

A Fracture Mechanics Approach to Durability Calculations for Adhesive Joints

Peter Heyes
HBM UK, Ltd.

Gunnar Björkman
Volvo Technology

Andrew Blows and Tim Mumford
Jaguar/Land Rover Cars

Paul Briskham
Coventry University

ABSTRACT

Effective use of adhesive bonding in automotive vehicle bodies requires analytical methods for durability, so that potential fatigue problems and unnecessary overdesign may be eliminated before the physical prototype stage and release of product with unquantified safety factors avoided.

This paper describes a fracture mechanics-based method for predicting the durability of adhesive joints, based on work previously carried out at Volvo [1]. The method requires relatively modest modifications to a typical vehicle body FE mesh. Adhesive bonds are represented by bar elements around the periphery of each bond. Grid point forces from shell elements adjacent to the adhesive bond are recovered and used to determine line forces and moments at the edge of the glued flange. These forces and moments are then transferred to an analytical sandwich model of the joint. This enables approximate calculations of the strain energy release rate (or rather the equivalent J-integral) to be made for assumed small cracks at the edge of the adhesive. Analytical results are compared with calculations based on detailed FE meshes for typical test specimen geometries, and with physical test data. J-integral values calculated for different geometries and loadings are shown to be able to correlate the durability of joints, particularly at longer lives, enabling useful estimations of joint durability to be made.

CITATION: Heyes, P., Björkman, G., Blows, A., Mumford, T. et al., "A Fracture Mechanics Approach to Durability Calculations for Adhesive Joints," *SAE Int. J. Mater. Manf.* 5(1):2012, doi:10.4271/2012-01-0731.

INTRODUCTION

Adhesive bonding is increasingly used in the development of lightweight vehicle bodies. In addition to improving structural rigidity, it provides a means of joining high strength steels, aluminium alloys and dissimilar materials, with good durability against fatigue compared to more traditional methods (spot-welding and seam-welding). As with any manufacturing technology, if full advantage is to be taken of its benefits, analytical methods are required in order to be able to evaluate and optimize the durability of the resulting structure. Without these analytical methods, and in an environment where development times are very short,

release of product with unnecessary over-design and/or unquantified safety factors is inevitable.

Ideally, adhesive joints should be designed so that they are loaded primarily in shear (rather than peel or cleavage modes), and in addition to having a uniform thin layer of adhesive, fillets are highly desirable in order to minimize stress concentrations at the periphery of the joint. In contrast, adhesive joints in mass-produced automotive vehicle bodies will in general experience complex loadings including elements of shear and peel, and may be underfilled (primarily for cosmetic reasons). Under sufficient static or cyclic loading, such joints are likely to fail by crack propagation through the joint. Cracks may be interfacial or cohesive, or a

mixture of these failure modes may occur, depending on factors such as the choice of adhesive, surface pretreatment and surface cleanliness prior to adhesive application. See Figure 1.

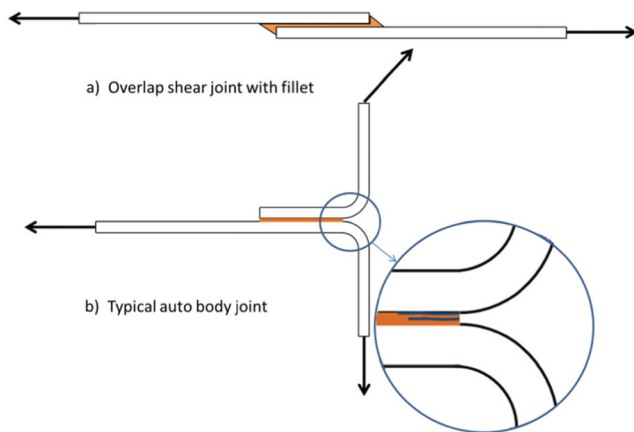


Figure 1. Adhesive joints: Figure 1(b) is more typical of an automotive vehicle body.

