Fatigue behaviour of adhesively bonded joints and its prediction using stress analysis

Georges Romanos, Henkel AG & Co KGaA, Global Engineering Center, Garching (Germany)

Copyright © 2015 International Automotive Body Congress

Abstract

Driven by the lightweight trends, adhesive bonding is increasingly used as assembly technology in body-in-white applications. The remarkable development in the toughening of high-strength adhesive polymers in the last decades, and improvements in the computational prediction of their mechanical performance under impact conditions, have enabled engineers to take advantage of the adhesive technology benefits and enlarged the application area significantly.

Fatigue loading is a common cause of failure in structural applications. Fatigue damage of adhesively bonded joints, in addition to crashworthiness, has therefore become subject of unquestionable relevance. The paper presents a concept for predicting fatigue strength of adhesive bonded joints based upon stresses that the joint interface may experience under operating conditions. The method proposed herein takes into account anisotropic failure behaviour of polymer materials and stress ratio effects. The paper demonstrates transferability of fatigue data evaluated using standard specimens, to bonded components and structures for fatigue life prediction by means of numerical stress analysis.

Introduction

Adhesive bonding has been initially introduced as backup technology for improving stiffness and performance under impact (crash) loading in body-in-white applications. Due to the lightweight trends, it gains now also considerable attention as primary assembly technology able to support the use of dissimilar materials than cannot be effectively jointed using mechanical fastening or welding techniques. Figure 1 demonstrates the stiffness benefit when using adhesive bonding as assembly technology in comparison to standard spot welding in body-in-white applications.

Figure 1: Load resistance (solid lines) and deformation energy (dotted lines) in 3-point-bending steel box beams assembled using
various jointing techniques
The load-displacement curves shown in Figure 1 were recorded in 3-point-bending 900 mm long box beam specimens made from 1 mm thick dual phase steel. The load is applied at a 45° angle to the overlap flanges in order to enable worst-case local loading conditions. The tests were conducted as part of a recent research program at the Engineering Center of the Henkel Corporation in Garching, Germany. The stiffness and strength increase is evident. Weldbonded double hat profiles, i.e. profiles with first bonded then spot welded overlap flanges, shown identical performance to only bonded ones.

Besides stiffness benefits, structural adhesives offer unique advantages under fatigue loading conditions due to their ability to transmit loads evenly and to avoid stress concentrations. The benefit in fatigue performance using adhesive bonding appears indeed more pronounced than the benefit in stiffness and strength. Usually, design engineers may consider 3 times higher load amplitudes to be transmitted by a weldbonded overlap than by a spot welded one.

Figure 2 illustrates a comparison of the fatigue performance of 420 mm steel box beams assembled by spot welding respectively by adhesive bonding. For the bonded components only, the flange width has been reduced to 50% of the initial one, as the torsional load required in low cycle fatigue testing would otherwise exceed the load ability of the torque actuator used. Despite the reduced bond area, adhesively bonded box beams withstand 2 to 3 times higher rotational amplitudes than those assembled by spot welding. The benefit becomes more prominent in the high cycle fatigue area, the fatigue life of adhesively bonded components being 20 to 100 times larger than the life of spot welded ones.

Spot welded box beams failed by cracks growing from the interface through the thickness of the substrates (compare also [1], p.412). The crack propagated tangentially to the weld and round the outside of the nugget. In contrary, bonded box beams failed by interface
failure without crack growing into the substrates. This indicates even load distribution throughout a sufficient joint area. Adhesive bonding is therefore suitable for jointing high strength materials that are frequently sensitive to stress concentration.

Taking full advantage of the adhesive bonding technology requires credible methods for predicting the mechanical behaviour of the intended application under the expected operating conditions. While remarkable progress has been made in predicting the behaviour of adhesive bonds under impact load conditions using finite element analysis (FEA), fatigue life prediction is still being under consideration.

