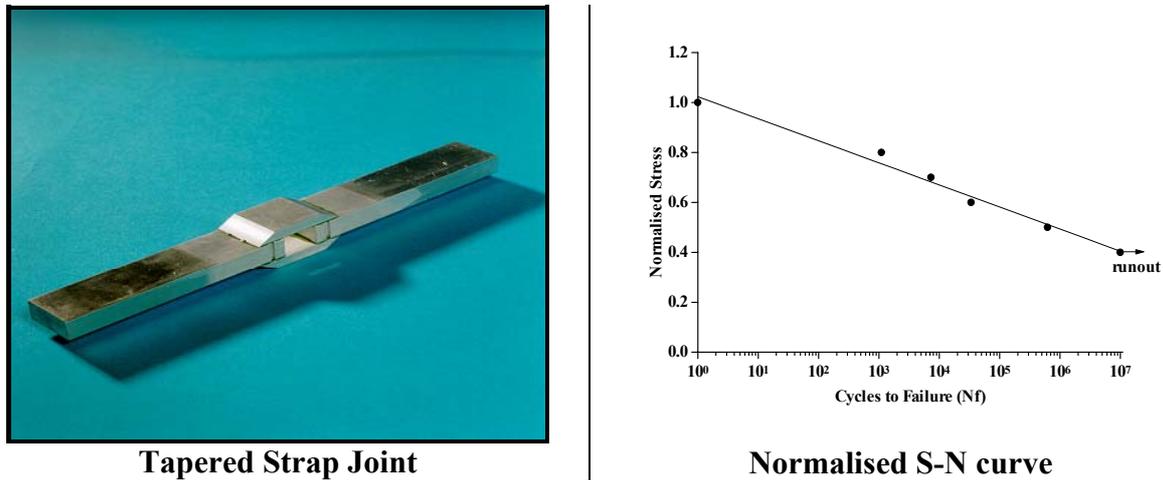


| **Tapered Strap Joint** | **Normalised S-N curve** |

**Figure 3:  S-N data for 5251 aluminium alloy/AF126-2 epoxy tapered strap joints [2].**

**Constant-life diagrams** are often used to represent the effects of mean stress and stress amplitude on fatigue performance of adhesive joints and composites [3-4].  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.  Figure 4 shows normalised stress-amplitude plots for different mean stress values that were obtained for tension-tension fatigue of a unidirectional GFRP composite material (E-glass/913 epoxy [3]).

The results shown in Figure 4 have been normalised with respect to the ultimate tensile strength, $\sigma_{UTS}$, 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).
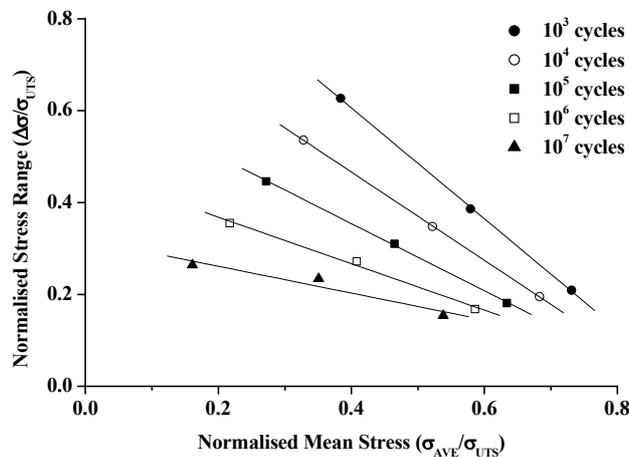


**Figure 4:  Stress amplitude-life plots for different mean stress values for E-glass/913 [3].**

A number of models have been suggested for determining stress amplitude-life plots of polymeric materials [3]. Crocombe *et al.* [4] have shown a Goodman type curve to be a valid method of representing the effect of mean stress and stress amplitude on fatigue performance of single-lap and T-peel joints. The Goodman relation is given below [3]:

$$\sigma_A = \sigma_{FS}\left(1 - \frac{\sigma_{MEAN}}{\sigma_{ULT}}\right) \tag{2}$$

where $\sigma_A$ is the stress amplitude (for a non-zero mean stress), $\sigma_{FS}$ is the fatigue strength (for a fixed life), $\sigma_{MEAN}$ is the mean stress and $\sigma_{ULT}$ is the ultimate strength of the material. The Goodman relation is known to match experimental data for brittle metals.

**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 have been developed to simulate the load sequence for aircraft transport and military aircraft (e.g. **T**ransport **WI**ng **ST**andard (**TWIST**) and **F**ighter **A**ircraft **L**oading **STA**ndard **F**or **F**atigue 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 have traditionally been fatigue tested under spectrum loading conditions to a minimum of two lifetimes to ensure adequate fatigue life [5]. 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.
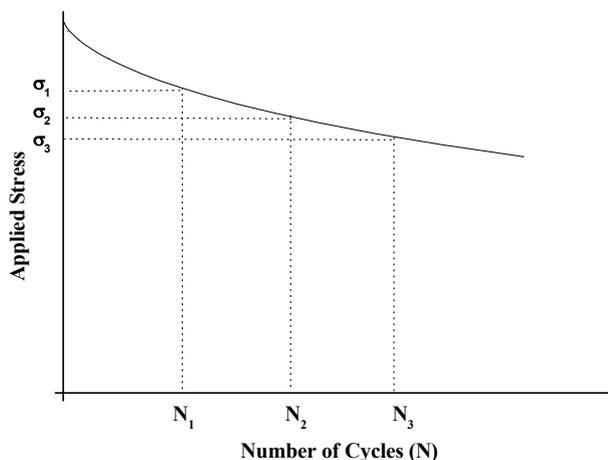


**Figure 5: Schematic of Palmgren-Miner rule.**

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**Palmgren-Miner Cumulative Damage Rule:** The most common tool for estimating the fatigue life of a structure under spectrum loading conditions is Palmgren-Miner (or Miner's) rule. This rule estimates fatigue life by the following expression (see also Figure 5) [6]:

$$Ng = \frac{1}{\sum\limits_{i=1}^{j} \alpha_i / N_i} \qquad (3)$$

where $N_g$ is the fatigue life under spectrum of loads, $\alpha_i$ is the fraction of fatigue life for each stress load $\sigma_i$ and $N_i$ is the fatigue life at constant stress amplitude for stress level $\sigma_i$. For metallic structures, the accuracy of Miner's rule is probably sufficient for preliminary design with test conformation required for fatigue-critical final designs.
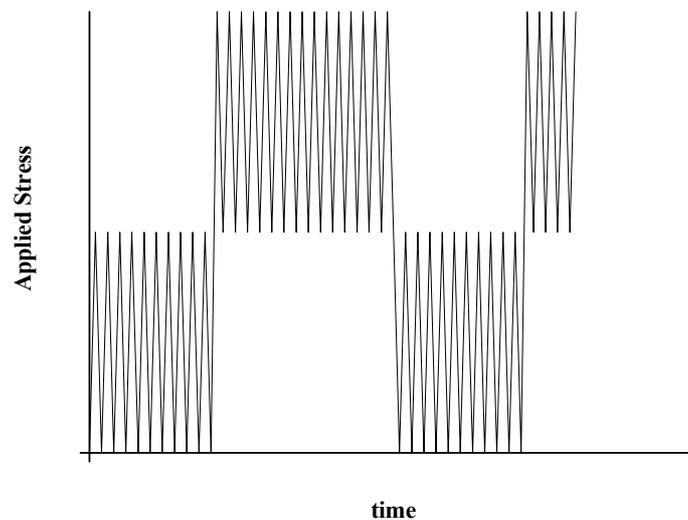


**Figure 6: Blocking of constant amplitude cyclic stresses.**

Stress-based characterisation of total fatigue life using Palmgren-Miner rule is only relevant when predicting the extent of damage induced under constant amplitude fatigue loading, which may be either continuous or blocked (see Figure 6). For composite materials, Palmgren-Miner rule tends to be generally unreliable, providing non-conservative estimates of fatigue life. However, empirically derived values might prove useful to design [6]. The order (i.e. load sequence) in which the stresses are applied can be expected to affect the fatigue life. The dependence of fatigue life on load sequence for composites can be attributed to the effect of cumulative damage on the residual strength of the composite. Modelling of this process is expected to be difficult.

The main proponents/users of stress-based design/analysis of bonded joints have been Goland and Reissner [7] and Hart-Smith [8-11]. Work in this area has focused on determining the shear and peel stresses within the adhesive layer under static loading conditions. Failure criteria (closed-form solutions) have been proposed for determining maximum shear and peel stresses in the adhesive layer of bonded joints [12]. Current design criteria recommend the elimination or drastic reduction of peel stresses, which are the main cause of failure within the adhesive layer or at the adhesive-adherend interface. Hart-Smith states that the bond strength is limited by the adhesive's shear strain energy per unit bond area (with linear-elastic modelling this can be expected to be equated to a stress). Stress-based design has been incorporated into computerized design programs used by the aerospace/defence industry.

**Fatigue Strength of Typical Details:** In this approach, the designer selects a standard detail (i.e. joint) that is expected to behave similarly to the new design and uses the fatigue strength that has been established for the standard detail. This approach is straightforward and usually very successful. Hence, it has widespread use for designing adhesively bonded and bolted metallic structures. However, the approach should only be considered for preliminary design purposes. Confirmation tests of assemblies representative of the actual structure need to be carried out to finalise the design.

For this approach to be used, a large (and reliable) database is required for the standard detail in order to determine design stress cycles for a given stress level, or vice-versa. The database needs to include the effects of different stress ratios on fatigue life and allow for out-of-plane deformation of the joint. FEA is used to determine the stress and strain distribution in the structure. The problem arises when the new detail does not match any standard details or new materials are involved, then fatigue strength becomes very uncertain. Table 2 provides values of stress that are stresses allowed in the net-section of a mechanically fastened double-lap joint manufactured from an aluminium alloy [13].

**Table 2: Design Stresses (MPa) for Fatigue Life of Bolted Double-Lap Joint [13]**
**(\*Allowance for out-of-plane bending in connected material)**

| Stress Ratio | Endurance or Fatigue Limit (cycles) | | | |
|---|---|---|---|---|
| | $< 10^5$ | $10^5 - 5.10^5$ | $5.10^5 - 2.10^6$ | $> 2.10^6$ |
| R < 0 | 83 | 57 | 41 | 28 |
| $0 \leq R \leq 0.5$ | 66 | 44 | 31 | 21 |
| $R \geq 0.5$ | 52 | 32 | 22 | 14 |
| Out-of-plane bending* | 52 | 32 | 22 | 14 |

**Note: Designers/Engineers need to conduct verification tests on any final design of a fatigue-critical structure.**

2.1.2   Fracture Mechanics

A major consideration in the design of adhesively bonded structures is the possibility of crack growth within the adhesive or at the adhesive-adherend interface. Crack propagation can be catastrophic when the strain-energy release-rate, **G**, or the stress intensity factor, **K**, of the adhesive-adherend system has been exceeded. Debonds or delaminations are probably the most life-limiting defects that occur in layered or laminated structures, and may arise during processing or subsequent service. Common structural features, such as drilled holes, machined corners, thickness changes or ends of bonded joints generate through-thickness (T-T) stress concentrations, which may initiate debonding under static or cyclic fatigue loading. Ultimate failure occurs when the remaining structure cannot withstand the applied maximum (or peak) stress.

The general approach is to relate the rate of crack growth, **da/dt**, through the joint to the applied strain-energy release rate **G** or the stress intensity factor **K**. This approach assumes a pre-existent crack and uses FEA to determine the stress state in the vicinity of the crack tip. It is worth noting that good progress has been made in applying fracture mechanics to predicting crack growth and failure of adhesively bonded joints for single-mode loading configurations [14-22], although the relevance and potential usage to actual bonded structures is regarded by the engineering community with some scepticism.

From an engineering perspective, prediction of crack growth rate is considered less important than determining the crack initiation stress or energy. A considerable amount of effort has been expended in attempting to predict crack initiation and the rate of crack growth in adhesively bonded joints (e.g. tapered double cantilever beam and single-lap) subjected to mode I (tension), mode II (shear), mode III (tear) and mixed-mode static and cyclic loading conditions. A number of these investigators [14-22] have found good correlation between strain energy release rates and joint fracture. The preferred approach has been to use the strain energy release rate; as stress intensity based analysis depends on accurately modelling the plastic zone that develops ahead of the crack tip [20]. The plastic zone in the adhesive is often restricted by the adherends. The input data requirements are far less demanding and easier to generate than those required for stress-based analysis of bonded joints. Yang *et al*. [21-22], using a traction-separation law to simulate the interfacial failure, have been able to accurately predict fracture of plastically deforming adhesive joints under mode I, mode II and mixed-mode I/II loading conditions.

The fracture mechanics approach is best suited to predicting failure subsequent to fracture propagation along a parallel crack path. However, failure is often catastrophic with no visible evidence of crack growth prior to the onset of failure. In these cases, failure is controlled primarily by the initial size of defects or flaws present in regions of high stress gradients (e.g. adhesive fillets). It has been suggested that failure occurs when the maximum stress/strain exceeds a critical stress/strain value over a certain distance, or when the stress acting over a certain volume exceeds a critical value [18].

The critical distance or volume, which is generally defined using FEA and experimental data, is a function of the specimen geometry and size, and the defect spectrum contained within the specimen. The adhesive in regions of highly localised stresses (e.g. joint fillets) may be intrinsically stronger than bulk adhesive test specimens because the former is less likely to contain a critical flaw. Weibull statistical analysis has been used to model the sensitivity of both distance and volume failure criteria to changes in local geometry and singularity strength (see [18]).

**Fracture Mechanics Theory**

A form of the Paris Equation can be used to relate crack growth rate **da/dN** per cycle to the maximum value of the applied strain-energy release-rate $\mathbf{G_{MAX}}$ [15, 19]:

$$\frac{da}{dN} = C(G_{MAX})^n \tag{4}$$

where **C** and **n** are material constants. This relationship applies only to the linear portion of the logarithmic-logarithmic plot of $\mathbf{G_{MAX}}$ versus **da/dN** (see Figure 7).

Alternatively, the crack growth rate can be expressed as a function of the range of strain-energy release-rate $\mathbf{\Delta G}$ (see Equation (6)) by [15, 19]:

$$\frac{da}{dN} = A(\Delta G)^q \tag{5}$$

where **A** and **q** are constants.
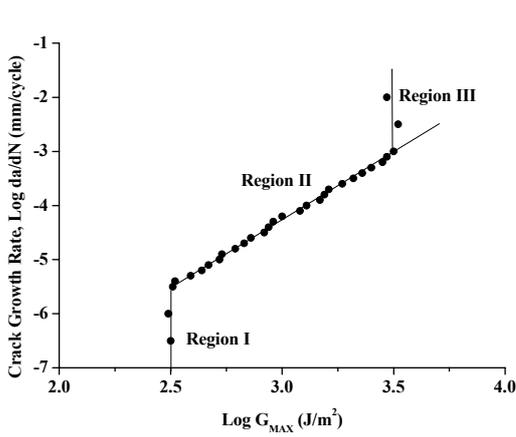
**Regions**

**Region I** – Threshold region ($G_{TH}$) associated with low crack growth rate **da/dN** and $G_{MAX}$ values ($G_{TH} \approx 0.1G_C$).

**Region II** – Linear region defined by the Paris Law given by Equation (4).

**Region III** – Value of $G_{MAX}$ approaches the adhesive fracture toughness $G_c$ measured under monotonic loading conditions.

**Figure 7: Typical log-log crack growth rate versus $G_{MAX}$ plot.**

The difference between maximum and minimum strain-energy release-rate per cycle $\Delta G$ is given by:

$$\Delta G = G_{MAX} - G_{MIN} \qquad (6)$$

This relationship only applies to the linear portion of the logarithmic-logarithmic plot of $\Delta G$ versus **da/dN**. Values for both sets of constants (i.e. **C** and **n**, and **A** and **q**) can be determined using linear regression fit to the linear region of the logarithmic-logarithmic plots.

Generally, the relationship between **log $G_{MAX}$** and **log $\Delta G$** and **log da/dN** is S-shaped (i.e. sigmoidal). This relationship can be described as follows [16-17]:

$$\frac{da}{dN} = C(G_{MAX})^n \left[ \frac{1 - \left(\dfrac{G_{TH}}{G_{MAX}}\right)^{n_1}}{1 - \left(\dfrac{G_{MAX}}{G_c}\right)^{n_2}} \right] \qquad (7)$$

$G_{TH}$ is the minimum (or threshold) value of the adhesive fracture energy, $G_c$, and **A**, **n**, $n_1$ and $n_2$ are material constants. $G_c$ is determined from constant rate of displacement tests (i.e. monotonic fracture energy).

In most practical applications, bonded structures experience mixed-mode loading conditions involving $G_I$ and $G_{II}$ (and in some cases $G_{III}$) due to the presence of peel and in-plane shear stresses, resulting in mixed-mode cracking. The total strain-energy release-rate $G_T$ (or $G_c$) is often represented by either a linear interaction or a quadratic relationship:

$$\frac{G_I}{G_{Ic}} + \frac{G_{II}}{G_{IIc}} = 1 \qquad (8)$$

$$\left(\frac{G_I}{G_{Ic}}\right)^2 + \left(\frac{G_{II}}{G_{IIc}}\right)^2 = 1 \qquad (9)$$

**Note: The equations used to determine the number of fatigue cycles to failure is generally complex. This poses particular problems in applying a fracture mechanics approach to actual structures, particularly where the loading configuration is not a well defined single-mode loading configuration. Design of bonded structures against fatigue using a fracture mechanics approach would require that the maximum fatigue loads be based on $G_{TH}$ values.**