This paper describes a method for evaluating the durability of such adhesive joints in vehicle bodies, based on work previously carried out at Volvo by Björkman [1]. It is based on the following premises:

- Because of the nature of the joints and their manufacturing process, their durability is best described using a fracture mechanics approach.
- Stress intensity factors are not particularly useful for cracks in inhomogeneous bodies such as an adhesive joint, or for interfacial cracks. The loading on a crack is therefore evaluated using the strain energy release rate, or rather since we are considering only limited plasticity around the crack tip, we can use the equivalent J-integral.
- Any calculations must be capable of being carried out on a typical body durability model, with minor modifications only. In the body CAE process, detailed FE modelling of the type normally used to carry out J-integral calculations is out of the question, so an approximate method is required.
- Several factors conspire against making accurate life estimations - the imprecise nature of the manufacturing process, rather flat SN curves (steep crack growth curves) for the adhesive, and the fact that the J-integral is a scalar quantity. For this reason, in the interests of making some useful calculations, the method described here is limited to the determination of maximum J-integral values and comparison with a threshold value. Finite life predictions are not attempted.

The calculation process is illustrated in Figure 2.

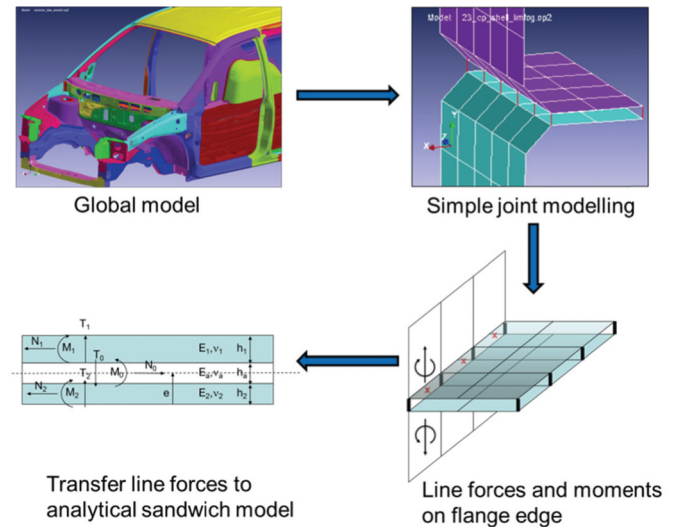


Figure 2. Summary of calculation process

The basic steps of the method are as follows:

1. A global finite element analysis of the structure is carried out. A simple, relatively coarse representation of the joints is used in which beam elements represent the adhesive joining the flanges.
2. Line forces and moments along the edges of the flanges are derived from the FE results, combined with applied loading histories.
3. The line forces and moments are applied to an analytical “sandwich” model of the bonded flange, and used to calculate the strain energy release rate G (actually the J-integral) based on an initial small crack in the adhesive.
4. At each point along the edge of the flange, the calculated maximum J-integral during the loading history is compared to the threshold required for crack growth G_{th} to determine whether or not the joint is likely to fail.

The method does not calculate fatigue life, but in addition to the maximum value of the J-integral for a given flaw size, an estimated reserve factor may be calculated.

THEORETICAL BACKGROUND

MODELLING GUIDELINES AND FE ANALYSIS

FE models for adhesive joint durability calculations require only minor changes from typical automotive body models. The modelling guidelines may be summarized as follows:

- Models should be created using linear shell elements positioned at the mid-planes of the sheet metal parts.

- Each adhesive bond should be represented by beam elements around the periphery of the bond, connecting the shell meshes of the joined parts. Typically beams will have a 5mm circular cross-section, with modulus 3000 MPa and Poisson ratio 0.4.
- Bonded flanges require more or less congruent and parallel meshes so that the beams are normal to the flanges.
- Beam elements should be as close as possible to the position of the edge of the adhesive in the bond, which means that ideally the bend radius adjacent to the joint is represented by at least one row of shell elements as in the “preferred” method illustrated in Figure 3.

