

Whitepaper

Fatigue Analysis of Seam Welded Structures using nCode DesignLife

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Introduction

Welding is used in many industries as an effective and economical method for making structural joints between metal parts. However, the nature of the welding process means that welded joints generally have a fatigue strength that is inferior to that of the parts being joined together. At the same time, welds often tend to be made at geometric features or changes in section in the structure. The result of these facts is that even in a well-designed structure, it is typically the welded joints that are most likely to fail by fatigue. Any evaluation of the durability of a welded structure must therefore place a high priority on a fatigue assessment of the welded joints.

A number of features have been implemented in nCode DesignLife to facilitate the fatigue analysis of welds, and there is an on-going program of development to improve these and extend their range of applicability. This paper outlines the main methods implemented, and provides some background and validation cases.

Fatigue of Welded Structures

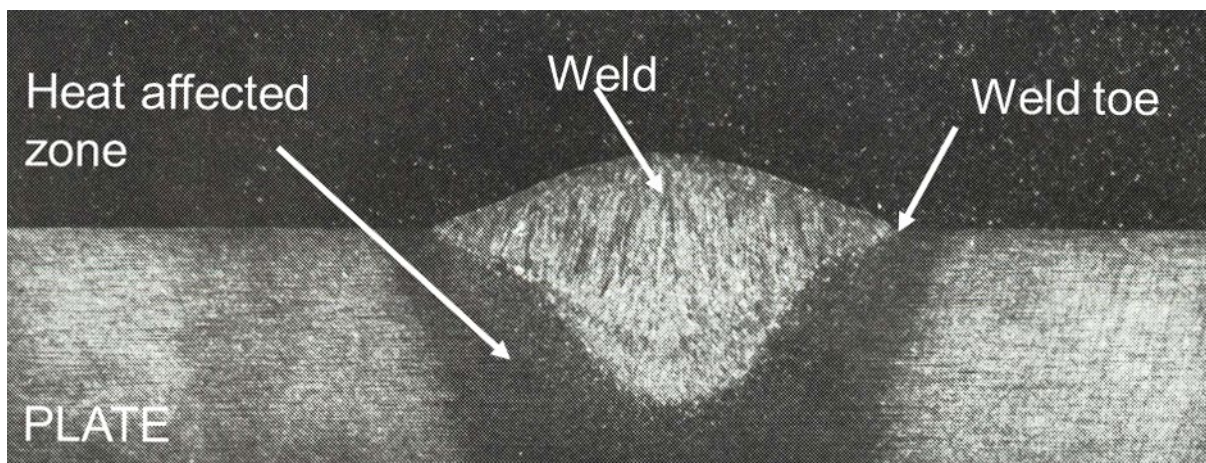


Figure 1: Weld cross section showing fusion (weld) and heat affected zones.

The fatigue strength of welded joints is in general significantly less than that of the parts which are welded together, or of the “parent plate”. There are a number of reasons for this:

- The geometry of a weld normally gives rise to a stress concentration (unless it is a butt weld, ground flat). The stress will typically be highest at the toe or root of the weld, and the shape in this area may not be well controlled.
- The welding process will very often produce defects which can act as crack initiation sites – slag inclusions, incomplete fusion, porosity etc. There is plenty of evidence that the fatigue life of welds is often dominated by crack growth.
- Around the fusion zone, there is a heat affected zone (HAZ) where the parent material has been heated to a high temperature and allowed to cool fairly rapidly. This may cause major changes to the microstructure and properties in this region – over-riding any previous heat treatment for example.
- The welding process will introduce residual stresses, which may be of the order of the yield strength of the material.

All these factors cause the fatigue strength of a welded joint to be quite different from that of the parts it joins together. It is therefore not reasonable to expect to be able to make good fatigue predictions of a joint based on the properties of the plates or other parts being joined. It is for this reason that the majority of historic methods and standards are based on a characterisation of the fatigue behaviour of whole joints, normally in the form of S-N curves. These curves effectively incorporate all the effects of defects, unknown residual stresses, notches, and material property changes that are introduced when the weld is made. The main features of this type of approach are outlined in the next section.

Standard Methods

There are a number of different standards that describe methods for fatigue assessment of welded joints, e.g. Eurocode 3[1], the ASME Boiler and Pressure Vessel Code[2], the Swedish Regulations for Welded Steel Structures[3] and British Standard BS7608[4]. These are rather similar in approach. A very similar approach for aluminium structures is described in BS8118[5]. For the sake of brevity, just the main features of the BS7608 welded approach are described here.

Note that BS7608, "Code of practice for fatigue design and assessment of steel structures", is very closely based on BS5400 Part 10, "Code of practice for fatigue design of steel, concrete and composite bridges" [6]. Its origins in civil engineering are obvious.

- Material specification does not feature heavily in this standard, beyond noting that it applies to structural steel with a yield strength of less than 700 MPa. Many researchers have noted that while the fatigue strength of steels can vary widely with composition and heat treatment, once welded, the strength of the resulting joints falls within a single scatter band. This is very convenient as it allows the same design curves to be used for a range of materials.
- Welds are classified based on the type of joint, the geometry, the loading direction and the potential failure direction. For example a full penetration weld of the shape illustrated in Figure 2, loaded in the direction of the arrow and with the assessed crack location being at the weld toe would be in class F.

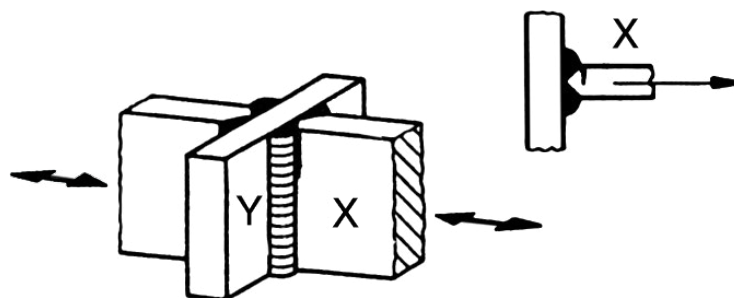


Figure 2: Weld classified as "F" according to BS7608

- Each weld class is associated with an S-N curve. The S-N curves for the BS7608 weld classes, taken from the nCode material database, are illustrated in Figure 3 below. These are the mean or B50 life curves. A designer would typically use a -2 standard deviation design curve corresponding to a 97.7% certainty of survival.

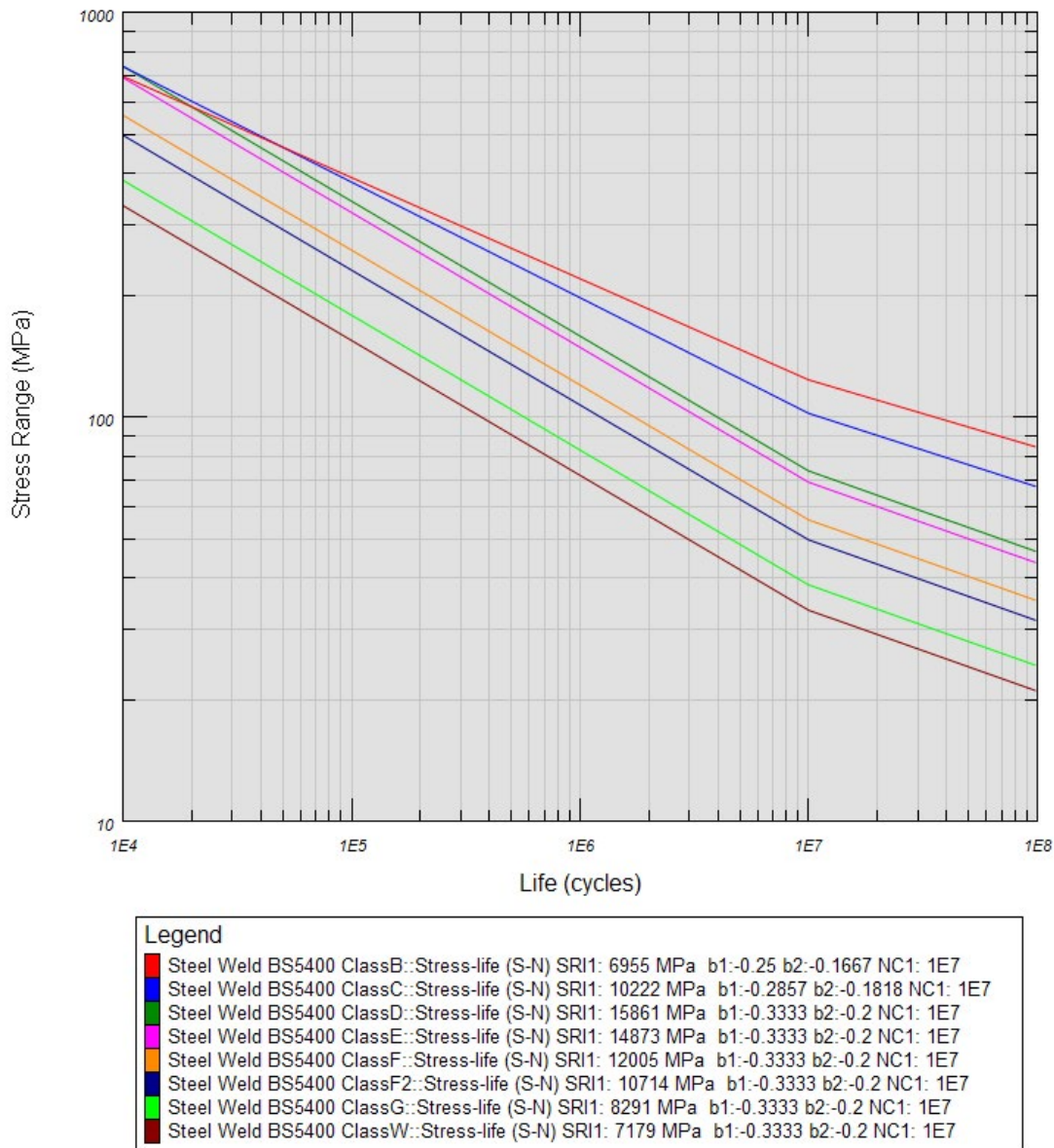


Figure 3: S-N curves for the BS7608 weld classes

- Note that the S-N curves have a change of slope at 10^7 cycles, corresponding with the fatigue limit. The lower part of the curve is used for variable amplitude loadings, in conjunction with rainflow cycle counting and Miner's rule, if the fatigue limit is exceeded at some time during the time history. This type of approach is a common way of capturing the fact that small cycles may have a damaging effect if larger cycles are able to overcome the barriers to crack growth. (If you are a DesignLife user you can reproduce this effect by setting the property "SmallCycleCorrection = BS7608" on the analysis engine).

- The location and nature of the stress that must be used for the fatigue calculation is defined in the standard for each case, but typically is based on the principal stress with the largest range, and may be a nominal stress or more often a “hot spot” or “structural” stress adjacent to the weld toe. For practical fatigue calculations based on FE, the determination of this stress is one of the main challenges. The basic idea is to determine a stress at the location of the weld toe, but excluding the effect of the local stress concentration due to the weld geometry detail. For a simple case with a combination of applied load and uniform bending moment, as illustrated in Figure 4, the structural stress may be simply calculated as the sum of the membrane and bending stresses in the plate adjacent to the weld.

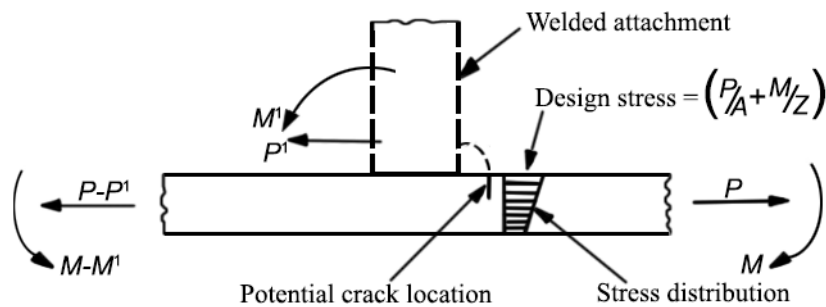


Figure 4: Calculation of structural stress in plate adjacent to weld.

If the geometry and/or stress distribution is more complex a strategy such as that illustrated in Figure 5 may be required.

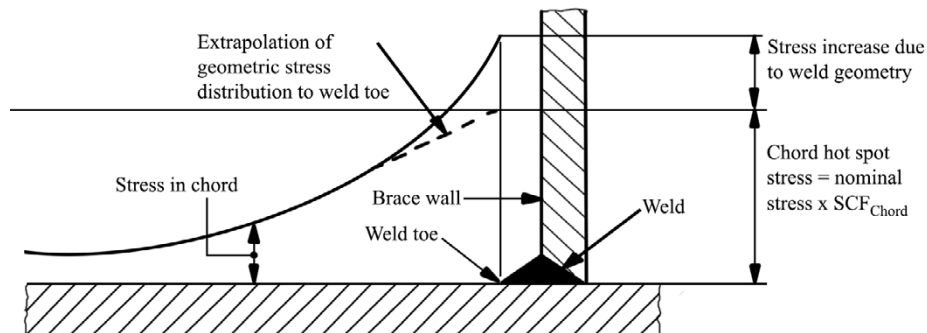


Figure 5: Determination of structural stress by extrapolation

The required stress (“Chord hot spot stress” in Figure 5) is determined by extrapolating the stress in the plate to the weld toe, but excluding stresses from locations close enough to the weld toe to be influenced by the detail of the weld toe geometry. In BS7608 stresses are extrapolated from stresses $> 0.4t$ from the weld toe, where t is the sheet thickness. Note that the standard also allows determination of stresses by experimental means, based on strain gauges placed with an edge $0.1t$ from the weld toe. See Figure 6.

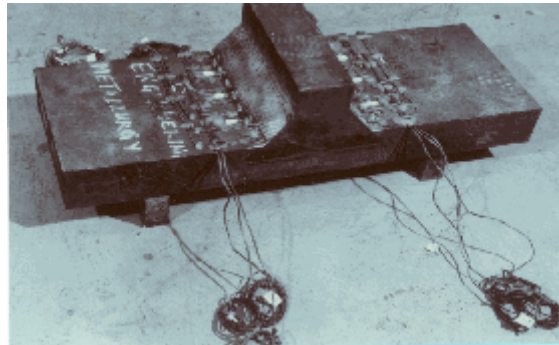


Figure 6: Experimental method of stress determination at weld toe.

In fact, the means by which stress is recovered from the finite element solution is a central issue for all weld fatigue analysis methods and standards.

- The welding process inevitably introduces residual stresses into a structure, which may be of the order of the yield stress. Since these stresses are normally unknown, standards typically treat them as an element of the scatter which is built into the S-N curves. BS7608 specifically excludes the use of any mean stress correction in the actual fatigue analysis, unless the residual stress is known, or the joint has been effectively stress relieved, in which case the compressive part of each cycle may be reduced to 60% of the calculated value. (Note that in DesignLife this correction can be reproduced using the FKM mean stress correction[7], with the following modifications to the standard S-N curves: setting the R-ratio of test to 0, and the mean stress parameters M1=M3=M4=0 and M2=-0.25)
- There is a size effect in fatigue, which definitely applies to welded structures, with thicker structures failing at lower stresses than thinner ones. There are a variety of reasons for this effect, including statistical reasons, stress gradients, manufacturing processes and residual stresses. In common with other standards, BS7608 has an empirical method for modelling this effect, reducing the fatigue strength for thicknesses above a reference value according to:

$$S = S_B \left(\frac{t_B}{t} \right)^{0.25}$$

where the subscript B denotes the reference thickness. In BS7608 the reference thickness is 16 mm, but studies such as that by Gurney[8,9] suggest that thickness corrections might be extrapolated to much thinner sheets, allowing the additional fatigue strength to be exploited.

The main features of BS7608 described here give a good indication of the issues that need to be addressed by someone designing or analysing a welded structure. Note that this standard does not address “finessing” the design of individual welds through close control of the welded geometry, stress relieving etc. For such cases a local approach such as described in[10] may be more productive.

Fatigue calculations based on various weld standards can be readily made using nCode DesignLife, where a number of features have been introduced to make this possible, including BS7608 and Eurocode S-N curves, small cycle and mean stress correction methods. The main challenge for an engineer wishing to make standard calculations based on a finite element is the extraction of a suitable stress parameter from the FE results. However, quite successful calculations have been made using fairly simple modelling strategies. For

example a weld joint may often be represented by a simple connection between sheets of shell elements. The most important thing is to ensure that there is a row of suitable stress recovery points – e.g. element centroids – at the position of the weld toe. The elements adjacent to the joint may be selectively thickened to better represent the stiffness of the weld. See Figure 7.

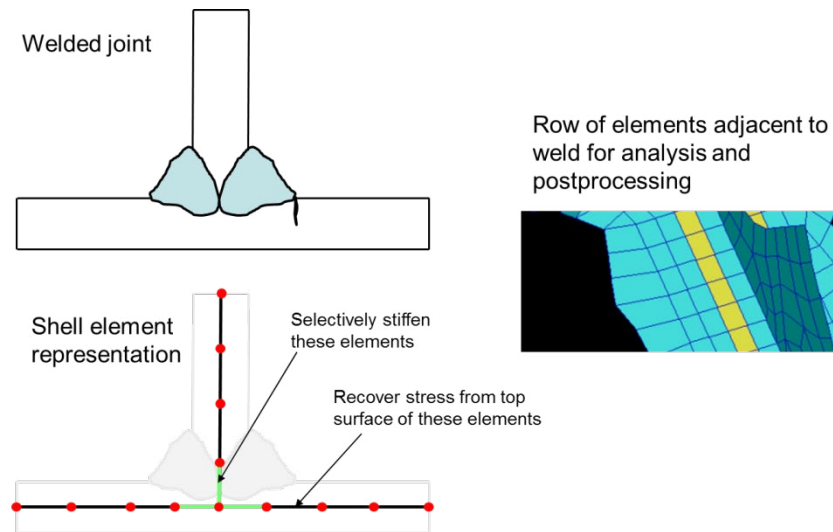


Figure 7: Simple modelling strategy for application of BS7608

“Volvo” Method

Standard methods such as described in BS7608 have not proved very popular for automotive applications, for a number of reasons:

- They were mainly developed for thick sheet structures, whereas the majority of automotive welds joint sheets of thickness 3 mm or less.
- Weld classification systems designed for engineering structures such as bridges and pressure vessels may be difficult to apply to many automotive structures due to the combination of weld geometry and complex loading.
- They do not always lend themselves to use in conjunction with a usually fairly unrefined (and almost always unconverged) finite element model such as is typically used for automotive body analysis.

These issues were the motivation behind a new method developed at Chalmers University at the behest of Volvo Car Corporation. The aim was to develop a software-friendly method suitable for making FE-based fatigue assessments of welded joints in typical automotive structures, subject to complex loadings, with the minimum of user intervention being required. The basic concept of the method, as described by Fermér et al in Reference[11] was based on the approach described for spot-welds by Rupp et al in Reference[12]. This method has been adopted and subsequently developed by nCode in conjunction with Volvo, and has been used successfully by automotive OEMs and suppliers. For information about typical application and performance, see Reference[13]. The method as implemented in software is described in detail in the nCode DesignLife Theory Guide [14], so it is just summarised briefly here.

- **Modelling strategy:** The method requires a simple modelling strategy for the seam welds, with each weld bead being represented by a single or double row of shell elements, as illustrated in Figure 8. No weld classification is required beyond identifying welds as fillet or overlap joints, and placing the weld elements in suitable groups or property sets. This allows the appropriate elements for analysis to be automatically identified.

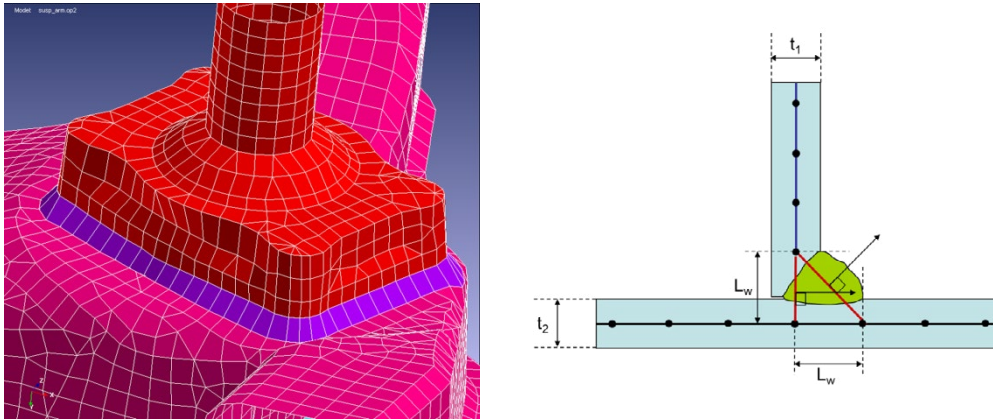


Figure 8: Weld modelling strategy for "Volvo" method.

- **Stress recovery:** In the original concept, the structural stresses at the weld toe – in essence, the same combination of membrane and bending stresses illustrated in Figure 4 – were calculated based on the nodal forces and moments acting in the weld toe or weld root elements. These are the elements adjacent to the weld elements (red in right hand part of Figure 8). In practice, there are a number of different ways to calculate the structural stresses needed for the fatigue calculation – this matter is discussed in more detail in the following section.
- **Bending ratio:** The structural stress used for making a weld toe fatigue calculation is the sum of the bending and membrane stress (see Figure 9). However, the results of extensive testing indicate that the fatigue strength is significantly greater for "flexible" joints where the stresses are predominantly as a result of bending as opposed to "stiff" joints where the majority of the contribution comes from membrane stresses. The method relies on defining a bending ratio which is the fractional contribution of bending to the overall stress.

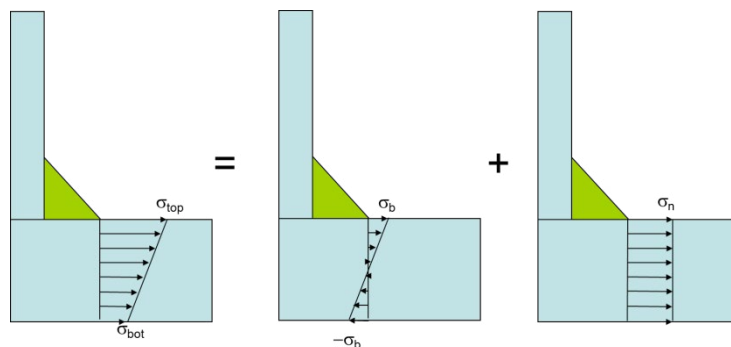


Figure 9: Top surface stress is a combination of membrane and bending stress.

The bending ratio is defined as:

$$r = \frac{|\sigma_b|}{|\sigma_b| + |\sigma_n|}$$

For a complex loading the bending ratio is determined as a weighted average over the time history of loading.

- **Material properties:** Weld fatigue performance is described by a pair of S-N curves which represent the fatigue strength of a weld under pure membrane (stiff) and bending (flexible) loading conditions. An interpolation is made between the curves based on the bending ratio at each calculation point. The more pessimistic or "stiff" curve is used until the bending ratio exceeds a threshold, and thereafter the curves are interpolated. These S-N curves may be generic for typical steels, but more refined results might be obtained if the user has data for specific materials. The generic curves shipped with the software are optimised for variable amplitude loading.

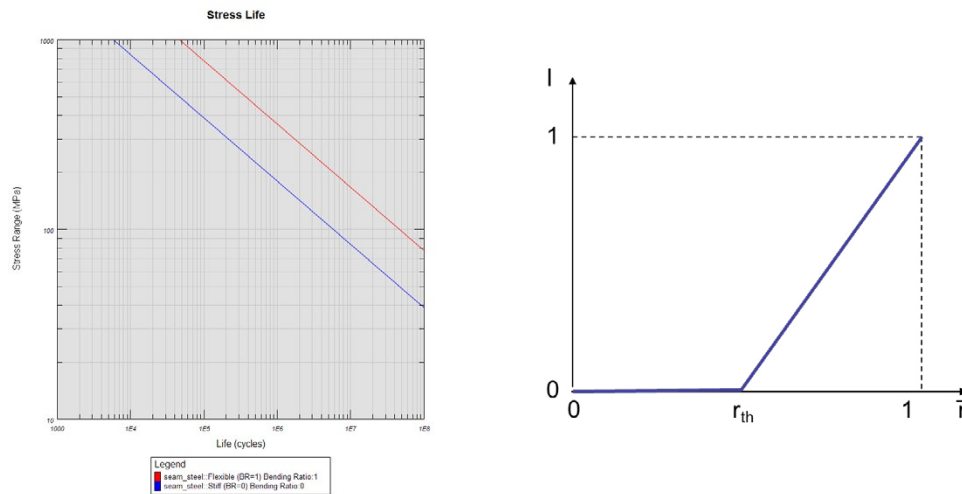


Figure 10: Generic weld S-N curves and interpolation function

- **Size effect:** There is a size effect included, similar in principle to that in BS7608 for example. If a reference thickness is exceeded, the fatigue strength is reduced by a factor:

$$factor = \left(\frac{T_{ref}}{r} \right)^n$$

Typical values of the reference thickness and the exponent are 1 mm and 1/6.

- **Mean stress effect:** There is an optional mean stress correction which uses the FKM approach in which the mean stress sensitivity is defined in 4 regimes. The effects of applied mean stresses are thought to be more important for thin sheet structures than for thick; in thinner structures, residual stresses due to the welding process are likely to be at least partly relieved by distortion.

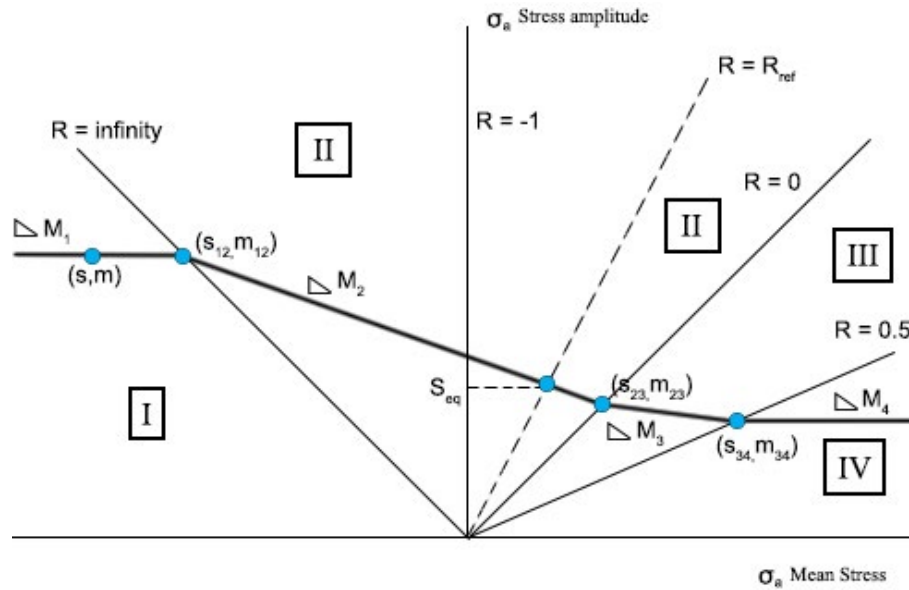


Figure 11: FKM mean stress correction (constant life diagram)

DesignLife can easily be configured to simulate the effect of in-service or test loads on welded structures. Figure 12 below shows a simple “flow” calculating fatigue damage on a welded rear suspension arm, using BS7608 and “Volvo” methods. Note that fatigue calculation results (e.g. damage or life) may be post-processed on the elements adjacent to the weld, allowing likely failure locations and lifetimes to be readily identified.

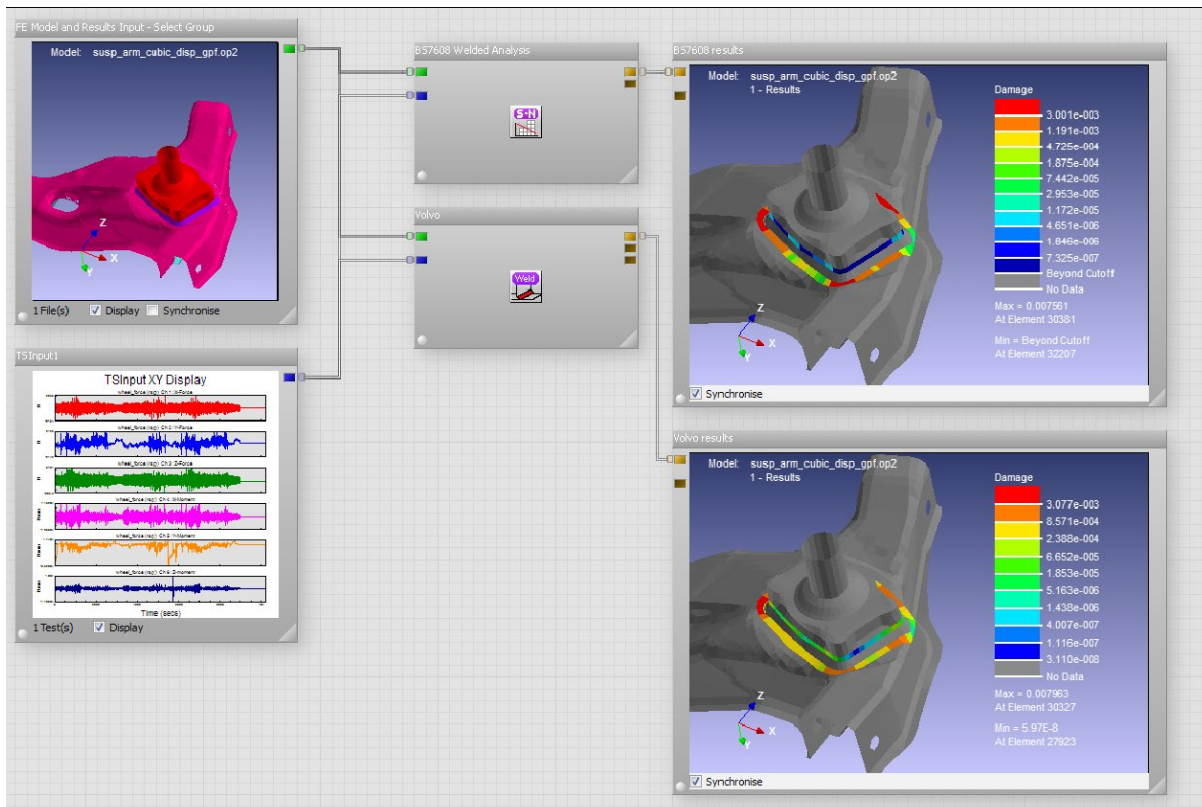


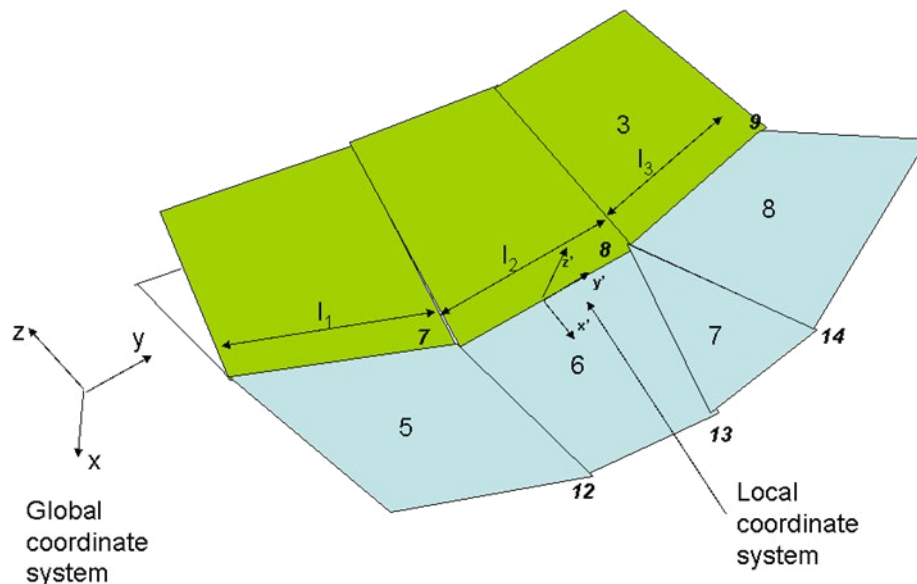
Figure 12: Fatigue analysis of welds using “Volvo” and BS7608 approaches in nCode DesignLife.

One of the key issues for anyone attempting a fatigue analysis on a welded structure concerns the recovery of stresses from the FE model. The “Volvo” method described in the previous section requires the structural stress (membrane and bending) at the weld toe. Most automotive welded structures will be modelled with thin shell elements, but there are a number of ways in which the stress can be calculated, with variations depending on the element formulation.

FE Modelling and Stress Recovery Methods

Thin Shells

- **Using conventional stress recovery methods:** The Volvo method requires stresses at the edge of elements, which means they will be extrapolated from the integration points. Because of the geometric features of seam welds, mesh convergence is not really possible, so conventional node-at-element stresses tend to be inconsistent and highly sensitive to the mesh density, mesh quality and element formulation.



$${}^6\sigma_{top,normal} = \frac{{}^6f_{x'}}{t} + 6\frac{{}^6m_{y'}}{t^2}$$

- **Based on grid point forces:** This is the original method proposed by Fermér et al in 1998 [11]. This has been refined in DesignLife so that the structural stress at the mid-point of the edge of each weld toe element is determined based on the line forces and moments at the weld toe, as illustrated in Figure 13.

Figure 13: Calculation of structural stress at weld toe from line forces and moments.

The line forces and moments are in turn calculated from the grid point forces in the weld toe elements. There is no such thing as a mesh-insensitive finite element method, but this approach does show considerably reduced sensitivity to mesh size and element shape or formulation, especially around the weld ends.

- **CUBIC stress recovery:** For users of NASTRAN there is another option which gives very similar performance to the grid point force approach. The CUBIC stress recovery option in NASTRAN is not based on extrapolation from the integration points, but rather on the relative displacements and rotations of the nodes. The element is used rather like a strain gauge rosette, with cubic displacement functions being fitted to the calculated displacements and rotations to determine strains and hence stresses at the corners.

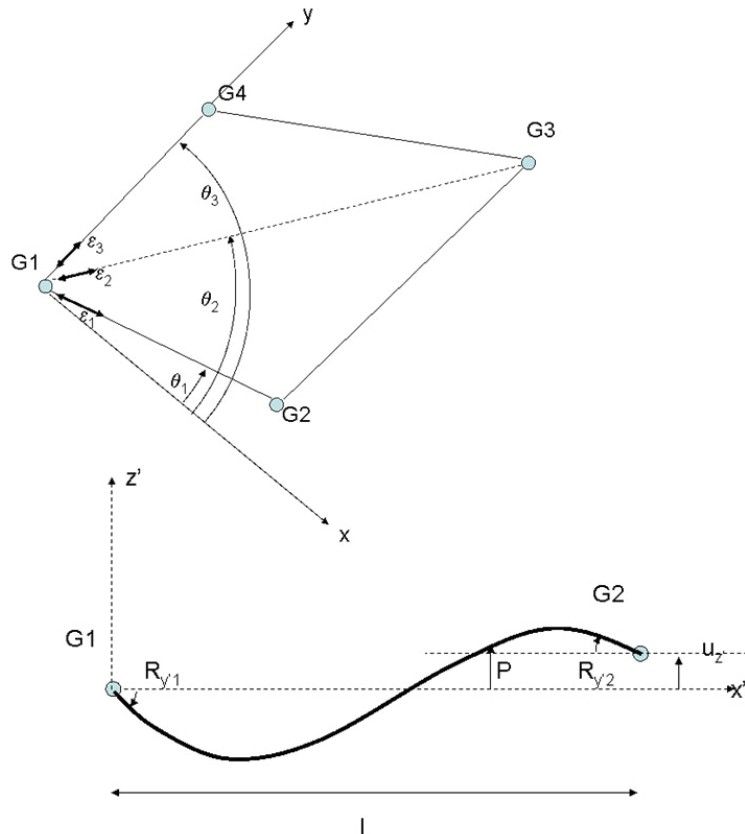


Figure 14: Stress recovery based on displacements using CUBIC displacement function

The CUBIC method is not available in most FE codes, but for users of other codes, there is an option in DesignLife that can derive stresses from nodal displacements and rotations. This gives very similar results to the NASTRAN CUBIC option, but does not perform very well with warped elements.

Solid elements

However, there are some structures that cannot be adequately modelled with thin shell elements and require more detailed solid models. This is actually not about whether the structure is “thick” or “thin”, but rather whether the shape of the structure – in particular whether there are geometric features on a similar scale to the sheet thickness – violates the basic assumptions associated with thin shell elements. For example, in the 6 mm T-tube specimen from the TWI study [9] illustrated in Figure 15, the proximity of the weld to the corner of the horizontal tube means that this region cannot properly be represented with thin shell elements.

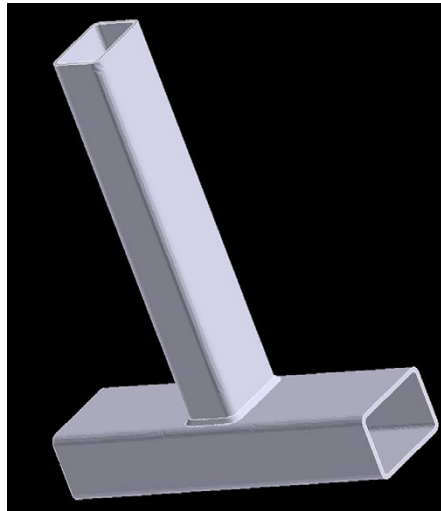


Figure 15: TWI T-tube specimen

The problem this presents is how to determine a structural stress from a non-linear stress distribution through a solid model. The method now implemented in DesignLife is similar in principle to that described by Niemi in Reference [16]. In this approach the stress distribution through the thickness of a structure adjacent to a weld is partitioned into membrane, linear bending and non-linear peak stress distributions, as illustrated in Figure 16.

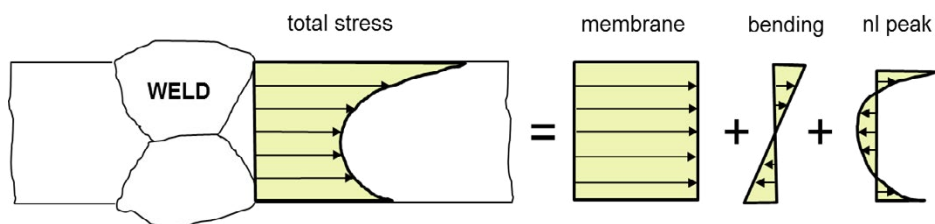


Figure 16: Partitioning total stress distribution at weld toe.

The membrane stress is simply the average stress at the weld toe, while the bending component is chosen so that the remainder – the non-linear peak stress – is in equilibrium. The structural stress is then the sum of the membrane and bending components. A similar approach is described in the ASME pressure vessel code [2]. In DesignLife, the membrane and bending components are determined by through thickness integration of stresses along a line projected down from the weld toe:

$$\sigma_m = \frac{1}{t} \int_0^t \sigma \, dx \quad \sigma_b = \frac{6}{t^2} \int_0^t \sigma \left(\frac{t}{2} - x \right) dx$$

This method provides a reliable and relatively mesh-insensitive method for determining the structural stress. An example is illustrated in Figure 17. Here a weld detail has been modelled using tetrahedral and hexahedral elements. Due to the stress singularity, the peak stresses are very mesh dependent (and in fact meaningless as a result); the peak stress from the ten-noded tetrahedral mesh is 1612 MPa compared to 695 MPa from the hexahedral mesh. The linearised peak stresses on the other hand, as determined by DesignLife, compare very well at 569 MPa and 553 MPa. In general results are very good so long as there are at least 3 layers of solids through the thickness.

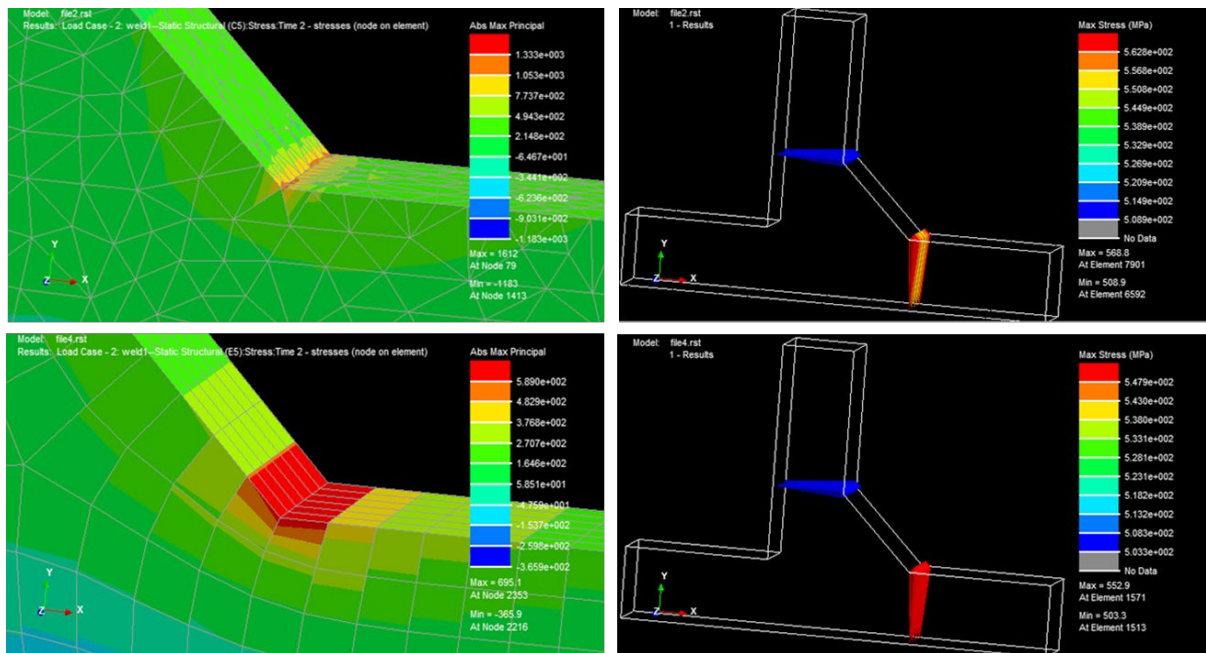


Figure 17: Comparison of local peak FE (left) and linearised (right) stresses.

The through-thickness integration method requires a series of points to be defined along the weld toe, together with a surface normal and another vector to define the orientation of the weld. These points may be defined and imported using an ASCII file. At the time of writing tools for creation of this file are under discussion with different CAE vendors.

As well as predicting cracking from the weld toe, the method can also be used for other crack paths, e.g. out through the weld from the root. This is illustrated in Figure 18.

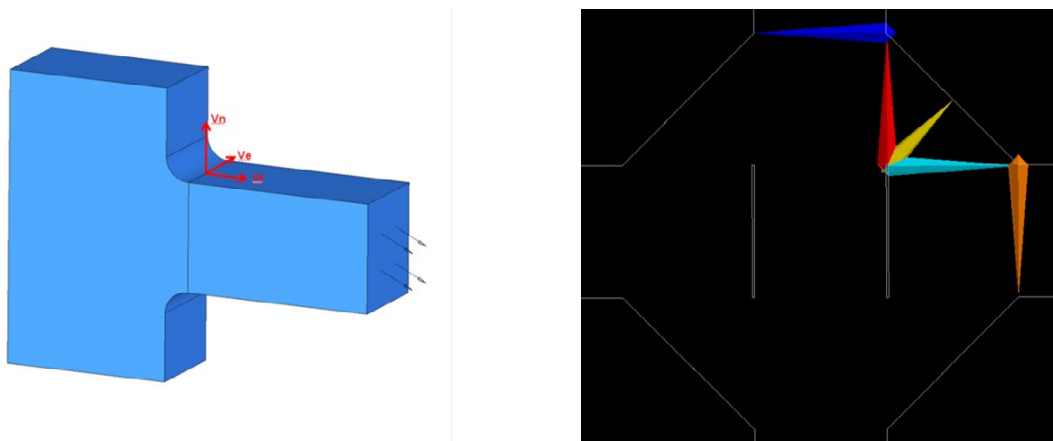


Figure 18: Definition of a calculation point at the weld toe (left) and alternative crack paths (right).

Validation/examples

For an independent assessment of the Volvo method, validated against tests on a variety of welded specimen geometries and sheet thicknesses, see Reference[15].

The following examples are based on test data for a variety of specimen geometries published in Reference[9]. Each geometry was modelled using solid elements, and then analysed in DesignLife using through-thickness integration for stress recovery in conjunction with the Volvo method, using the generic weld S-N curves from the DesignLife database. All the specimens were constructed from 6 mm plate.

The first example is based on the T-tube specimen already illustrated in Figure 15. This was subjected to in-plane and out of plane bending, as illustrated in Figure 19.

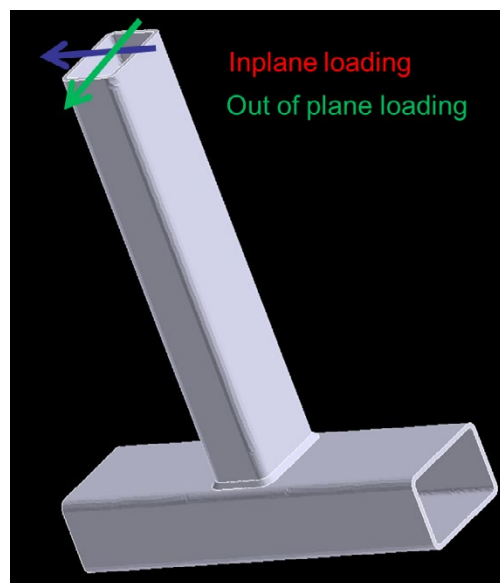


Figure 19: T-tube specimen subjected to in-plane and out-of plane loading.

The specimen was modeled using tetrahedral elements, with calculation points being defined at weld toes in the horizontal and vertical sections of the specimen, as illustrated in Figure 20.

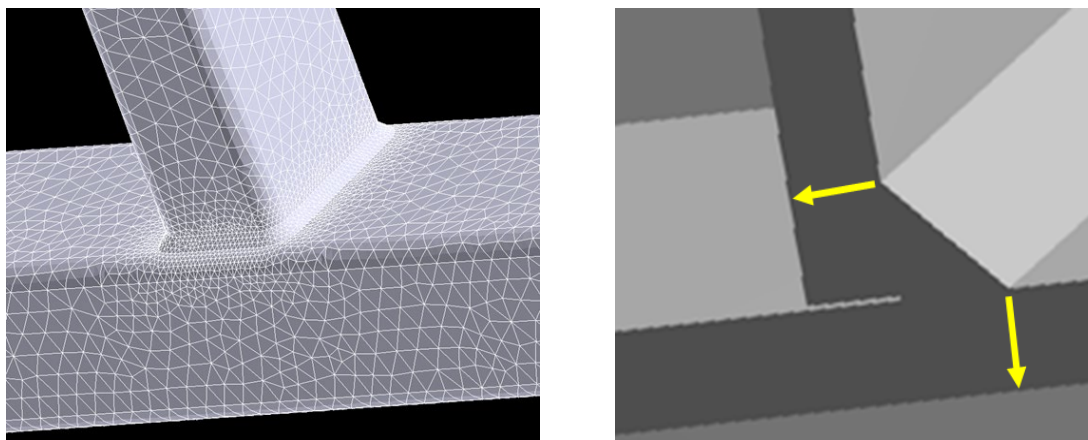


Figure 20: T-tube finite element mesh, and integration paths for weld structural stress.

The correlation between predicted and actual crack locations and lifetimes for a couple of loading cases is illustrated in Figures 21 and 22. Note that the failure criteria for the "Volvo" method is the presence of a visible crack.

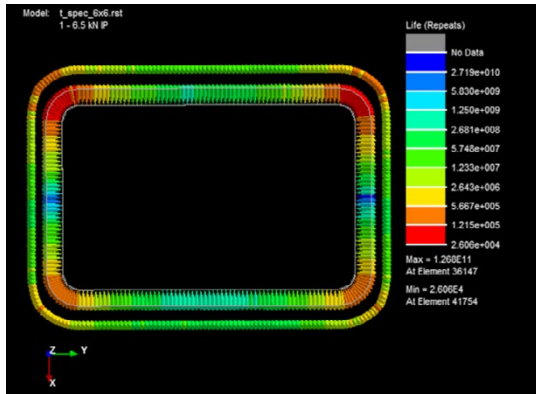


Figure 21: In-plane bending. Predicted life = 26000 cycles. Test = 15 mm crack at 71000 cycles

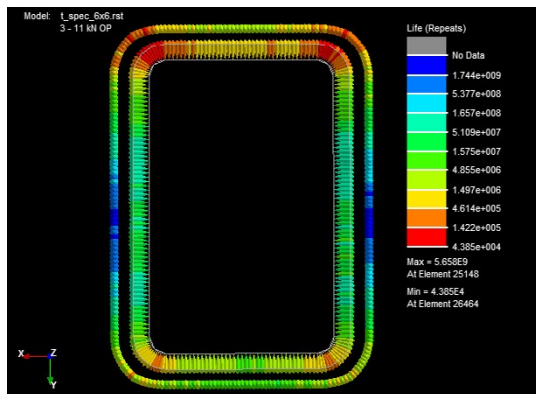


Figure 22: Out-of-plane bending. Predicted life = 44000 cycles. Observed cracks "A" at 47000 cycles. Further validation cases were carried out on specimens with different types of welded attachment. The results are summarised in Figures 23-25.

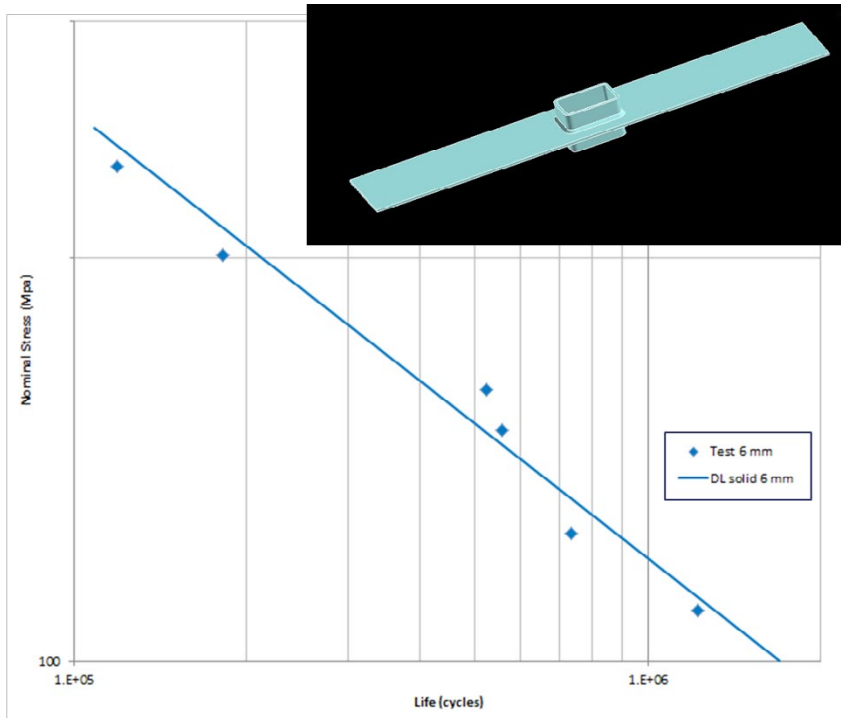


Figure 23: Test-analysis correlation. Welded tubular attachment.

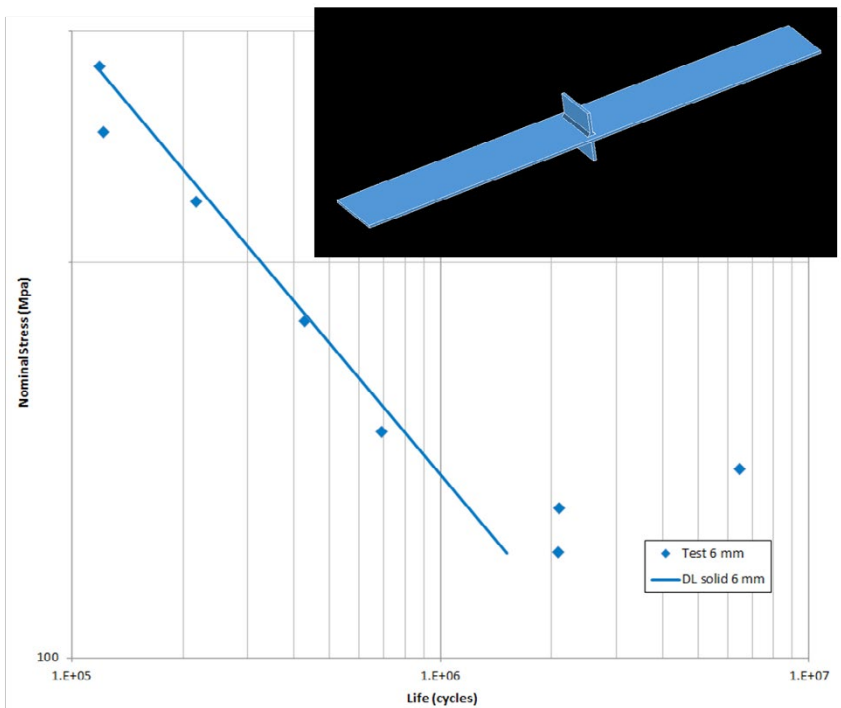


Figure 24: Test-analysis correlation. Short welded attachment.

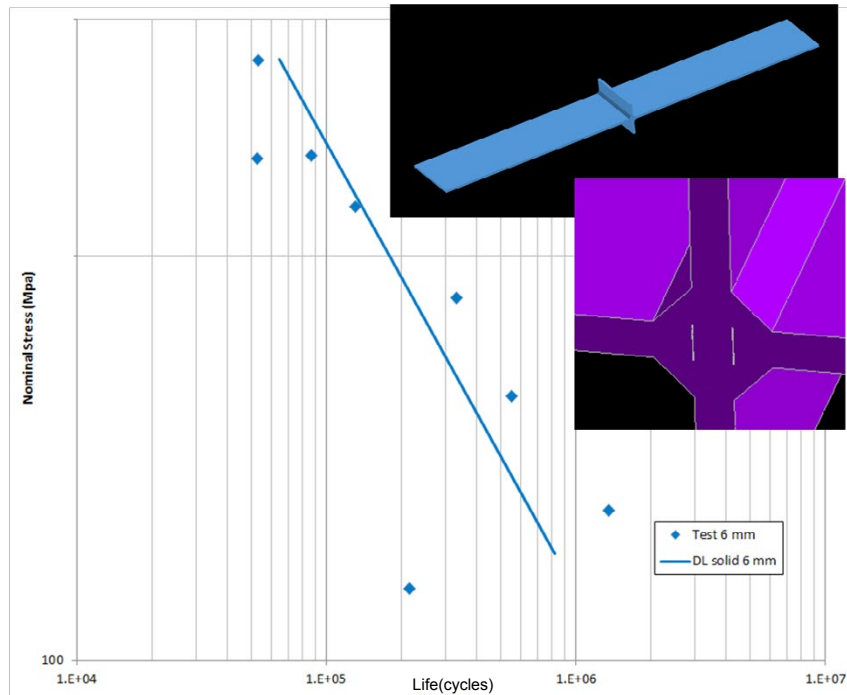


Figure 25: Test-analysis correlation. Welded attachment with partial penetration.

The examples described show good correlation of predicted life and failure location.

Concluding Remarks

nCode DesignLife provides a flexible, powerful and well-validated set of tools for fatigue analysis of seam-welded structures. These tools allow fatigue calculations to be made:

- According to existing standards such as BS7608, ASME and Eurocode 3 or using the “Volvo” approach for which the software is well-known
- Based on shell or solid element models
- Based on normal FE stresses, or structural stresses derived from grid-point-forces, CUBIC stresses or through thickness integration.
- Taking into account the effects of loading type, sheet thickness and where appropriate, mean stresses and small cycles.
- Using standard, generic or user defined S-N curves
- Applicable to a wide range of sheet thicknesses

The methods are all available in nCode DesignLife’s easy-to-use, process oriented user interface.

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Dr. Peter Heyes
Fatigue Simulation Methods

HBM-nCode

Jeff Mentley
Application Engineer

HBM-nCode

HBM nCode at HBM United Kingdom Limited
Advanced Manufacturing Park - Technology Centre
Brunel Way
Catcliffe, Rotherham S60 5WG UK

Tel: +44 (0) 845 620 6060
Web: www.ncode.com
Email: info@hbmncode.com

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