2.1.3   Strain-Based Approach

Stress-based analysis assumes the material undergoes unconstrained deformation. In fact, engineering structures experience a certain degree of structural constraint, particularly in regions of high stress concentrations. In these situations, it is therefore more appropriate to assume strain-controlled conditions when modelling fatigue behaviour [3]. Strain-controlled fatigue is commonly used as a basis for structural design in components where cyclic fatigue crack initiation is of concern near stress concentrations.
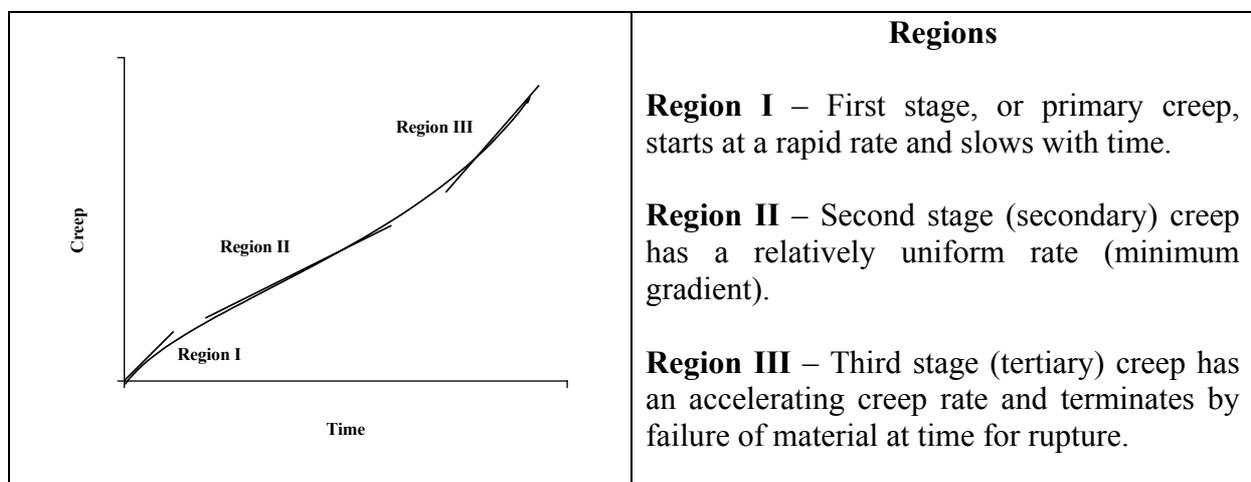
The following strain-life model forms the basis of a widely used approach by industry for predicting the fatigue life of engineering alloys [3]:

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma_f'}{EN_f}\left(2N_f\right)^b + \varepsilon_f'\left(2N_f\right)^c \tag{10}$$
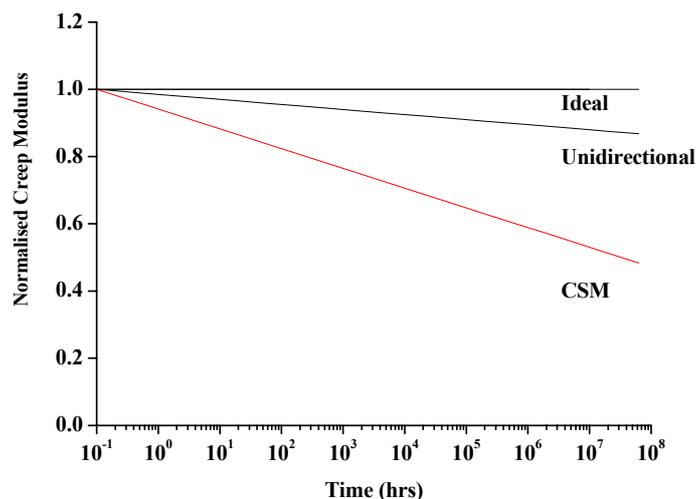
where $\Delta\varepsilon/2$ is the strain amplitude, $\varepsilon'_f$ is the fatigue ductility coefficient (approximately equal to the true fracture strain $\varepsilon_f$ in monotonic tension), $\sigma'_f$ is the corresponding stress, $E$ is the Young's modulus, and $b$ and $c$ are constants. The model accounts for both elastic and plastic strain.

2.2   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 composites and adhesives can undergo creep deformation at room temperature (referred to as cold flow).



| | 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. |

**Figure 8:  Creep versus time plot.**

**Figure 9: Normalised creep modulus for composite materials loaded in tension**

Creep data is usually presented as a plot of creep versus time with stress and temperature constant (Figure 8). 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. Figure 9 compares the creep modulus for unidirectional and chopped strand mat (CSM) composite materials in tension.

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:

$$E(t) = \sigma/\varepsilon(t) \tag{11}$$

may be modelled by the following relationship:

$$E(t) = E_o t^{-n} \tag{12}$$

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.

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** (see Figure 9). Low values of **n** can be expected for elastic materials reinforced with continuous aligned fibres. The value of **n** is lower in the fibre direction for these materials. In fact, loading along the fibre direction is unlikely to result in significant creep deformation. Creep index increases for random fibre formats (e.g. CSM) and unfilled materials. For design purposes, creep modulus and creep index should be obtained from direct experiments on the composite or metallic system.

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. Figure 10 shows the time-to-failure for a 5251 aluminium alloy strap joint bonded with AF126-2 epoxy adhesive.
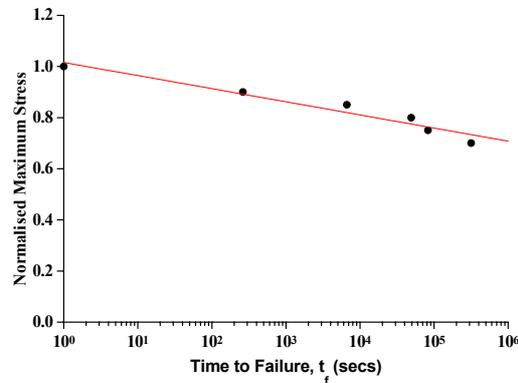


**Figure 10: Creep rupture of 5251 aluminium alloy/AF126-2 epoxy tapered strap joints.**

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.

## 3. BONDED JOINTS

Generally, the basic rules of good design of adhesive joints apply to most loading (i.e. static and cyclic) and environmental conditions. For example, thickening the adhesive at the ends of an overlap through the use of large adhesive fillets or by internal tapering can reduce peel and shear stresses at the ends of an overlap (Figure 11), thus improving creep and fatigue performance. Increasing the bond thickness spreads the strain over a larger volume, resulting in lower strain in the adhesive and therefore, a lower stress concentration (although at the expense of joint stiffness). This section examines the design issues associated with adhesively bonded structures.
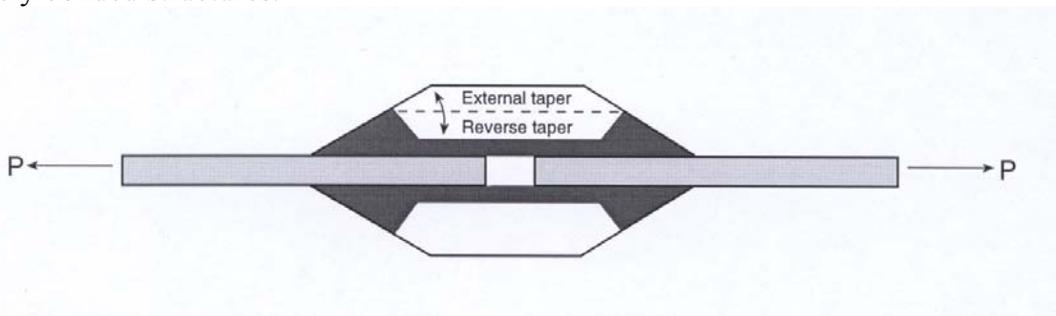


**Figure 11: Bevelled strap joint.**

## 3.1 DESIGN 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 <u>all</u> 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 3)

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.

**Table 3: Recommended Values for Partial Safety Factors [23]**

| Joint Configuration | Safety Factor ($\gamma_m$) |
|---|---|
| **Adhesive Properties, $\gamma_{m1}$** | 1.5 |
| **Adhesive thickness, $\gamma_{m2}$** | 1.5 |
| **Long-term loading, $\gamma_{m3}$** | 1.5 |
| **Environmental conditions, $\gamma_{m4}$** | 2.0 |
| <u>**Fatigue (non-fail safe joints), $\gamma_{m5}$**</u> | |
| **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 $\gamma_m$ by which the adhesive properties should be divided to give design values is shown below (refer to [23]):**

$$\gamma_m = \gamma_{m1}\gamma_{m2}\gamma_{m3}\gamma_{m4}\gamma_{m5} \qquad (13)$$

**For long-term testing, the overall partial safety factor $\gamma_m$ should be no less than 4.0.**

There are a number of potential failure modes, including:

- Tensile, compressive or shear failure of the adherends
- Shear or peel in the adhesive
- Shear or peel in the composite near surface plies
- Shear or peel in the resin-rich layer on the surface of the composite
- Adhesive failure at the metal/adhesive or composite/adhesive interface

Designers aim to ensure that the joint fails by bulk failure of the adherends. A margin of safety is generally included to provide tolerance to service damage and manufacturing defects in the adhesive layer (see Table 3). The aim is to maintain the adhesive in a state of shear or compression. Tension, cleavage and peel forces should be avoided, or their effect minimised. Structural adhesives have relatively poor resistance to T-T (peel) stresses, and therefore to obtain maximum efficiency, joints are designed to minimise these stresses. For fibre composite laminates, resistance to peel stresses may be considerably lower than for structural adhesives, so even greater care must be taken with composites to minimise these stresses. Generally, the adhesive is not allowed to become the weakest link. If composite adherends are used, care needs be taken to ensure that the T-T strength of the adherends does not become the weakest link.

Design of a joint should satisfy the following conditions (see 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 [12, 23-25]:

- Minimise shear and peel stress concentrations - presence of adhesive fillets or taper ends will decrease the adhesive peel stresses (avoid stress concentrations). The use of stiff or thick adherends will minimise peak stress levels and yield a more uniform adhesive stress distribution. For example, it is recommended that the taper ends of lap joints should have a thickness of 0.76 mm and a slope of 1/10.
- 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.
- 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 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).

Fatigue damage can be minimised by maintaining the adhesive in the elastic state for most of its service life. Design criteria in the aircraft industry require structures to withstand limit loads with no permanent deformation and ultimate loads without any failure. Limit load is the highest load expected during service life. The ultimate load is 1.5 times the limit load (even at this load the strain should not approach the failure strain). Appendix 2 describes an analytical procedure for producing satisfactory single-lap joints.