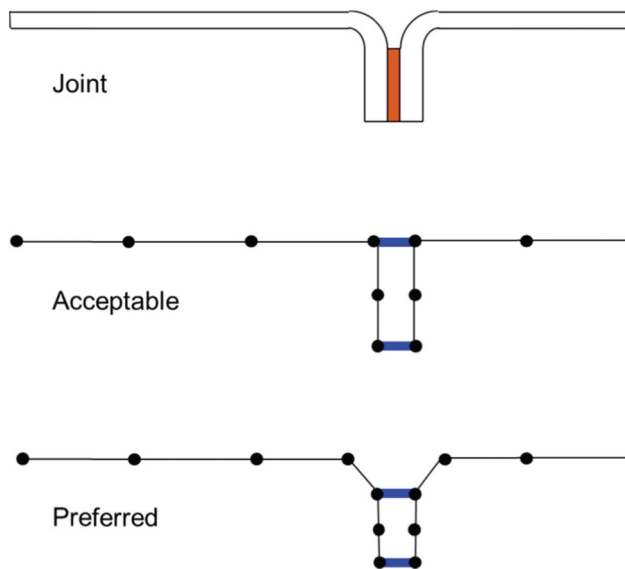


Figure 3. Mesh cross section - preferred modelling practice.

Any offset between the position of the beam elements and the actual edge of the adhesive (e.g. if the “acceptable” method in Figure 3 is used) will result in errors in the forces and particularly the bending moments at the edge of the joint. The method compensates for this when it estimates the bending moments at the crack tip, but any offset should be minimized for best accuracy.

The purpose of the FE analysis is to provide information that can be used to derive time-histories of the J-integral for a small crack at various points along the edge of the adhesive bond. An intermediate step is the derivation of time-histories of force and moment per unit length (line forces and moments) applied to the “sandwich” (flanges plus adhesive) by the adjacent elements. These time histories can be generated by linear superposition (e.g. by combining static unit loadcase results and corresponding load-time histories), direct from FE time-steps, or from a transient analysis, whether a direct transient or using a modal superposition approach. In any case, the data required from the FE analysis are the nodal forces and moments from the shell elements adjacent to the flanges.

At each calculation point, the nodal forces and moments in the shells adjacent to the bonded joint are used to determine line forces and moments on the upper and lower elements of the joint, in a local co-ordinate system.

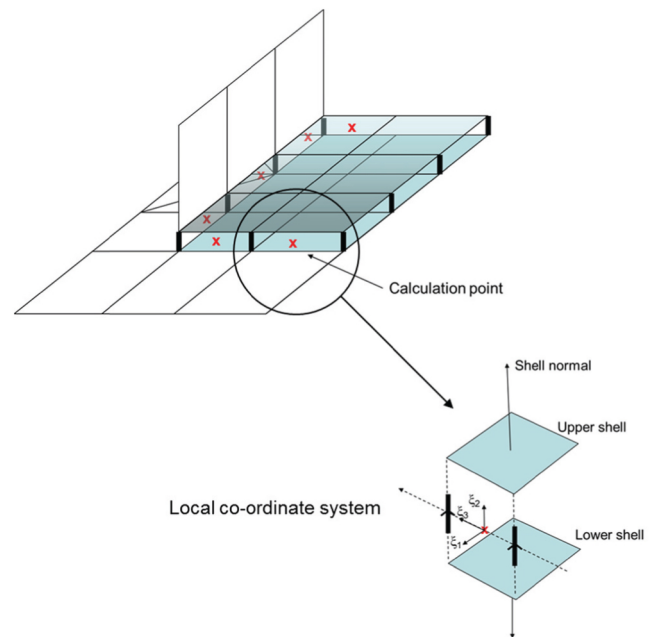


Figure 4. Calculation points and local co-ordinate system

The method for determination of the line forces and moments is based on that originally proposed by Fernér et al [2] for determination of structural stresses at the toe of a seamweld, and refined by Heyes in nCode DesignLife [3].

J-INTEGRAL CALCULATION

The concept of using a sandwich model to calculate J-integrals and predict fracture has been described by Fernlund et al [4]. This idea was developed further by Björkman [1], working at Volvo, who extended its application by transferring loads from a shell model, and critically, improved the accuracy of the calculation through the introduction of a distortion term, as outlined below.

The basic premise of the method is that the strain energy release rate G associated with a small crack in the periphery of an adhesive bonded joint in a vehicle body can be determined by transferring the line forces and moments along the edge of the representation of the bond in a shell FE model of the body to an analytical “sandwich” model of the joint, which is used to calculate G . In practice, what is actually calculated is the equivalent path-independent J-integral as described by Rice [5]. There are two assumptions implicit here:

1. In order for the strain energy release rate G to be equivalent to the J-integral, any yielding must be limited to a small region around the crack tip. This is reasonable when

considering the endurance domain; widespread plasticity would give rise to low-cycle fatigue failure.