Fatigue failure prediction is related to two main tasks: (a) Failure behaviour of the adhesive material under cyclic loading including failure anisotropy, multiaxial and mean stress effects; (b) appropriate failure criteria to be used in numerical analysis, and methods to transfer fatigue data to arbitrary joint geometry.

A rational basis of fatigue design must be based on the ability to predict the stresses, which are likely to be encountered in practice under complex loading conditions, and the probability of damage or the fatigue life expected. Apart from some simple (however typical) joint configurations accessible to analytical solutions, stresses and strains or crack tip deformation can be evaluated in realistic manner only using finite element analysis (FEA) or comparable numerical methods. The FEA offers highly detailed geometric capabilities and allows for consideration of complex material behaviour and large deformations. However, it is still a numerical treatment method, the results of which are strongly dependent on the modelling techniques and the properties implemented.

Fatigue Life Prediction Methods

The total fatigue life of a component is considered consisting of a damage initiation phase and a propagation phase. The later suggests that a macro crack has formed, propagating afterwards under cyclic loading until total joint failure.

Most of the research efforts reported on the fatigue behaviour of adhesively bonded joints have focused on fatigue crack growth, and used fracture mechanics theory for predictive purposes [3-6]. Although fracture mechanics based approaches have been found to satisfactorily describe crack propagation, crack initiation can dominate the service life of bonded structures (particularly at low stresses resp. in high cycle fatigue regime), and therefore, predictions based only on crack propagation may underestimate the real fatigue life [7, 8].

Fracture mechanics approaches are in general difficult to be used in the design of large structural applications, as they require continuous remeshing when crack propagation is to be predicted [3]. Considerable progress has been achieved by the development of cohesive zone modelling approaches [9] that can capture fatigue degradation of the cohesive interface. However, degradation and crack growth can be predicted only incrementally, this leading to high computational effort. In addition, complex property calibration is required.

Some attempts have been also made to relate crack nucleation, and the corresponding lifetime sequence needed for, to the singular stress field located at the joint corners [10, 11]. However, stress singularity approaches need suggestions on fillet and bondline geometry that cannot be defined in this detail depth in automotive applications. They are rather useful for evaluating delamination probability in bonds that exhibit sharp and well-defined corners of adhesive fillets as typically found in microelectronic applications.

Stress- (or strain-) life concepts have instinctive appeal to design engineers because of their direct approach, and their similarity to methods established for metallic materials. One may also appreciate that such methods can be easily implemented in post-processing algorithms using data from FEA. However, the assessment of a critical stress or strain is still complex as stress concentration occurs at the joint end and the overlap is generally modelled as sharp corner providing infinite stress or strain. Stress singularity can be removed to some degree by the use of elastic-plastic adhesive material response, but stresses and strains at the overlap ends are still sensitive to the mesh density or geometry assumptions.

A convenient choice is to use the stress at the distance at which the stress becomes practically independent of mesh refinement for a given level of mesh size, or an empirical distance from appropriate testing on a range of specimen types [12] (“stress/strain at a distance criterion”). A similar approach that has been successfully used to describe static failure of a range of joints, states failure to occur when the maximum principal stress exceeds the ultimate (tensile) stress of the adhesive over a finite zone [13] (“stress/strain over a zone criterion”). An application to the fatigue life prediction of bond repairs for fibre-composite material has however shown limited correlation between numerical prediction and experimental results [14].

In general, it is reasonable to evaluate adhesive failure using an average stress failure criterion for a characteristic distance over which the adhesive stress is averaged [15]. Crocombe et al [16] used an average modified strain over two elements in FE analysis comprising of the plastic strain and a term related to hydrostatic stress to evaluate strain-life curves for varying joint types. The critical distance theory (CDT) proposed by Taylor [17], which has been found to apply to metals and polymer materials, uses both, stress at distance and average stress over a zone criterion. Whitney et al. [18] applied three decades before Taylor the critical distance theory to the
(static) failure of notched composite structures. A critical distance or zone is associated with the fatigue process zone but mechanistic approaches that directly relate the critical distance to threshold toughness and plain fatigue limit are still questionable.

Critical distance theories have been early postulated for metal fatigue by Neuber and Peterson (s. [1], pp. 196-198). They have been mainly used indirectly to reduce the classical stress concentration factor to a reasonable fatigue stress measure. The difference between stress concentration and notch factor is in general believed to be related to the stress gradient and localized plastic deformation at the stress root. The reasoning for stress gradient influence is that the notch stress controlling the fatigue life may be not the maximum stress on the surface of the notch root, but rather an average stress acting over a finite volume of the material at the notch root [1]. A gradually decreasing stress distribution from a notch root has different effect on material fatigue damage from that of a step decrease in stress. It is obvious that the former is more damaging [2].

Pluvinage and co-workers [19] proposed a method to estimate the effective stress over a critical zone using a weighted average of the stress field depending on the stress gradient. Moreover, also the length of the integration zone has been suggested to depend on the stress gradient. The method has been yet applied with success to metal fatigue only.

The question of an appropriate equivalent stress (or strain) for structural adhesives has been frequently addressed [16, 21] for joints under monotonic loading conditions. Fatigue testing on bonded CFRP-metal joints has indicated the maximum principal stress to be appropriate for fatigue strength estimation under various multiaxial load conditions [22].

**Stress-Life Approach for Structural Adhesives**

A stress-life approach for adhesive joints will be presented based upon stress over a zone (averaged stress) criterion. It takes into account failure behaviour of polymer materials under multiaxial stress conditions and stress ratio effects. The approach has been developed in a research program using a toughened structural adhesive of the Henkel Corporation for body in white applications, but has been meanwhile successfully applied to various other adhesives grades and applications.

The method makes use of fatigue data evaluated using standard specimens, applying them to bonded components and structures by means of numerical stress analysis (FEA).

**Fatigue Properties of the Adhesive Bond**

Experimental testing is required to generate the mechanical properties and the parameters associated with fatigue damage and failure. A potential difficulty is that fatigue life and crack growth may be geometry dependent and hence data from a stress-number of load cycles-curve (S-N-curve) cannot easily be used to predict the fatigue life for a different sample geometry.

Reference fatigue limit can be evaluated using specimens with uniform stresses throughout overlap or cross section. Uniaxial tensile bulk specimens are easy to test but also subjected to statistical size effects as the number of inherent flaws is higher than in thin bondline joints. In addition, they cannot be used to generate properties in shear, which shall be the predominant loading case for bonded joints, or to provide information for anisotropic behaviour and response under multiaxial stress conditions. Moreover, bondline thickness in in-situ testing may essential affect toughness and hence the apparent strength of the joint as plasticity at a potential crack tip may be constrained by the adherends.

For the purpose of fatigue limit evaluation, butt-bonded hollow cylinders test procedures based upon ISO 11003-1 [23] have been employed. The butt torsion test is known to provide a very uniform shear stress in the adhesive layer with negligible variation across the adhesive ring. Shear strains and stresses to failure determined for adhesives in this test are larger than those determined in thick-adherend shear testing. The test consists of applying torque to two tubular adherends butt joined by an adhesive ring (Figure 3) but can be also used to impose tensile stresses perpendicular to the adhesive layer, or combination of tensile and shear stresses in arbitrary ratios if the tests are carried out in an appropriate bi-axial test machine.

Tensile loading the butt bonded specimen results in triaxial stress conditions as the load carrying adherends are restrained from contracting freely and lateral tensile stresses are developed [24]. The additional stress components of the adhesive can be readily determined from the normal stress $\sigma_x$ in loading direction by

$$\sigma_x = \sigma_y = 1 - \nu \sigma_x$$  \hspace{1cm} (1)

suggesting plain strain conditions [24], where $\nu$ denotes the Poisson's ratio of the adhesive material. The triaxial stress state conditions have been verified by appropriate FE analysis.
Fatigue testing was carried out for a typical bondline thickness of 0.2 mm at 10 Hz in shear and axial tensile loading as well as in combination of them in a ratio of 1:1. The stress ratio R employed, i.e. the ratio of minimum stress to the maximum stress during a load cycle, was 0.1, but test series were also conducted for pure shear and combined (1:1) loading with varying stress ratio including R=−1 (pure alternating loading) and R=1 (static loading with no cyclic load fraction).

Tests were stopped after failure has propagated throughout the bond area, or after the specimen has reached 10 Mio of load cycles (number of fault test cycles \( N_F \)) without to fail. Load amplitudes were chosen such that the failures occur between \( 10^4 \) and \( 10^7 \) cycles.

Fatigue data is plotted in the form of an S-N diagram, which is a plot of the number of cycles required to cause failure in a specimen against the amplitude of the applied stress. The S-N-curve in the fatigue zone is suggested to follow the relationship

\[
N = N_F \cdot \left( \frac{\sigma}{\sigma_0} \right)^k \cdot N_F
\]

(2)

where \( k \) is a factor that represents the slope of a curve when both, service life \( N \) and stress amplitude \( \sigma \) are expressed in logarithmic scale. \( \sigma_0 (N_0) \) denotes the limit of endurance, i.e. the stress determined at the specific number of fault test cycles (here \( 10^9 \)). The stress amplitude is defined in accordance to ISO 9664 as half the difference between maximum and minimum stress \[25\]. Data points at \( 10^7 \) load cycles are run outs passed the full test duration without to fail. They are positioned into the diagram but not used in statistical S-N evaluation.

The S-N curve provided by linear regression trough the centre-line points represents the relationship between the service life \( N \) and the stress amplitude in an average sense, i.e. this is the median S-N curve valid for \( p=50\% \) survival probability. S-N curves for \( p \) and \( 100- p \) survival probability can be defined using statistic methods, \( p \) talking usually values between 90\% and 99\%.

Due to the uniform stress distribution over the bond overlap, results from butt-bonded hollow cylinders testing are suggested to correspond to the intrinsic fatigue properties of the adhesive material formed in a bond between metallic substrates with a given bondline thickness and under appropriate curing conditions.

**Equivalent stress condition**

It is convenient to express complex or deviating stress states in terms of a single (equivalent) stress amplitude corresponding to the maximum distortion strain energy criterion, usually denoted also as von Mises equivalent stress, \( \sigma_e \)

\[
\sigma_e = \sqrt{3} \cdot J_2
\]

(3)

where \( J_2 \) is the second deviatoric stress invariant. Figure 4 illustrates S-N-curves obtained by fatigue testing appropriate butt-bonded hollow cylinder specimens bonded with a toughened structural adhesive grade in shear, axial tensile and combined loading at a stress ratio R=0.1.
By reviewing the S-N curves in Figure 4 it becomes evident that von Mises equivalent stress does not apply to the fatigue strength behaviour of structural adhesives.

In general, linear, non-linear and plastic failure of polymer materials has been suggested to depend on both, deviatoric and volumetric stress components and this has led to appropriate modifications for taking into account hydrostatic stress effects, which the von Mises equivalent stress does not consider. Among others, a relatively simple but sufficiently general yield and failure condition for anisotropic materials is proposed in the form

\[ \sigma^2 + 3(\lambda - 1)\sigma_m \cdot \sigma_T - \lambda \cdot \sigma^2 = 0 \]  

(4)

where \( \sigma_m \) is the hydrostatic stress, \( \sigma_T \) denotes the ideal failure stress limit under uniaxial stress conditions, \( \sigma_e \) is derived from Equation (3) and \( \lambda \) expresses the strength differential effect. This condition has been first applied to polymer materials by Raghava et al. [20], then successfully used to described yielding for structural adhesives (e.g. in [21]). For convenience, the failure stress limit can be expressed as a modified equivalence stress \( \sigma_{e,mod} \) in terms of the hydrostatic stress and von Mises stress, and hence Equation (4) turns to

\[ \sigma_{e,mod} = \lambda \cdot 3 \cdot \sigma_m + \sqrt{(3 \cdot \sigma_m)^2 + \lambda \sigma^2} \]  

(5)

A third failure condition that has been considered, is the maximum principal stress condition:

\[ \sigma_1 = \sigma_T \]  

(6)

Although expected to apply rather to brittle adhesives than to toughened structural ones, the use of the principal stress as equivalent stress is reasonable because of the tendency of adhesives to fail through tensile mechanisms.

Application of the failure conditions corresponding to equation (5) (modified equivalent stress condition) and (6) (maximum principal stress condition) have enabled to represent all available test data from different loading states each with a single, narrow-scatter S-N curve (Figure 5). We assume hence them, in contrary to the standard von Mises equivalent stress condition, to be appropriate for describing the fatigue behaviour of structural adhesive bonds under arbitrary modes of loading using a single mode-independent stress measure.
Both failure conditions have been found also to apply with good accuracy to cyclic loading with a stress ratio of \( R = -1 \) (pure alternating stress conditions) and 0.3, the corresponding tests carried out under shear and combined, shear and tensile loading. It is therefore reasonable to assume both conditions appropriate for arbitrary stress ratio of practical interest.

The factor \( \lambda \) best fitting the available fatigue data for the anisotopic failure condition is 1.9 approximately. The factor applies also to the static strength properties (ultimate stress) derived from in-situ testing in varying load modes.

**Mean Stress Effects**

Mean stress effects were also investigated by testing butt-bonded hollow cylinders specimens at several stress ratios. For each stress ratio, a separate S-N curve was evaluated. The effect of the mean stresses is best illustrated in a constant life diagram, in which stress amplitude (fatigue limit) is plotted against the mean stresses for a given fatigue life. A constant life diagram for a fatigue life of 10 Mio load cycles is shown in Figure 6. Stress amplitudes and mean stresses have been normalised with respect to the static strength as measured for each test series prior to fatigue testing.

Figure 6: Constant life diagram for a fatigue life of 10 Mio load cycles; stress amplitudes and mean stresses normalized with respect to the static strength
For simplicity, maximum principal stresses have been used to express varying stress states in terms of a single (equivalent) stress amplitude. The use of modified equivalent stresses in accordance to Equation (5) leads to comparable constant life diagrams.

For stress ratios $R \leq 0.3$ a Goodman (linear) relationship fits with reasonable accuracy to the fatigue limits plotted against mean stress, the intersection point at the mean stress axis being equivalent to the static strength value:

$$\frac{\sigma_a \sigma_{\text{mean}}}{\sigma_{ar} + \sigma_0} = 1$$

(7)

where $\sigma_a$ is the fatigue stress limit expressed as amplitude, $\sigma_a$ is the fatigue strength under full reversing stress conditions (stress ratio $R=-1$), $\sigma_{\text{mean}}$ denotes the mean stress and $\sigma_0$ the static strength.

For higher stress ratios $R > 0.3$ creep effects become predominant as mean stress becomes very high. The relationship between stress amplitude and mean stress can be now fitted with a straight line connecting the static strength on the ordinate axis to the creep strength, $\sigma_c$ on the abscissa axis:

$$\frac{\sigma_a \sigma_{\text{mean}}}{\sigma_0 + \sigma_c} = 1$$

(8)

The creep strength is the constant static stress causing rupture after a time $t$ corresponding to the desired number of load cycles or fatigue life at a given load frequency.

**Effective Stress**

While the equivalent stress condition has been evaluated based upon tests with uniform stress distribution along the overlap, the effective stress

$$\sigma_{\text{eff}} = \frac{1}{l_{cr}} \int_0^{l_{cr}} \sigma(x) \, dx$$

(9)

can be assessed as an average over a critical distance area $l_{cr}$ by appropriate testing joints with typical stress gradients.

For the purposes of the approach proposed, fatigue tests have been carried out using single lap shear specimens subjected to tensile loading and a T-peel joint specimen (Figure 7). The substrates with a thickness of 1 mm were made from dual phase zinc-coated steel with a yield point of 340 MPa approximately, and an ultimate tensile stress $\geq 600$ MPa. Single lap shear specimens undergo, in addition to shear, sufficient normal (peel) loading due to substrate bending when stressed by tensile loads. These are typical stress conditions, which may occur in practical joints. In a T-peel joint specimen, the adhesive bond is subjected to pure tensile loading normal to the joint interface.

In BiW applications, thin metal substrates shall usually be jointed with overlap lengths $l$ typically varying between $7.5 \leq l \leq 25$, where $t$ denotes the substrate thickness. For the single lap shear specimens two overlap lengths have been considered: (a) 12.5 mm, corresponding to the standard tensile lap-shear specimens as defined in ISO 4587:2003 [26], and (b) 25 mm, sufficient to provide maximum stress variation along the overlap. Distance from overlap ends to the clamping area have been chosen equal as to eliminate potential effects from deviating free arm bending conditions. A larger overlap has been omitted in order to avoid the required tensile load exceeding the fatigue limit of a single substrate cross section.

Bondline thickness (0.2 mm) and curing procedures were identical with those used in butt-jointed hollow cylinder testing. In lap shear test specimens, any excess of adhesive that may produce a fillet out of the overlap has been removed immediately after assembly and prior to curing. In T-peel test specimens, the fillet formed has been only partly removed due to limited access to the joint root (s. also Figure 10). All specimens have been tested under cyclic (sinusoidal) stress-controlled loading at constant stress ratio $R=0.1$ up to a total duration of 10 Mio. of load cycles or to specimen failure throughout the overlap. The load frequency used was 10 Hz. The specimen temperature has been monitored using infrared camera ensuring that no noticeable temperature increase has occurred during the test.
The test results from fatigue testing lap shear and T-peel test specimens are shown in Figure 8. All specimens have failed due to cohesive failure in the adhesive film. At moderate to high stress amplitudes lap shear specimen substrates exhibit noticeable remaining (plastic) deformation due to bending in the areas close to overlap ends.

In the corresponding stress analysis using FEA, substrates have been modelled as elastic-nonlinear plastic material using stress-strain curves from experimental testing. Adhesive has assumed to behave linear elastic, the data generated by appropriate tensile testing bulk
material specimens. Adhesive layer has been modelled using 4 rows of solid elements through thickness. However stresses have been evaluated in the mid plane of the adhesive layer since, due to stress averaging, singularities at the bondline corners do not considerably affect the effective (average) stress.

The analysis of the tested specimens consists of several independent computational runs, each of them reproducing a single load amplitude from the corresponding S-N diagram. Stress components, and equivalent stress distribution along the overlap were recorded as a function of the overlap distance for both, the maximum and minimum load, and their amplitude was derived.

For the purposes of the stress life approach, stress amplitude has been averaged over a certain characteristic length, the latter being incrementally increased from 0.5 to 6.25 mm. The procedure is illustrated in Figure 9 for the standard overlap length shear lap specimen at a (nominal) shear amplitude of 4.95 MPa and stress ratio R=0.1.

![Figure 9](image)

**Figure 9** Stress amplitude distribution along overlap in lap shear subjected to nominal shear amplitude of 4.95 MPa, and average stress values for varying average zone lengths

Stress analysis and subsequent evaluation of the fatigue data indicated that the zone length over which the effective stress shall be assessed as an average, might vary depending on the stress gradient. Higher stress gradients corresponds to larger zone lengths. While stress gradients in lap shear tensile loading are still comparable and thus, a common zone distance can be evaluated, the spew fillet in T-peel specimens provides a stress distribution with low stress gradient that may require a much smaller averaging zone.

Excess of adhesive that may produce a fillet out of the overlap has been immediately removed after assembly in the case of lap shear specimens. In T-peel specimen, where access to the fillet is difficult, removing of the adhesive excess has been done using an appropriately shaped PTFE tool. Therefore, a finite but controlled fillet has been obtained, the dimensions of which have been carefully measured after curing (Figure 10) using 3D optical digitizer ARAMIS (GOM Optical Measurement Techniques).

![Figure 10](image)

**Figure 10:** Typical fillet geometry in T-peel specimen
Figure 11 illustrates the stress distribution derived when talking into account the fillet, and the comparison to the theoretical stresses occurring in an ideal T-peel joint subjected to the same tensile load.

In order to enable a common critical zone size for all specimen configurations, the stress distribution for an equivalent T-peel specimen geometry with no fillet (s. Figure 11) has been assumed to apply. Suggesting the maximum principal stress as appropriate equivalent stress measure, critical zone distances larger than 2 mm will result in rapidly decreasing stress average due to the existence of compressive stresses besides the peel front caused by substrate rotation and thus, to reasonable stress average values.

Although empirical, the use of the equivalent T-peel geometry has enabled the fatigue data from varying test geometries to be fitted to the S-N curve of the adhesive material (as evaluated in butt-jointed hollow cylinders testing) using a single distance (zone length) for averaging stresses to the corresponding effective stress measure (Figure 12).
The agreement for all test joints to the adhesive bond fatigue properties is good although the scatter from lap shear and T-peel specimens is as expected larger. This may be to some degree attributed to tolerances inherent to substrate and bondline manufacturing, while butt-jointed specimens provide consistent bondline quality and controlled loading conditions.

A zone length of 2.65 mm was found to apply to joint fatigue assessment of the investigated toughened structural adhesive. Comparable measures were evaluated for other adhesive grades in subsequent test work or university research programs.

**Validation**

For validation of the proposed effective stress-life approach, records from box beam comparative fatigue testing have been used. Good prediction can be achieved as shown in comparison to the test results in Figure 13.

![Box beam torsional fatigue test results](image)

*Figure 13: Box beam torsional fatigue test results at stress/deformation ratio R=-1, and prediction by means of the proposed effective stress-life approach*

**Industrial Application**

In large structures, the stress-averaging zone has to be chosen as a whole element for numerical convenience, i.e. the element length must correspond to the desired zone distance.

The proposed effective stress-life approach has been applied to fatigue life prediction for industrial components by the IFAM Fraunhofer Institute, Bremen, within the frame of the RelBond reasearch project funded by the Henkel Corporation [27]. Adhesive bonded joints in a wheel house tested at constant amplitude with a load ratio of R=-0.7 have been compared to FE analysis, and the stress-life prediction based upon the effective stress approach. Figure 14 illustrates reasonable correlation of the theoretical prediction and the fatigue life resulted by experiments.
Variable Amplitude Loading

Structures and components in service experience fatigue spectra that are irregular and cannot be represented by constant amplitude S-N curves. A popular cumulative damage method to predict variable amplitude fatigue by means of S-N data is the Palmgren-Miner rule. The application of this method is subject to current research activities.

Summary

Besides stiffness benefits, structural adhesives offer unique advantages under fatigue loading conditions due to their ability to transmit loads evenly and to avoid stress concentrations. Despite the progress in the use of fracture or damage mechanics, stress-based concepts for fatigue life prediction still have instinctive appeal for engineers because of their direct approach and their applicability in industrial design work.

The paper presents an effective stress-life approach to predict fatigue properties of adhesive joints bonded by means of structural adhesives. The equivalent stress conditions that may be used are able to describe multiaxial strength behaviour. The approach is based upon effective stress as average stress over a critical zone. The paper demonstrates transferability of fatigue data evaluated using standard specimens, to bonded structures and components tested in fatigue.

Acknowledgement

The test campaign leading to the present paper was part of a corporate research project carried out at the Henkel Engineering Center in Garching, Germany. The author would like to gratefully acknowledge Jürgen Becher, Daniel Reif, Renate Kreuzer, Gerhard Zill and Klaus Busl for their contribution. The application of the effective stress method to industrial components was part of the Relbond
Research Project funded by the Henkel Corporation. The relevant work package has been managed at the IFAM Fraunhofer Institute in Bremen, Germany by Christof Nagel.

References