---

Chamis and Murthy [25] 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:
        $l = F/S_{as}$
        where **F** = load in adherends per unit width and $S_{as}$ = 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.

---

As previously mentioned, fatigue performance of an adhesive joint can be approximated by (rule of thumb) Equation (1) where **k** the fractional loss in strength per decade of cycles is a measure of fatigue resistance of the joint. The lower the **k** value the better the fatigue performance. Table 4 shows typical **k** values for a number of metal and composite joints bonded with epoxy adhesives. 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 major cause of failure (e.g. T-peel tests).

**Table 4:  Typical k Values for Bonded Joints**
**(R = 0.1 and f = 5 Hz)**

| 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-peel (mild steel adherends) | 0.130 |

## 3.2 FAILURE CRITERIA

A number of approaches, listed below, have been adopted by engineers/designers for predicting the static strength (failure load) of adhesives and adhesively bonded structures.

- Strength of materials based models (e.g. average stress, maximum stress and maximum strain failure criterion).
- Plastic yield criteria (e.g. von Mises and Tresca yield criterion, and Drucker-Prager plasticity model).
- Void nucleation (cavitation) models.
- Fracture-mechanics analysis (see Section 2.1.2).

All approaches mentioned above have been reviewed in NPL Report MATC(A)27 [26] (see also [27-28]).

## 3.3 FINITE ELEMENT APPROACHES FOR MODELLING INTERFACES

The assumption of a perfect bond means that the finite element analysis takes no account of the adhesion properties of the interface. There is no obvious method for predicting interface strength, as often the input data needed requires knowledge of the interface strength, or the failure stress/strain of the adhesive. However, there are methods of accounting for adhesion in an FE analysis, by modelling adhesive failure. This section highlights some of the options available. There are two main approaches for modelling the failure of an adhesive interface. One approach is to model the growth of a crack (fracture mechanics energy-based approach) and the other is to model adhesive debonding (stress- or strain-based approach). Approaches using the commercial FE software package **ABAQUS** are illustrated, although other suitable FE codes are reviewed in Section 5.1.

### 3.3.1 Fracture Mechanics

As previously mentioned (see Section 2.2), the fracture mechanics approach assumes that a sharp crack exists within the material and failure occurs through propagation of this crack. A debonded area or delamination can be classified as a crack. This approach does not predict the crack initiation stress or energy. Advancing the crack front when the local strain-energy release-rate rises to its critical value can simulate crack propagation. In **ABAQUS** [29], potential crack surfaces are modelled using contact surface definitions. Surfaces may be partially bonded initially, but may debond during crack propagation. The three debonding criteria are:

- Crack opening displacement
- Critical stress criterion at a critical distance ahead of the crack tip
- Crack length as a function of time.

After debonding, the interface behaviour reverts to standard contact, including any frictional effects. Guild *et al.* [30] used the virtual crack growth method to model crack propagation. Cracks were introduced into the mesh by renumbering nodes of adjacent elements. The virtual crack growth method for assessing strain-energy release-rate is based on the concept that the energy released in growing the crack is equal to the energy which would be required to close the crack. Forces and displacements at the crack tip are obtained from FE simulations and the energy can be calculated.

### 3.3.2   Strength of Materials Approach

Interface elements located along the debonding interface can be used to predict crack growth. In this method, a softened decohesion material model is provided with traction/relative displacement relationships that are constructed so that the enclosed area is equated to the critical fracture energy. Initial flaws are not required with initiation being governed by a strength criterion. In the work of Chen *et al*. [31] interface elements were used via a user subroutine in **ABAQUS,** and were embedded directly in **LUSAS** (FEA software). Two material properties were required for interface elements: $G_c$, the total energy from experiments and $S_t$, the assumed interfacial material strength.

Another method of analysing failure of an interface is using a cohesive zone model [32-34]. Jagota *et al*. [32] suggest that when incorporated between element edges, cohesive elements extend conventional FE in a way that allows independent specification of interfacial fracture and bulk constitutive behaviour. The cohesive elements describe the deformation and failure of the interface between two bulk finite elements by specifying the tractions that resist relative motion. The cohesive zone model was implemented in **ABAQUS** for this work.

Coupling elements have also been used to study adhesion. Sebastian [35] modelled composite plates debonding from concrete beams using nonlinear FE analysis with a hybrid element for the beam and the plate along with a coupling element for the adhesive connection. The coupling element was programmed to rupture and to disallow further interaction between the hybrid elements on either side once a predetermined stress level corresponding to the shear bond strength of the plate to concrete connection had been activated. The author suggests that the FE program can be improved by replacing the stress-based failure criterion used for determining connection failure with a fracture mechanics based criterion.

An alternative debonding method can be employed in **ABAQUS** using the debond option in which an amplitude-time function is used to give the relative magnitude of force to be transmitted between the surfaces at time $t_0 + t_i$ ($t_0$ being the time when debonding begins). When the fracture criterion is met at a node, the force at that node is ramped down according to the debonding data. The force at the node must have a value of 1.0 at zero time and must end with a magnitude of zero at the final time (i.e. node debonded).

Damage modelling is another method for modelling failure of an adhesive. Feih *et al*. [36] modelled a bending test. Failure of the adhesive was assumed to occur when the maximum tensile plastic strain was reached under tension. Onset of adhesive cracking was modelled by reducing the transferable stress to 10%. This damage method simulates the stress redistribution inside the adhesive and was employed as a **UMAT** (i.e. user model) within **ABAQUS**.

**ABAQUS/Explicit** also offers two damage models; a shear failure model driven by plastic yielding; a tensile failure model driven by tensile loading. These failure models provide simple failure criteria that are designed to allow stable removal of elements from the mesh as a result of tearing, ripping or tensile spalling of the structure. Each model provides several failure choices including the removal of elements from the mesh.

## 4.    BOLTED JOINTS

This section reviews existing procedures and industrial codes of practice for design of mechanically fastened (bolted) joints (see Figure 12) as used for metallic and composite material systems.  Consideration is given to design for static, fatigue and creep loading regimes.  Mechanical fastening using screws and/or rivets is not considered.  The design guides reviewed in this report are listed in Table 5, although a complete review of all guidelines/standards was not possible as some documents, such as Naval Engineering Standards were restricted.  Consultation with industry and universities was also carried out to ensure access to the latest industrial practices and research findings.

### Table 5 – Summary of Design Guidelines for Bonded and Bolted Joints

| Guideline | Summary |
|---|---|
| European Space Agency (ESA) Structural Materials Handbook [37] | 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 [12] | 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 [5] | 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 [1] | 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 [38] | 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.) [39] | 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. |
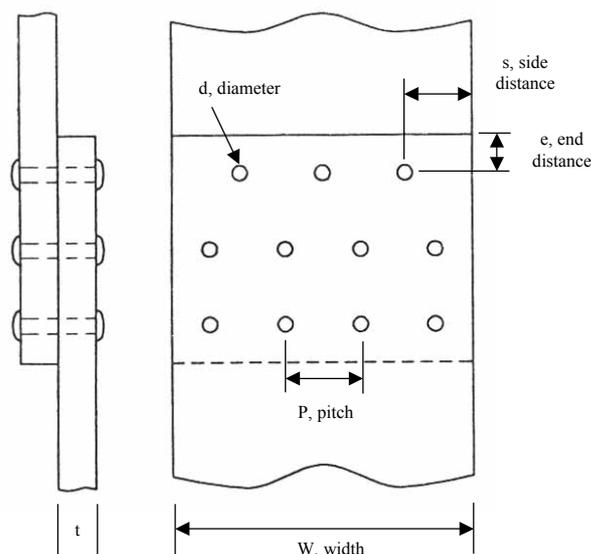


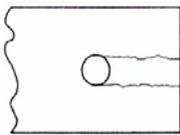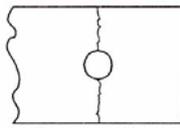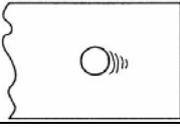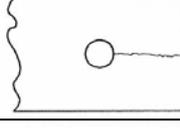**Figure 12:  Multi-array bolted joint with commonly used terminology.**

## 4.1   FAILURE MODES

Most guidelines detail the four main failure modes as (see also Table 6):

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

However, mixed mode failures do occur as shown in Table 6.  Failure is usually gradual with progressive damage eventually resulting in loss of load bearing capacity.

**Table 6:  Typical Failure Modes for Bolted Joints**

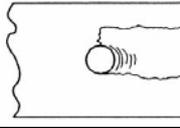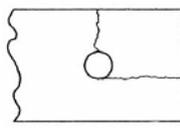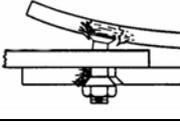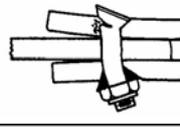| Failure mode | | Comment |
|---|---|---|
| **Shear out** | | Caused by shear stresses and occurs along shear out planes on hole edge, typical failure mode when end distance is short. |
| **Tension (net-section)** | | Caused by tangential tensile or compressive stresses at the edge of the hole.  For 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 |
| **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. |
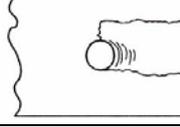
## 4.2    DESIGN CONSIDERATIONS FOR BOLTED JOINTS

The use of bolts is considered an effective means of fastening load-carrying members. However, the design guidelines reviewed stated this to be the case 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

### 4.2.1   Material Parameters

When designing bolted joints a variety of material parameters must be considered.  Most of the guidelines reviewed identified the following material variables to be worthy of consideration during design:

- 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)

**Table 7:  Design Guidance on Lay-Up for Bolted Joints**

| Guideline | Guidance |
|---|---|
| **European Space Agency (ESA) Structural Materials Handbook [37]** | • Inclusion of $\pm 45^{\circ}$ plies to reduce stress concentration factors around holes in CFRP.<br>• Optimum tensile properties obtained with a ratio of $0^{\circ}$ and $\pm 45^{\circ}$ plies of 2:1.  Optimum shear strength is achieved with ratio of 1:1.<br>• Recommends placement of $90^{\circ}$ plies on surface of GFRP laminates. |
| **EUROCOMP Design Code and Handbook [12]** | • Ideally, balanced symmetrical lay-up to be used with different orientation plies distributed throughout the thickness.<br>• Preferably there should be 25% plies in $0^{\circ}$, 25% plies in $90^{\circ}$ and 50% plies in $\pm 45^{\circ}$.<br>• There should be at least 12.5% of plies in each of the four directions. |
| **Composites Engineering Handbook [1]** | • Lay-up chosen should be approximately quasi-isotropic (i.e. based on $0^{\circ}$, $\pm 45^{\circ}$ and $90^{\circ}$ plies).<br>• Non-zero plies to reduce stress concentrations and avoid shear-out or cleavage failures.<br>• $0^{\circ}$ plies to carry main bearing and tensile loads. |

**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 (see Table 7).

### 4.2.2  Bolt Parameters

Care should be taken when selecting the appropriate bolt material for use with composites, as galvanic corrosion can be problematic. 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.

### 4.2.3  Bolt/Washer Size

The guidelines reviewed made several recommendations on bolt, hole and washer sizes in order to achieve the maximum strength of a bolted joint. These are listed in Table 8.

**Table 8 – Design Guidance on Fastener Parameters**

| Guideline | Guidance |
|---|---|
| **European Space Agency (ESA) Structural Materials Handbook [37]** | • Holes should be reamed and bolts should be a good fit.<br>• Washer hole size should be close to the size of the bolt shaft. |
| **EUROCOMP Design Code and Handbook [12]** | • 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.<br>• Bolts should be self-locking or fitted with locknuts.<br>• 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.<br>• 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 [38]** | • The handbook provides a brief guidance on the type of fasteners to be used. |

### 4.2.4  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 suppresses delamination driven failure modes and 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.

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 [39].

## 4.3    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 Table 9.

**Table 9:  Design Guidance on Joint Configuration**

| Guideline | Guidance |
|---|---|
| **European Space Agency (ESA) Structural Materials Handbook [37]** | • Provides guidance on the effects of end-distance, width, row and pitch distances, hole patterns and multiple rows of bolts.<br>• Recommends values of ratios for **e/d**, **w/d** and **d/t**. |
| **EUROCOMP Design Code and Handbook [12]** | • Guidance given on minimum distances between holes and minimum end-distances.<br>• Recommends values of **d/t**, **e/d** ratios. |
| **Composites Engineering Handbook [1]** | • Provides general discussion on the effects of parameters such as **w**, **d**, **t** and **e**. |
| **Joint Aviation Requirements (JAR) 25 – Large Aeroplanes [38]** | • 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 [40]** | • No guidance provided.  Requires bolt configuration to be specified for certification. |

## 4.4    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

## 4.5    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:

(i)     Two-dimensional classical elasticity methods with complex variables; or

(ii)    Full, three-dimensional finite or boundary element analyses (FEA/BEA).

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 [1]:

• 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. These are described in references [1, 5, 12, 37]. 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 [37]:

$$\left(\frac{\sigma_{xx}}{X}\right)^2 + \left(\frac{\sigma_{xy}}{S_c}\right)^2 = e^2 \quad \text{(if } e < 1 \text{ no failure occurs and if } e \geq 1 \text{ failure occurs)} \tag{14}$$

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

The Yamada criterion is generally used with a failure hypothesis, such as the Whitney/Nuismer point stress criterion [41]. 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 characteristic curve shown in Figure 13 is described by the following relationship [13]:

$$r_c(\theta) = \frac{D}{2} + R_{ot} + (R_{oc} - R_{ot})\cos\theta \tag{15}$$

where **-π/2 ≤ θ ≤ π/2,** and **$R_{ot}$** and **$R_{oc}$** are the characteristic lengths for tension and compression.

**Figure 13: Whitney/Nuismer failure hypothesis [37].**

The parameters $\mathbf{R_{ot}}$ and $\mathbf{R_{oc}}$ 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. $\mathbf{r} = \mathbf{r_c}$) when $\mathbf{e} > \mathbf{1}$.

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.

## 4.6 FATIGUE

The fatigue performance of bolted composite joints is generally superior to that of metal joints [5]. Composite bolted joints have been observed to exhibit considerable fatigue life under high stress fatigue loads with minimal reduction in residual strength (i.e. $\sigma_{\mathbf{MAX}} = \mathbf{0.7}\ \sigma_{\mathbf{UTS}}$). The predominant failure mode under cyclic loading is usually bearing failure, which is in the form of hole elongation. Previous studies suggest that the conditions required for maximising joint strength under static loading conditions may also apply to ensure good fatigue performance. However, there are limited data available as to the influence of key variables, such as those identified in Section 4.2 to be confident that the two are inseparable (i.e. coupled). Issues, such as fretting and stress relaxation may pose particular problems, which may not necessarily improve when optimising joint strength under static loading conditions.

Constant amplitude (sinusoidal waveform) tension-tension fatigue tests carried out at NPL [42] on pin-bearing and open-hole tension coupon specimens in load control (**R = 0.1** and test frequency **f = 5 Hz**) showed that the normalised **S-N** curves for a range of composite materials (see Table 10) could be represented by Equation (1). The value of **k** was shown to be independent of specimen geometry and in the case of the glass fibre-reinforced systems k was approximately 0.1. The **S-N** curve for a woven fabric pin bearing tests is shown in Figure 14.

**Table 10:  Values of k for Different Structural Elements and Composite Materials [42]**

| 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 |
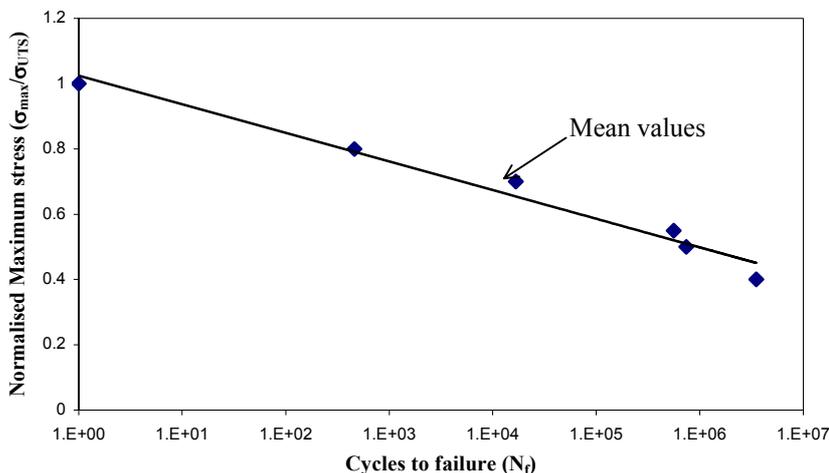


**Figure 14:  S-N curve for pin bearing woven glass-fibre/epoxy [42].**

## 5.    SOFTWARE REVIEW

Techniques for stress analysis of a joint generally fall into two main categories: analytical, closed-form methods and FE 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.  FE 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 FEA and analytical based software developed for the analysis and design of bolted and bonded structures, and materials selection.

### 5.1    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 11) [37, 43].  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**).

**Table 11:  Finite Element Packages (see also [37])**

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

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, FE stress analysis can be used as a tool for optimising the design of a joint.  Rispler *et al.* [44] used the evolutionary optimisation method **EVOLVE** to optimise the shape of adhesive fillets.  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.

## 5.2 ANALYTICAL SOFTWARE

In the literature 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 [45].  Reviews of these analytical theories and their assumptions have been published [46-47].

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 [8-11] 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** [48]).

**Table 12: PC Based Software Packages (see also [24])**

| Name | Supplier | Application | Features |
|---|---|---|---|
| **BOLT** | G.S. Springer Stanford University | • Design of pin-loaded holes in composites. | • Prediction of failure strength and failure mode.<br>• Three types of bolted joints:<br>Joints with a single hole<br>Joints with two identical holes in a row<br>Joints with two identical holes in tandem<br>• 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.<br>• Adhesive peel and shear stress predictions.<br>• Allowance for plasticity in adhesive layer.<br>• 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.<br>• Orthotropic adherends.<br>• Torsional and axial loading.<br>• Allowance for plasticity in adhesive layer.<br>• 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.<br>• Parametric analyses.<br>• Panel and beam design.<br>• Bonded and bolted double shear joints.<br>• 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.<br>• Linear-elastic and linear-elastic/plastic behaviour.<br>• Tension and shear loading<br>• Symmetric and asymmetric lap joints.<br>• 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<br>• Tensile/shear/bending moment/pressure loading.<br>• Linear and non-linear analysis<br>• Peel, shear and longitudinal stress predictions in adhesive layer and adherends.<br>• Thermal stress analysis for adherend and adhesive. |
| **PAL** | Permabond, UK | • "Expert" system for adhesive selection. | • Joined systems include:<br>Lap and butt joints<br>Sandwich structures<br>Bushes/gears/bearings/shafts/pipes/threaded fittings<br>• Elastic analysis.<br>• Creep/fatigue effects on joint stiffness (graphical). |
| **RETCALC** | Loctite, UK | • Interactive windows based software General purpose | • Joint strength.<br>• Correction factors (temperature and fatigue). |
| **SAAS** | Surrey University, UK | • 2-D stress analysis FEA based package for adhesively bonded joints.<br>. | • Single and double lap, top hat, right angle butt and clinch flange joints.<br>• Isotropic and anisotropic adherends.<br>• Free, translational and rotational loads.<br>• Peel, shear and von Mises stress in adhesive layer. |

Although simplified analytical procedures for designing adhesively bonded joints are available in the form of PC compatible software [49], 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 Table 12. 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 13). **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 [30]. 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.

## Table 13: 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.<br>• Experimentally derived stress-strain curves for 3 adhesives.<br>• Adhesive behaviour modelled using shear-lag anlysis.<br>• 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.<br>• Stress and strain distributions in multi-step joints.<br>• Shear-lag analysis.<br>• Balanced (symmetric), rigid joints. |
| **ESDU 80011** | • Elastic stresses in the adhesive in single step double lap bonded joints. | • Single-step double bonded lap joints.<br>• Applicable to thin bonded joints.<br>• Tension and compression loading.<br>• Peak elastic shear and direct tensile stresses in adhesive. |
| **ESDU 80039** | • Elastic adhesive stresses in multi-step lap joints loaded in tension. | • Computer program.<br>• Single and double lap joints.<br>• Tension loading.<br>• Allowance for dissimilar adherends.<br>• Assumes adhesive and adherend elastic behaviour.<br>• 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.<br>• Aluminium, steel or titanium alloy adherends.<br>• Steel or aluminium bolt.<br>• 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.<br>• Tension loading.<br>• Aluminium, steel or titanium alloy adherends.<br>• Steel or aluminium bolts.<br>• Bolt loads, joint extension and joint stiffness predictions. |

## 6.    SANDWICH PANELS

Sandwich structures (Figure 15) consist of two thin outer layers of stiff, strong material (in this case a composite) separated by a thick, lightweight layer of core material (e.g. foam, honeycomb or corrugated).  These structures are widely used because of the associated low costs and the high flexural stiffness that can be obtained at low areal weights.  Virtually all types of composites from CSM to prepregs can be used as skins with a wide range of polymeric, metallic and ceramic materials used as core materials.  Aerospace applications tend to incorporate carbon fibre-reinforced skins with polymer honeycomb or corrugated cores.  This section covers general design issues relating to flat composite sandwich structures (including failure modes).
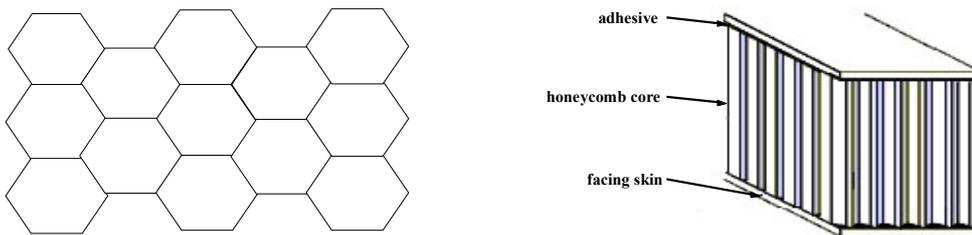


**Figure 15:  Sandwich structure with honeycomb core.**

Sandwich structures are often compared with I-beams with the role of the facings and core in the former corresponding to the flanges and web of an I-beam, respectively.  The facings support flexural loads with one facing in compression and the other in tension.  The core must have sufficient shear stiffness to prevent the facing skins from sliding over each other under bending and resist twisting due to torsional loads.  The core unlike the web of an I-beam gives continuous support to the facing skins and, hence, must resist crushing and buckling due to concentrated loads.  Table 14 summarises possible failure modes and driving forces that need to be considered when designing composite sandwich structures.
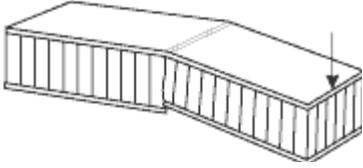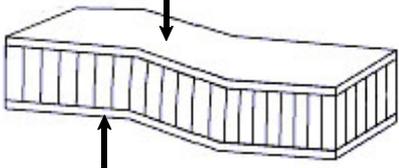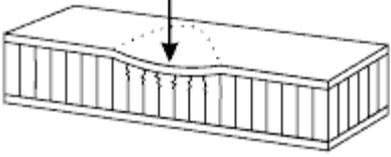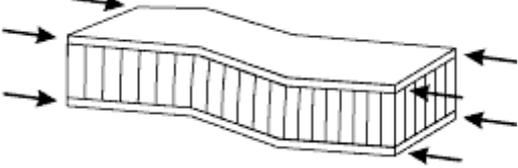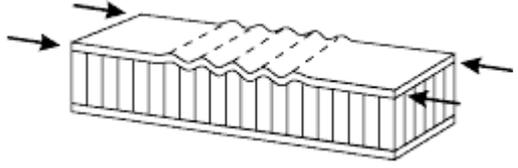
Sandwich structures should be designed to meet the following structural criteria [37]:

- Skin facings should be sufficiently thick to withstand tensile, compressive and in-plane shear stresses induced by the design loads.
- The core should have sufficient strength to withstand transverse shear stresses induced by the design loads.
- The core should have sufficient flexural and shear stiffness to avoid excessive deflections.
- The core should be sufficiently thick and have sufficient shear stiffness to prevent panel (or general) buckling of the sandwich under load.
- Compressive modulus of the core and compressive strength of the facings should be sufficient to prevent wrinkling of the faces under design loads.
- The core cells should be sufficiently small to prevent intracell dimpling of the facings under design loads.
- The core should have sufficient compressive strength to resist crushing by design loads acting normal to the panel facings or flexure induced compressive stresses.
- Material strength in the vicinity of cut-outs and attachments should be sufficient to prevent failure in these regions of stress concentrations.

## 6.1    FAILURE MODES

Table 14 summarises the causes and modes of failure that can occur in sandwich panels.

**Table 14:  Sandwich Structure Failure Modes**
**(Courtesy of Hexcel Composites)**

| | |
|---|---|
| **Facing Failure** | Insufficient panel thickness, facing thickness or facing strength may result in failure occurring in either the compression or tension face. |
| **Transverse Shear Failure** | Transverse shear failure is caused when either the shear strength of the core or panel thickness is insufficient. |
| **Local Crushing of Core** | Local crushing of core is caused when the compressive strength of the core material is too low. |
| **Panel Buckling** | Panel (or general) buckling is caused when either the panel thickness or core shear stiffness is too low. |
| **Shear Crimping** | Shear crimping, which can occur as a consequence of general buckling, is caused when either the shear modulus of the core material or the shear strength of the adhesive is low. |
| **Face Wrinkling** | Core compression failure (buckling inwards) or adhesive bond failure (buckling outwards) may occur depending on the relative strengths of the core in compression and adhesive in flatwise tension. |
| **Intracell Buckling (Dimpling)** | Intracell buckling (applicable to cellular cores only) occurs where the skins or faces are very thin and the cell size is large.  This effect may cause failure by propagating across adjacent cells, thereby inducing face wrinkling. |

## 6.2    SANDWICH PANEL ANALYSIS
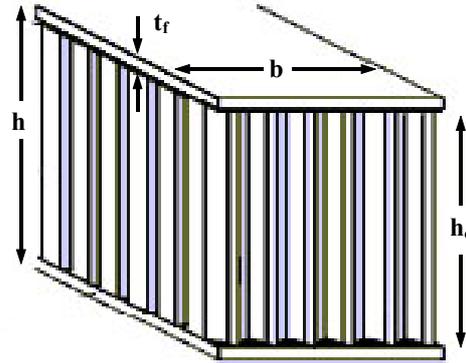


**Figure 16:  Sandwich panel with dimensions.**

### 6.2.1    Flexural Rigidity

Flexural rigidity **EI** of a symmetric sandwich beam (i.e. materials and skin thickness are identical) can be determined using the following relationship (Figure 16) [50]:

$$EI = \frac{1}{12}\left[E_f b\left(h^3 - h_c^3\right) + E_c b h_c^3\right]$$

(15)

where $E_f$ and $E_c$ are the moduli of the core respectively, b is the width of the beam, $h_c$ is the core thickness and **h** is the overall thickness of the sandwich panel.  If $E_f \gg E_c$ then Equation (15) reduces to:

$$EI = \frac{E_f b}{12}\left(1 - \frac{h_c^3}{h^3}\right)$$

(16)

### 6.2.2    Facing/Skin Design

Under flexural (or bending) loads one face is in tension and the other is compression.  If the stresses in either facing exceed the corresponding ultimate stresses of the constituent materials of the facings, then the sandwich panel will fail in a catastrophic manner.  In design calculations, the strength verification of the facings is usually carried out by comparing the stresses caused by external loads with the allowable stresses for the constituent materials of the facing.  The allowable stresses are obtained by dividing the strengths by suitable factors which take into account the variable properties of the materials, the approximations in structural design, accidental loads, fatigue performance, etc.

When the calculated stresses exceed the allowable stresses, a change in sandwich design is required.  In such a case, the following is recommended [51]:

* Use a material with higher allowable stresses (i.e. strengths) for the facing; or
* Increase facing thickness, thus reducing the applied stresses; or
* Increase core thickness, thus reducing the applied stresses (preferred method).

**Note:  A higher density (i.e. stiffer) core does not affect the stresses in the facings.**

6.2.3   Core Design

If the shear stress induced in the core is greater than the shear strength of the core material the core will fail, resulting in failure of the sandwich structure.  As before, the allowable shear stress is obtained by dividing the shear strength of the core by a suitable safety factor.  When the calculated shear stress exceeds the allowable shear stress, a change in sandwich design is required.  In such a case, the following is recommended [51]:

- Use a core material with higher allowable shear stress; or
- Increase the core thickness (preferred method).

**Note:  Using a different material for the skins or increasing skin thickness has no affect on the shear stress in the core.**

6.2.4   Buckling

Panel (or general) buckling (see Table 14) may occur when the stress in the facings and in the core is lower than the allowable stress.  The load that determines the sandwich instability depends on the beam dimensions and constraint conditions.  Important factors include [51]:

- Flexural rigidity of the sandwich
- Thickness and elastic properties of the facings
- Thickness and shear modulus of the core

In order to avoid this mode of failure, it is necessary that the buckling load is higher (according to a suitable safety factor), than the predicted compression strength of the panel. The following steps are recommended:

- Use facings constructed from materials with a high elastic modulus; or
- Increase facing thickness; or
- Increase core thickness; or
- Use a core material with a high shear modulus.

For a sandwich panel with a relatively low length/thickness ratio and where the shear rigidity is far less than the flexural rigidity, general buckling will occur in the form of shear crimping (see Table 14).  The total load per unit length capable of producing crimping is virtually independent of the skin properties and increases linearly with core thickness and shear modulus of the core.

The critical stress causing general buckling of a sandwich beam under compressive load is given by [37]:

$$\sigma_{cr} = K_d \frac{E_f}{\left(1 - v_f^2\right)} \left(\frac{t_f}{s}\right)^2 \tag{17}$$

where **s** is the size of honeycomb cell (i.e. diameter of the largest circles that can be inscribed in a core cell and $\mathbf{K_d \approx 2}$.  For orthotropic facing materials, $\mathbf{v_f = (v_{xy}v_{yx})^{1/2}}$.

## 6.2.5   Local Buckling (or Dimpling)

The core prevents the buckling of the skins under compressive loads by laterally supporting the skins.  If the compressive stress exceeds the compressive stress limit, the core will be unable to prevent local buckling.  This mode of failure is a local phenomenon and is therefore independent of the panel geometry and constraint conditions.   The main causes of local buckling are:

- Interfacial failure between skin and core (i.e. debonding); or
- Failure under tension or compression of the core.

The failure mode is evident by wrinkling (see Table 14), which is catastrophic and is influenced by the following parameters [51]:

- Elastic modulus of the facings
- Elastic and shear modulus of the core

In order to prevent local wrinkling, the compressive stress in the sandwich should be less than the critical buckling stress multiplied by a suitable safety factor.  Prevention can be achieved through the use of a high-density core material and ensuring good adhesive bonding between the core and facings.  Buckling may occur in the free spaces within the single cells (i.e. dimpling).  Increasing the thickness or modulus of the facings, or using a core with a small cell size can prevent dimpling [51].

The ultimate face wrinkling stress of a sandwich panel with either an isotropic or orthotropic core (i.e. honeycomb) is given by the following relationships [37]:

$$\sigma_{cr} = Q \left( \sqrt[3]{\frac{E_c E_f G_c}{\left(1 - v_f^2\right)}} \right) \qquad \text{isotropic} \qquad (18)$$

$$\sigma_{cr} = Q \left( \sqrt[3]{\frac{E_c E_f t_f}{\left(1 - v_f^2\right) h_c}} \right) \qquad \text{orthotropic} \qquad (19)$$

The recommended safe design value of **Q** for isotropic and orthotropic core materials is 0.5 and 0.33, respectively.

## 7.   DISCUSSION AND CONCLUDING REMARKS

The tendency within industry is to design to suit a particular application, often relying on previous experience, as there is minimal time for a comprehensive design/test programme. Standard detail design is such an approach that has been successfully employed in preliminary design of bonded and bolted structures, although this design approach based on the results from simple specimens and standard joints is still far from satisfactory.  Whichever approach is adopted, it is necessary to ensure that the range of stress and strain variation in regions of high stress are kept below the endurance limit of the adhesive, and that peel and cleavage stresses in these regions are minimised.

The approach commonly adopted by industry is to use large safety factors to account for environmental degradation, fatigue loading and the level of maintenance and inspection of the structure in service. Partial safety factors used in design are generally based on previous experience or values derived from published literature [12, 23]. If there are any uncertainties, the safety factors are increased. There are no national or international standards specifically aimed at the design of adhesively bonded or bolted joints. Confirmation tests of assemblies representative of the actual structure need to be carried out to finalise the design. Testing of safety critical structures is advised and generally required as part of the quality assurance process. The loading conditions should be representative of the in-service conditions to be experienced by the structure.

In order to maintain or improve market position, there is a drive to minimise development and production time and cost. There is a generally a small profit margin, and therefore there is a need to have quick, easy-to-use formulae for designs, similar to those highlighted in this report. This situation is further exacerbated where fatigue data is required, as extensive testing to generate the database is required. The historical approaches to design of iterative design prototyping or detail design are inflexible and expensive. Although, computer design can facilitate rapid design of complex engineering structures (i.e. FEA), there is a need for simple and easy to use design rules and analytical procedures that can be used at the preliminary design stage for prescribing the basic test geometry, and material and geometric parameters.

Within the framework of MMS11 "Design for Fatigue and Creep in Joined Systems" it is intended that a generic structure (i.e. slotted T-joint) be used as a template for assessing both bonded and bolted design approaches. It is recommended that the static, creep and fatigue performance be assessed using both stress-based and fracture mechanics based design methodologies in order to ascertain the most suitable approach and to provide guidance on the use of either design methodology. As designers often use strain as the controlling factor in design, strain-based design requirements will be considered within the programme.

## ACKNOWLEDGEMENTS

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# Appendices

# Appendix 1:  Test Methods

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

| Material Property | Standard/Test Method |
|---|---|
| **Elastic Properties - Adherends**<br>Metals<br>E, G, $\nu$<br>Composites<br>In-plane ($E_{XX}$, $E_{YY}$, $\nu_{XY}$)<br>Through-thickness ($E_{ZZ}$, $\nu_{XZ}$, $\nu_{YZ}$)<br>In-plane shear ($G_{XY}$)<br>Through-thickness shear ($G_{XZ}$, $G_{YZ}$) | <br><br>Tensile test of plastics - BS EN ISO 527-2<br>**m.d = multidirectional, u.d = unidirectional**<br>Tension - BS EN ISO 527-4 (m.d)/BS EN ISO 527-5 (u.d)<br>T-T tension and compression - NPL draft procedures<br>±45° tension method - BS EN ISO 14129 (u.d)*<br>V-notched beam test - ASTM D 5379 |
| **Strength Properties - Adherends**<br>Metals<br>Tension<br>Compression<br>Shear<br>Composites<br>In-plane tension ($S_{XX}$, $S_{YY}$)<br>Through-thickness tension ($S_{ZZ}$)<br>In-plane compression ($S_{XX}$, $S_{YY}$)<br>Through-thickness compression ($S_{ZZ}$)<br>In-plane shear ($S_{XY}$)<br>Through-thickness ($S_{XZ}$) | <br><br>Tensile testing of metallic materials - BS EN 10002-1<br>Compression testing of metallic materials - ASTM E9<br>Shear modulus - BS EN 10002-1*<br>**m.d = multidirectional, u.d = unidirectional**<br>Tensile - BS EN ISO 527-4 (m.d)/BS EN ISO 527-5 (u.d)<br>Through-thickness tension - NPL draft procedure<br>Compression - BS EN ISO 14126<br>Through-thickness compression - NPL draft<br>±45° tension method - BS EN ISO 14129 (u.d)<br>V-notched beam method - ASTM D 5379 (u.d) |
| **Elastic Properties - Adhesives**<br>E, G, $\nu$ | <br>Tensile test of plastics - ISO 527-2<br>V-notched beam method - ASTM D 5379 |
| **Strength Properties - Adhesives**<br>Tension<br>Compression<br>Shear<br>Maximum principal strain | <br>Tensile test of plastics - BS EN ISO 527-2<br>Compressive testing of rigid plastics - IS0 604/ ASTM D695<br>V-notched beam method - ASTM D 5379<br>Tensile test of plastics - BS EN ISO 527-2 |
| **Fracture Toughness**<br>Mode I – composites<br>Mode I – adhesive joints<br>Mode II – composites<br>Mode II – adhesive joints | <br>Double cantilever beam (DCB) test - ISO 15024/prEN 6033<br>As above – draft BSI under review<br>End notched flexure (ENF) test - prEN 6034<br>As above – no national or international standards |
| **Joint Coupon Tests**<br>Tension shear strength and modulus<br>Shear strength and modulus<br>Compression strength and modulus | <br>Butt joint<br>Thick adherend shear test (tension) - ISO 11003-2<br>Butt joint |
| **Additional Tests**<br>Tensile strength of lap joint<br>Tension-tension fatigue<br>Moisture absorption/conditioning<br>Effect of water/moisture<br>Effect of chemicals<br>Effect of hear ageing<br>Test and conditioning atmospheres<br>Tensile creep behaviour of plastics<br>Failure patterns<br>Dynamic Mechanical Analysis<br>Differential Scanning Calorimetry | <br>BS EN 1465<br>BS EN ISO 9664<br>BS EN ISO 62<br>ISO 62/ISO 175<br>ISO 175<br>ISO 216<br>ISO 291<br>ISO 899-1<br>EN 923<br>ISO 6721-4<br>ISO 11357 |

Symbols: E = modulus of elasticity, G - shear modulus, $\nu$ = 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)

# Appendix 2:  Design Procedure 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.

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.



Shear stress along adhesive
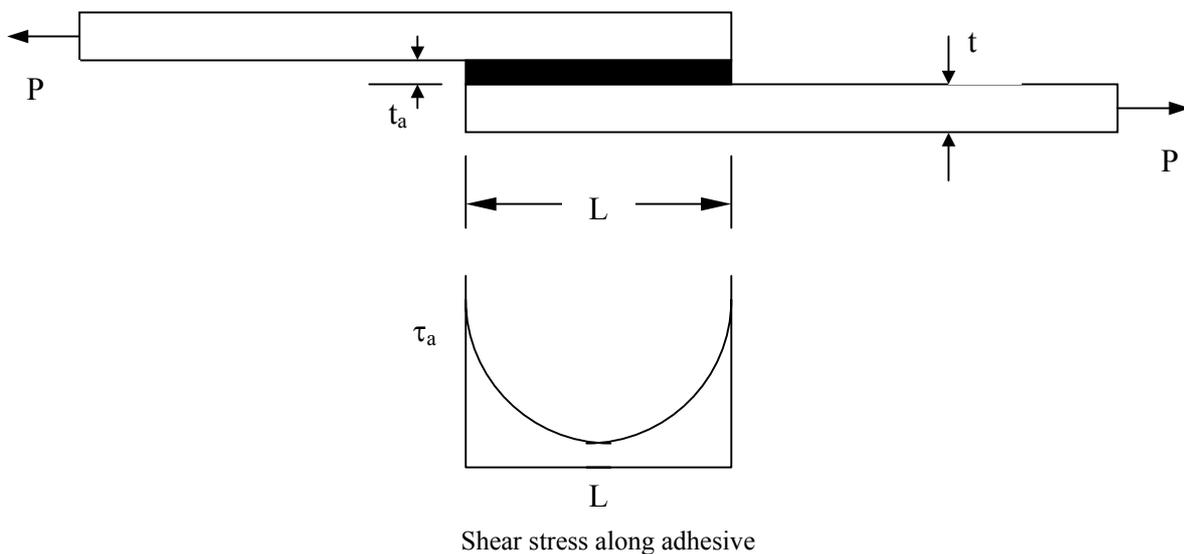
**Figure A2.1:  Schematic of single-lap joint.**

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.

The following design procedure is used in the EUROCOMP design code and handbook [12].

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

$$\frac{\beta}{t} = \sqrt{8\frac{G_a}{Et_a t}} \tag{A2.1}$$

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:

$$\tau_{0max} = \frac{\sigma}{8}(1+3k)\sqrt{8\frac{G_a t}{Et_a}} \tag{A2.2}$$

where:

$$\sigma = \frac{P_d \gamma_f}{t} \tag{A2.3}$$

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

$$u_1 = 2\sqrt{2}\,u_2 \tag{A2.5}$$

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

and

$P_d$ = design load per unit width
$\nu$ = adherend's Poisson's ratio
$\gamma_f$ = partial safety factor

**Step 3:** The calculated value of **$\tau_{0max}$** should be checked against the maximum allowable adhesive shear stress, i.e.

$$\tau_{0max} \leq \tau_{0allowable} \tag{A2.7}$$

**Step 4:** Calculate the value of $\lambda$ as follows:

$$\lambda = \frac{c}{t}\left(\frac{6E_a t}{Et_a}\right)^{0.25}$$

(A2.7)

where:

$$c = \frac{L}{2}$$

(A2.8)

and $E_a$ = adhesive tensile modulus.

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

$$\sigma_{0max} = \sigma\left[\frac{k}{2}\sqrt{6\frac{E_a t}{Et_a}} + \frac{k't}{c}\sqrt{6\frac{E_a t}{Et_a}}\right]$$

(A2.9)

where

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

(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:

$$\sigma_{0max} \leq \sigma_{0allowable}$$

(A2.11)

$$\sigma_{0max} \leq \sigma_{z\,allowable}$$

(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 $\tau_{0max}$ and $\sigma_{0max}$.

# Appendix 3: Sandwich Test Methods

## Table A3.1: List of Sandwich ASTM Test Methods

The following is a list of ASTM sandwich test methods:

- C271 - Standard Test Method for Density of Core Materials for Structural Sandwich Constructions.
- C272 - Standard Test Method for Water Absorption of Core Materials for Structural Sandwich Constructions.
- C273 - Standard Test Method for Shear Properties of Sandwich Core Materials.
- C297 - Standard Test Method for Flatwise Tensile Strength of Sandwich Constructions.
- C364 - Standard Test Method for Edgewise Compressive Properties of Sandwich Cores.
- C365 - Standard Test Method for Flatwise Compressive Properties of Sandwich Cores.
- C366 - Standard Test Method for Measurement of Thickness of Sandwich Cores.
- C393 - Standard Test Method for Flexural Properties of Sandwich Constructions.
- C394 - Standard Test Method for Shear Fatigue of Sandwich Core Materials.
- C480 - Standard Test Method for Flexure-Creep of Sandwich Constructions.
- C481 - Standard Test Method for Laboratory Aging of Sandwich Constructions.
- D1781 - Standard test Method for Climbing Drum Peel for Adhesives.

## Table A3.2: Summary of ASTM Sandwich Test Methods

| TEST METHOD | C273 IN-PLANE SHEAR | C297 FLATWISE TENSION | C364 EDGEWISE COMPRESSION | C365 FLATWISE COMPRESSION | C393 FLEXURAL PROPERTIES | D1781 CLIMBING DRUM PEEL |
|---|---|---|---|---|---|---|
| TEST CONFIGURATION |  |  |  |  |  |  |
| PROPERTIES MEASURED | In-plane shear strength and modulus of sandwich core. | Flatwise tensile strength of sandwich core and/or skin/core bond strength. | Sandwich facing (skin) compressive strength. | Flatwise compression modulus and strength of sandwich cores. | Core shear strength, facing (skin) bending strength, effective panel shear and flexural moduli. | Torque resistance of adhesive between skin and core. |

## Table A3.2:  Summary of ASTM Sandwich Test Methods (cont'd)

| TEST METHOD | | C273 IN-PLANE SHEAR | C297 FLATWISE TENSION | C364 – 94 EDGEWISE COMPRESSION | C365 FLATWISE COMPRESSION | C393 FLEXURAL PROPERTIES | D1781 CLIMBING DRUM PEEL |
|---|---|---|---|---|---|---|---|
| SPECIMEN DIMENSIONS | SHAPE | Rectangular | Square (or round of equivalent area) | Rectangular | Square (or round or equivalent area) | Rectangular | Rectangular |
| | THICKNESS | = Sandwich thickness | = Sandwich thickness | = Sandwich thickness | As agreed upon by purchaser and seller | = Sandwich thickness | = Sandwich thickness, thick enough for sandwich panel not to bend |
| | WIDTH | > 50 mm | For square specimens; > 25 mm for continuous cores > 51 mm for open-cell cores of diameter <6mm > 77 mm for open-cell cores of diameter > 6 mm | At least 50mm, not less than 2 x sandwich thickness, nor less than four complete honeycomb cells | For square specimens; > 25 mm for continuous cores > 51 mm for open-cell cores of diameter <6mm > 77 mm for open-cell cores of diameter > 6 mm | Not less than twice the total thickness, not less than three times the dimension of a core cell, nor greater than one half the span length | 76 mm |
| | LENGTH | > 12 x sandwich thickness | For square specimens; > 2 5 mm for continuous cores >51 mm for open-cell cores of diameter < 6 mm >77 mm for open-cell cores of diameter > 6 mm | Unsupported length not greater than eight times the sandwich thickness. | For square specimens; >25 mm for continuous cores >51 mm for open-cell cores of diameter < 6mm >77 mm for open-cell cores of diameter >6 mm | Equal to span length plus 50 mm or plus one half the sandwich thickness whichever is greater. Some guidance given as to appropriate span lengths for property required | 305 mm plus 25 mm overhang of one facing at each end |
| QUANTITIES MEASURED | | Load and displacement between loading plates | Load | Load, strains in sandwich facings | Load and displacement between loading platens | Load and displacement measured at mid-span | Load and crosshead displacement |
| APPLICABLE TO: | | Sandwich construction and core | Sandwich construction and core | Sandwich construction | Sandwich construction and core | Sandwich construction | Sandwich construction |
| NUMBER OF SPECIMENS | | No guidance | At least 5 for acceptance tests | No guidance | At least 5 for acceptance tests | No guidance | Two or more from each panel being tested |
| METHOD FOR MEASURING DISPLACEMENT/STRAIN | | Deflectometer, compressometer or extensometer across loading plates | N/A | Strain gauges | Deflectometer or compressometer positioned through centre of specimen | Deflectometer or dial gauge | Crosshead |
| SUGGESTED LOADING SPEED | | 0.5 mm/min | 0.5 mm/min | 0.5 mm/min | 0.5 mm/min | 0.5 mm/min | 25.40 ±2.54 mm/min |
| STRENGTH | | Based on maximum load | Based on maximum load | Based on maximum load | Based on maximum load or load at 2% strain for cores that continue to compress | Based on maximum load | Peel strength quoted as a torque value based on maximum load with compensation made for torque to roll skin |
| MODULUS | | Based on slope of initial portion of load - displacement curve | No modulus required | No modulus required | Based on slope of initial portion of load - displacement curve | Derived from solution of simultaneous mid-span deflection equations | No modulus required |