2. Cracks are not large enough to cause significant redistribution of load around the adhesive joints.

For convenience, the J-integral is evaluated along a path just before and just after the crack front, as illustrated in [Figure 5](#).

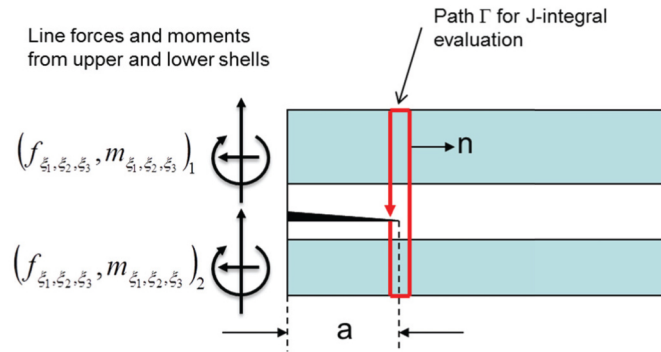


Figure 5. Transfer of line forces and moments to sandwich model and J-integral evaluation path

The basic inputs to the analysis are:

$(f_{\xi_1, \xi_2, \xi_3}, m_{\xi_1, \xi_2, \xi_3})_1$: the line forces and moments on the upper shell

$(f_{\xi_1, \xi_2, \xi_3}, m_{\xi_1, \xi_2, \xi_3})_2$: the line forces and moments on the lower shell

These are used to estimate the cross sectional line forces and moments (N,M,T) in the upper and lower parts of the sandwich just before the position of the crack front, and in the sandwich just beyond the crack front, as illustrated in [Figure 6](#). The effective total crack depth a is the sum of the distance (offset) from the position of the beam elements in the shell model to the edge of the adhesive plus the actual flaw depth.

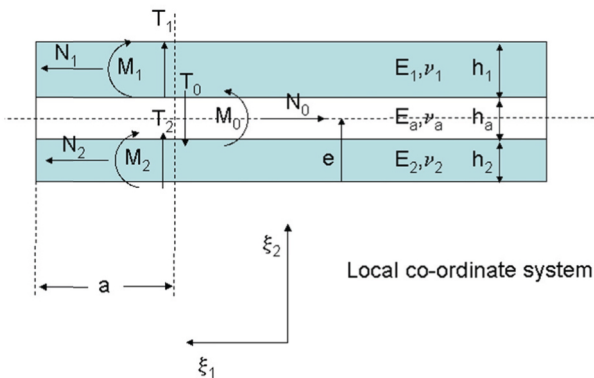


Figure 6. Sandwich model schematic

The total strain energy release rate is then computed from the J-integral. The J-integral is evaluated along a path which has non-zero contributions from three sections, corresponding to the top sheet, and bottom sheet just behind the crack front, and the sandwich just ahead of the crack front. These are denoted by subscripts 1,2 and 0 respectively in the following equations.

First the modulus and effective thickness of the sandwich are defined:

$$E_0 = E_1 \quad (1)$$

$$h_0 = h_1 + \frac{E_a}{E_1} h_a + \frac{E_2}{E_1} h_2 \quad (2)$$

Then additional moduli, for $i = 1,2$:

$$\bar{G}_i = \frac{E_i}{2(1-\nu_i)} \quad (3)$$

$$Q_i = \frac{E_i}{1-\nu_i^2} \quad (4)$$

...and for $i = 0$:

$$G_0 = G_1 \quad (5)$$

$$Q_0 = Q_1 \quad (6)$$

Then the distance from the bottom to the centroid of the sandwich

$$e = \frac{\left(\frac{h_2}{2}\right)\left(\frac{Q_2}{Q_1}\right)h_2 + \left(h_2 + \frac{h_a}{2}\right)\left(\frac{Q_a}{Q_1}\right)h_a + \left(h_2 + h_a + \frac{h_1}{2}\right)h_1}{\left(\left(\frac{Q_2}{Q_1}\right)h_2 + \left(\frac{Q_a}{Q_1}\right)h_a + h_1\right)} \quad (7)$$

Second moment of area for $i = 1,2$

$$H_i = \frac{h_i^3}{12} \quad (8)$$

...and for $i = 0$: