

# FATIGUE DESIGN OF WELDED JOINTS USING THE FINITE ELEMENT METHOD AND THE 2007 ASME DIV. 2 MASTER CURVE

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## **ABSTRACT**

Fatigue design of welded structures is primarily based on a nominal stress, hot spot stress methods or local approaches that have some limitations when coupled with finite element modeling. An alternative recent structural stress definition is discussed and implemented in a post-processor. It provides an effective mean for the direct coupling of finite element results to the fatigue assessment of welded joints. The applications presented confirm the main feature of the method: mesh-insensitivity, accurate crack location and life to failure predictions.

## **1. INTRODUCTION**

Welding is the most widespread joining technique for metallic structures due to its applicability to many geometric configurations. The principal failure mode in welds is fatigue cracking, [1], as a consequence many efforts have been put in the study of the strength of welded joints and in the definition of design guidelines, [2]. As the finite element method has become the favorite tool in structural analysis, there is an unquestionable need for an effective connection of the fatigue assessment approaches to the simulations.

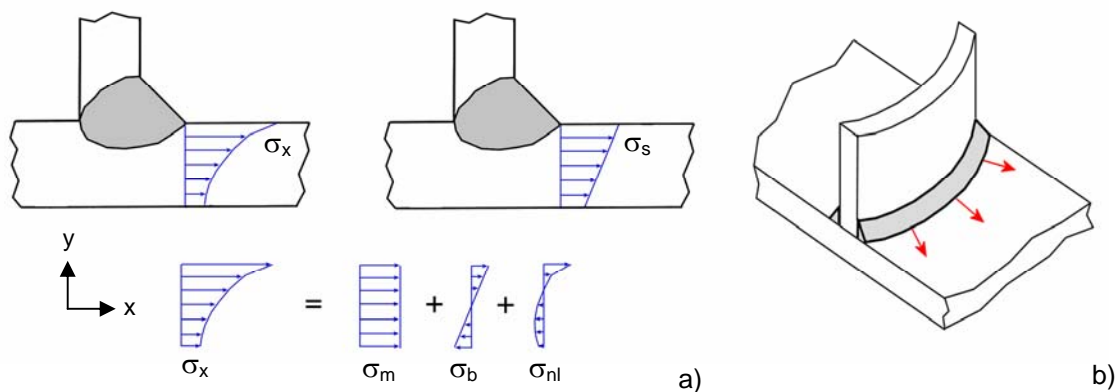
In this work, after a brief review of some well established techniques in Section 2, a recent approach developed at Battelle Institute is described in Sections 3-4. Some applications to experimental tests are then presented in Section 4. This method allows a relaxation of some finite element modeling difficulties, mainly the mesh sensitivity, and grants the possibility of using alternatively solid and shell elements to model welded joints. The results of the FE analysis are then used in combination of a fatigue master curve that consolidates a large number of weld joint configurations.

## **2. APPROACHES TO FATIGUE ASSESSMENT OF WELDED JOINTS**

It is common practice to separate the approaches to fatigue assessment of welded joints into “global” and “local” families, [3-4]. The former consists of the long standing nominal stress method which, in spite of its simplicity, is still the most used and is the basis of all the in force standards and design codes. With nominal stress approach no attempt is made of taking into account the stress concentrations due to macro- and micro-geometric effects of the joints. It makes use of several empirical S-N curves that are associated with detail categories and on corrective factors. The selection of a detail class for a welded joint type and loading mode is often subjective and, in many common situations, even a skilled engineer might have a hard

time in choosing a suitable detail class. This is especially true when the geometry of the structure is complex or when the stress state is not reducible to a simple main component. Moreover it must be added that the real structures can develop cracks in locations different to those indicated in the details present in the standards so this method has severe limitations. The group of “local” methods comprise many different strategies, ranging from the notch stress and notch strain approaches to the fracture mechanics approach. A brief generalization of them is not possible since they differ in the local parameter (being a stress, a strain or a stress intensity factor) and in the phase of the fatigue damage where they can be applied (for example local notch stress is suitable for the crack initiation while fracture mechanics is ideal for crack propagation), [5]. Even if these approaches are sophisticated and have a significant theoretical foundation, the applicability is very often confined to specific cases and therefore they cannot be easily generalized to cover the variety of situations typically found in engineering. This is the main reason why they have not seen a straightforward acceptance in the standard codes, [3].

An intermediate approach between “global” and “local” methods uses a definition of the stress based on a idealized distribution in the thickness of the members in proximity of the welds. Different names have been adopted for this stress depending on the field of application and on the way it was calculated (geometric stress, structural stress, hot-spot stress) but the fundamental idea is to consider the stress component orthogonal to the weld line and reduce it to a linearized distribution, Figure 1a.



**Figure 1:** a) Decomposition of the through thickness stress at the weld toe; b) stress component acting normal to the weld fillet..

These approaches are suited for the assessment of fatigue failures occurring at the weld toes, accordingly it is the stress component normal to the crack plane, i.e. normal to the weld line, that is the driver for crack propagation (Figure 1b). This stress can be inferred by surface measurements and extrapolations, leading to the traditional hot-spot technique. The procedure can be replicated with numerical simulations using finite element models and is present in several standard codes such as Eurocode3. Linearization of the stress over the section thickness can be done only through simulations using nodal stresses, in a way similar to the structural stress calculation typical of pressure vessel standards such as EN 13445. As a result of the linearization, the structural stress  $\sigma_s$  at the weld toe is composed by a membrane part  $\sigma_m$ , constant in the thickness, and a bending part  $\sigma_b$ , as depicted in Figure 1a. The remaining self-equilibrated non-linear  $\sigma_{nl}$  is not considered, therefore the structural stress includes only the effects of gross structural discontinuities but disregards the local notch effect due to the weld geometry. The notch-induced complex stress state at the weld toe can then be simplified

and only the two components  $\sigma_m$  and  $\sigma_b$  are taken into account. The finite element simulations required for this kind of study are linear elastic and the fatigue assessment is performed using structural stress S-N curves that are in limited number with respect to the nominal stress curves.

### 3. STRUCTURAL STRESS APPROACH BASED ON NODAL FORCES

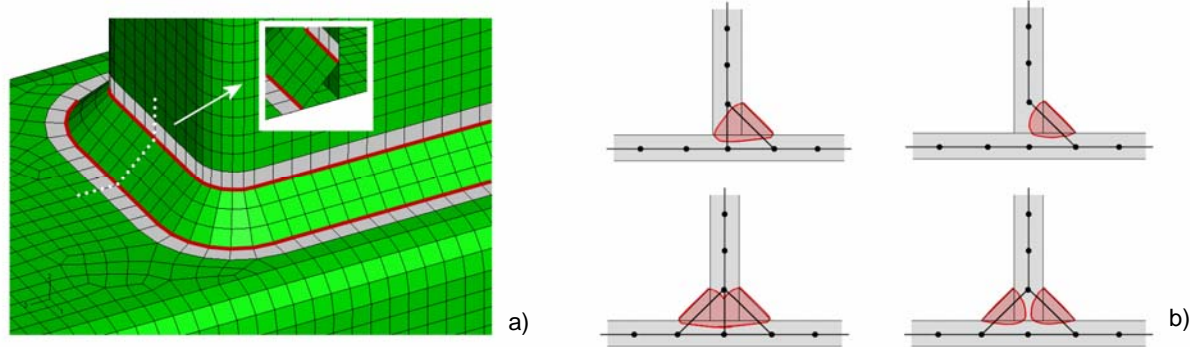
The finite element framework allows the calculation of a structural stress based on forces and moments at the nodes of the mesh. This method has the distinctive advantage of providing a structural stress fairly insensitive to the mesh features (element size and element type) in the areas corresponding to the weld toes. The typical mesh-dependence that is found in the traditional surface extrapolation method and, partly, in the through thickness linearization, is therefore overcome. Elemental stresses or stresses extrapolated to the nodes are in fact influenced by the element formulation and by the geometric characteristics of the finite elements, whereas nodal forces directly come from the equilibrium of the structure. Nodal forces (and moments) for each element are calculated from the stiffness matrices and the nodal displacements (and rotations). The displacements are the primary output of displacement-based FE codes and the equilibrium at each node in the mesh is satisfied regardless of the element size and element formulation. A few different variants of this approach has been proposed and they mainly come from the automotive field for either spot or seam welds, see for example Ferner and co-workers [6]. The literature reports also a recent implementation of a similar approach in the commercial code Femfat, [7]. Researchers headed by Dong at Battelle Institute, have formulated an effective procedure for the calculation of the structural stress from forces and moments at the nodes of a finite element mesh, based on work-equivalence considerations, [8]. In Dong's method, first distributed line forces (and moments) are determined along the edges of the weld toe lines starting from balanced nodal forces (and moments), then at each node the structural stress is calculated as:

$$\sigma_s = \sigma_m + \sigma_b = \frac{f_y}{t} + \frac{6 \cdot m_x}{t^2} \quad (1)$$

where  $t$  is the section thickness of the plate,  $f_y$  is the line force in the local  $y$  direction orthogonal to the weld line and in the plane of the shell,  $m_x$  is the line moment in the local  $x$  direction tangent to the weld line. Forces and moments have to be preliminarily rotated into local coordinate systems defined at the nodes of the weld line. The line forces (and moments) that result are continuous along the weld toe lines and so do the structural stress. The detailed procedure is described in several publications, for example [8-9].

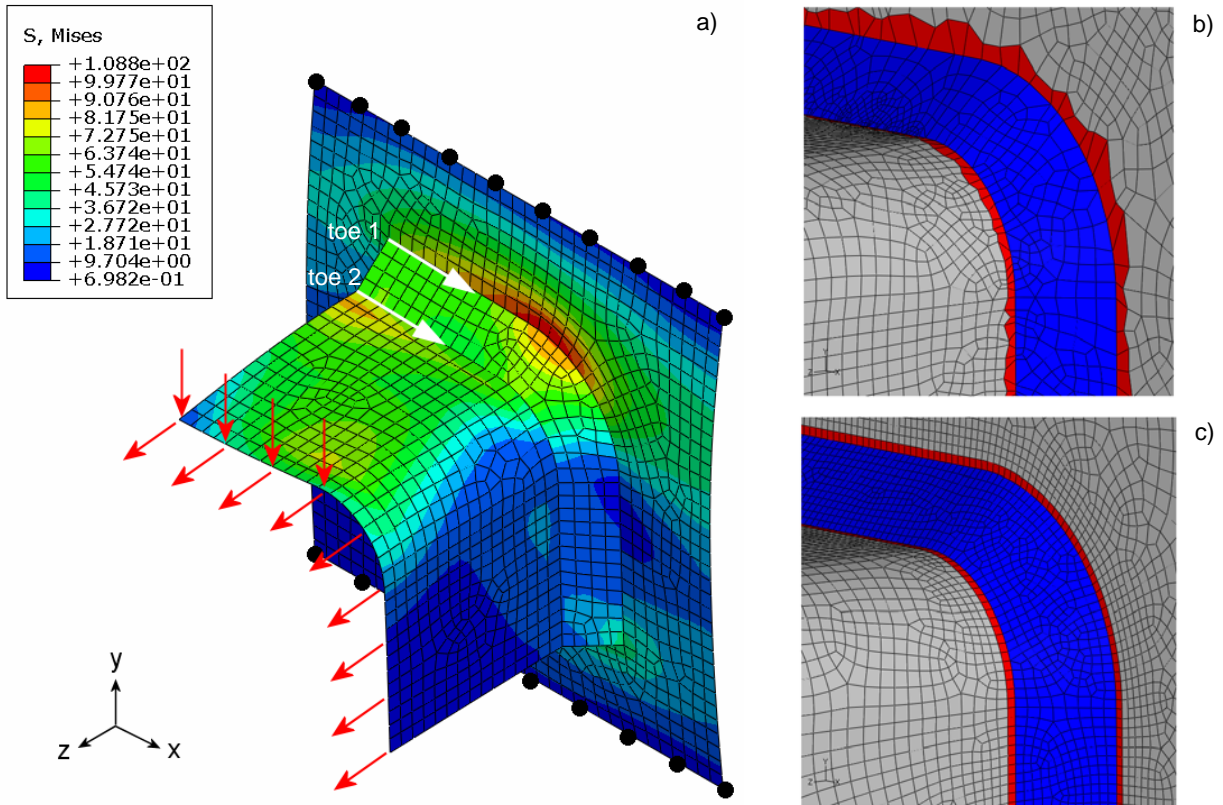
Even if these concepts can be extended to solid elements (2D and 3D), the approach is particularly suited for shell elements. Nevertheless, since shell and plates are often the preferred choice for many engineering structures that make large use of welded joints (for example trucks, ships, cranes, bridges, ...), the potential applications of the method are many. It must be stressed that the finite element simulations has to be linear elastic therefore a fatigue assessment of the welded joints in the components can be a precious additional outcome of a standard stress analysis. The only specific requirement concerns the modeling of the welds because the fillets have to be explicitly included to correctly represent the stiffness of the joints. This is usually done using inclined elements, as shown in Figure 2a that reports an example of a T-joint connection between two tubular parts. In Figure 2b some strategies for a realistic modeling of partial and complete penetration seam welds are collected. The

manual creation of the elements in the welds is a tedious and time consuming part of the procedure and the automation of the weld definition is to be pursued. Note how for a given fillet, in general both the two toe lines have to be analyzed since a priori it is not known which is the most prone to fatigue propagation and in what location.

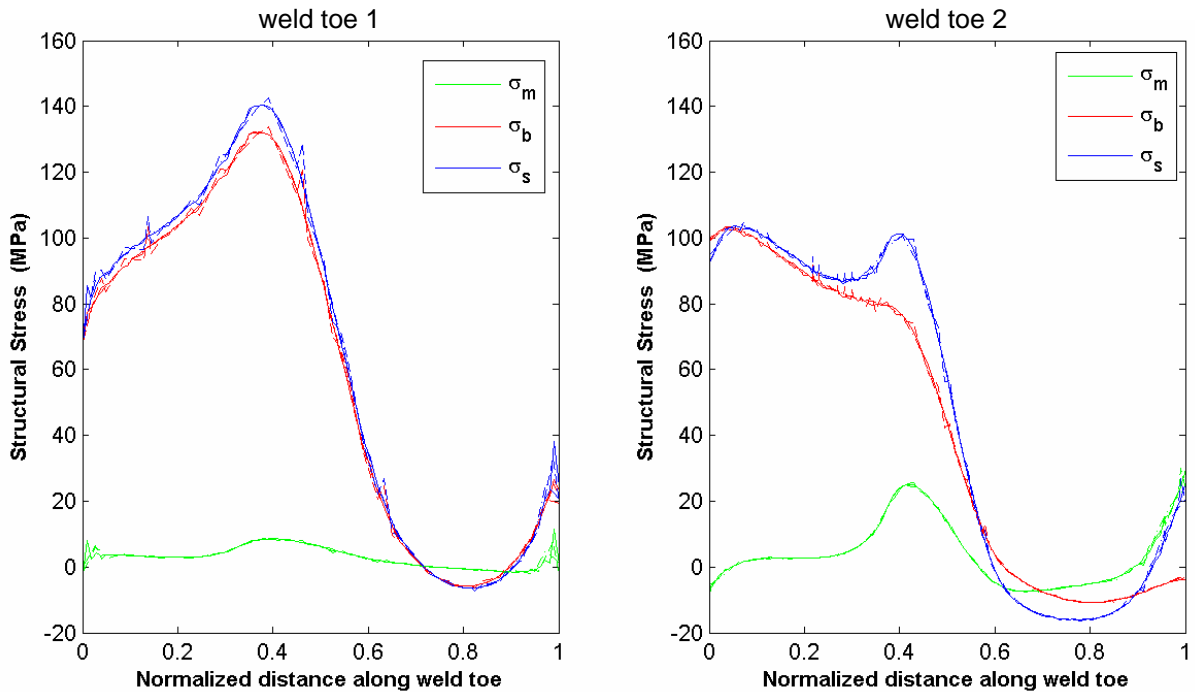


**Figure 2:** a) Tubular connection (T-joint) modeled with shell elements; b) fillet welds with partial and complete penetration.

The mesh-insensitivity of the structural stress implied above, is demonstrated with the example of Figure 3. Here a curved plate is joined on the external side to a flat plate by a full penetration fillet weld. The main dimensions of the flat panel are 100x100 mm, the curved plate fits in a cube of 50 mm edge and the thicknesses are 5 mm. A uniform traction is applied longitudinally to the curved plate and a transverse loading acts on the top edge as shown in Figure 3a; top and bottom nodes of the flat plate are pinned. Three different meshes are studied, the first one is shown in Figure 3a together with the resulting Mises stress on the visible surfaces of the shells. This is a rather coarse mesh but regularly spaced and with a low aspect ratio in the rows of elements corresponding to the two weld toes. The second case considered, Figure 3b, has weld toes characterized by distorted elements having a wide spectrum of shape metrics. Figure 3c shows the third mesh with a regular and fine discretization. Any experienced finite element analyst would prefer the last one for a local stress investigation but such level of refinement may not be necessary for a fatigue analysis based on the present procedure. The proof comes from Figure 4 where the structural stress ( $\sigma_s$ ), the membrane ( $\sigma_m$ ) and bending ( $\sigma_b$ ) components are plotted along the weld toe lines 1 and 2 indicated in Figure 3a. The results from the three meshes are drawn with different line styles (refined mesh: continuous lines, distorted mesh: dashed lines, coarse mesh: dash-dot lines) but they are barely distinguishable since they are closely overlapped. The distorted mesh plots have some little jerks in correspondence of some elements but the trends are completely captured and so are the peak values and positions. Even the coarse mesh does not fail in revealing the maximum values and the locations where the fatigue cracks would most probably propagate. If the combined loading was proportional, in-phase and constant amplitude, then these plots suggest that a fatigue crack propagation in the through thickness of the plate would take place in correspondence of the maximum structural stress in the two toes, that is at about  $0.4 \cdot L$  of the toe 1 ( $L$ : length of the toe line). These graphs give also useful information about the relative magnitude of the structural stress components, membrane and bending, so the analyst has a clear picture of the loading mode in each point along the weld toes. In the example discussed here there is a strong predominance of bending stress in most parts of the toe lines.



**Figure 3:** Curved profile welded to a flat plate: a) shell model using a coarse regular mesh at the toes; b)-c) close views of the fillet for the irregular mesh and the refined regular mesh.



**Figure 4:** a) Comparison of the structural stress and its membrane and bending components along the weld toes of Figure 3.

One of the major drawbacks of the structural stress approaches in their basic forms is that they usually take into account only the stress component normal to the weld line. In analogy with Equation (1), a structural shear stress along the fillet could be calculated as:

$$\tau_s = \tau_m + \tau_b = \frac{f_x}{t} + \frac{6 \cdot m_y}{t^2}. \quad (2)$$

As of now only few studies on multiaxial fatigue are available, [10], so a complete understanding of the correct combination of  $\sigma_s$  and  $\tau_s$  is yet to be available.

The whole procedure has been implemented by the authors in Matlab code that acts as a post-processor to the FE code Abaqus. Together with the calculations described above, the code detects the toes for all the fillets in the shell mesh, thus providing an automatic, quick and complete analysis of the seam welded joints.

#### 4. THE ASME MASTER S-N CURVE

According to this approach, the structural stress defined in Equation 1 is consistent with the far-field stress typically used in fracture mechanics to compute the stress intensity factors  $K$  for a given crack shape and size. Since the life of welded joints is dominated by crack propagation, the structural stress and its components correlate the actual geometry and loading of any joint to simplified fracture mechanics configurations where crack growth models can be applied. As a result of the analytical procedure developed in [9] for a two-stage growth model, an equivalent structural stress parameter can be defined as:

$$\Delta S_s = \frac{\Delta \sigma_s}{t^{(2-m)/2m} \cdot I(r)^{1/m}} \quad (2)$$

where the structural stress range  $\Delta \sigma_s$  is modified by a loading mode function  $I(r)$  and by a thickness correction factor. The polynomial  $I(r)$  is a function of the ratio  $r$ :

$$r = \frac{\Delta \sigma_b}{\Delta \sigma_m + \Delta \sigma_b} \quad (3)$$

that represents the content of bending stress over the total structural stress,  $t$  is the section thickness of the plate and  $m=3.6$  represents the slope of a Paris-like crack propagation curve. It has been shown in several publications by Dong and co-workers that, using this procedure, it is possible to define a single S-N curve for many different weld geometries and loading configurations, therefore proving its robustness. This master S-N curve, which has also been incorporated in the ASME Boiler and Pressure Vessel Codes (2007) Division 2 as an alternative prediction method, has the following form:

$$\Delta S_s = C \cdot N^h \quad (4)$$

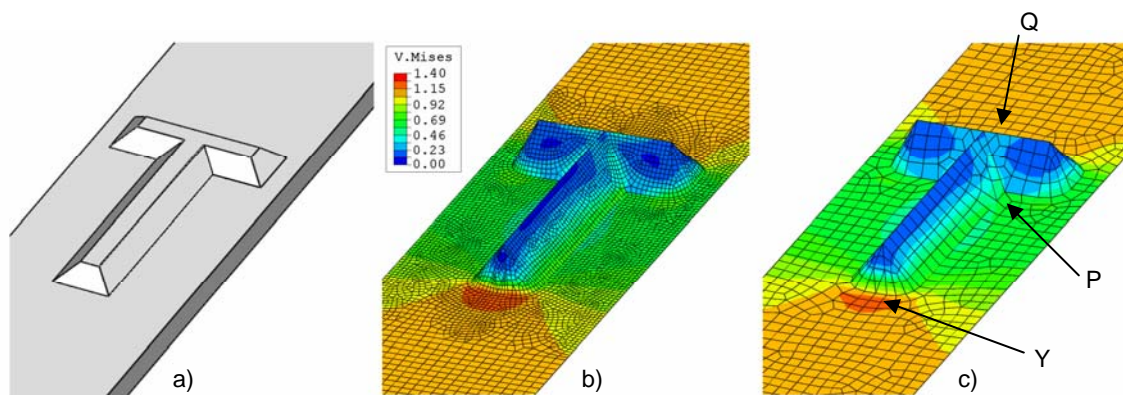
$C$  and  $h$  are parameters of the material and are tabulated for different prediction intervals. Contrary to most of the standards, this norm does not provide neither a cut-off limit (fatigue strength for infinite life) nor any knees in the curve: all the cycles (after a rainflow filtering) are considered damaging. This is consistent with the recommendations given in [11].

## 5. APPLICATIONS

Some experimental tests were performed to further validate the structural stress approach and ASME master S-N curve. Several specimens of three different geometries were subjected to pulsating tensile constant amplitude loading (stress ratio  $R=0$ ). To assess the predictive capabilities of the method, the maximum load was set to values corresponding to certain number of cycles. Given the target life, the structural stress was deduced from Equations (2-4) and compared to the maximum structural stress found in a finite element simulation of the specimen subjected to a known load. The linearity of the solution allows then an easy scaling of the applied load to determine the force required in the test. The target life of this experimental campaign ranged from  $N=1.0E+5$  to  $N=5.0E+5$ .

The specimens consisted of a plate (thickness  $t=8$  and  $10$  mm, width  $w=50$  mm, length  $L$  about  $250-300$  mm) with attachments on the top surface. The attachments were H-shaped, T-shaped and a solid cubic block (H, T, B in the following). The H and T specimens had  $5$  mm thick attachments that were centered and aligned with the longitudinal direction, while the solid block ( $37 \times 37 \times 30$  mm) was not symmetrically positioned on the top of the plate. The material was a typical structural steel Fe510. The failure criterion adopted was the complete propagation of the fatigue cracks through the plate thickness, so the tests were continued until final fracture had occurred and the total number of cycles could be determined.

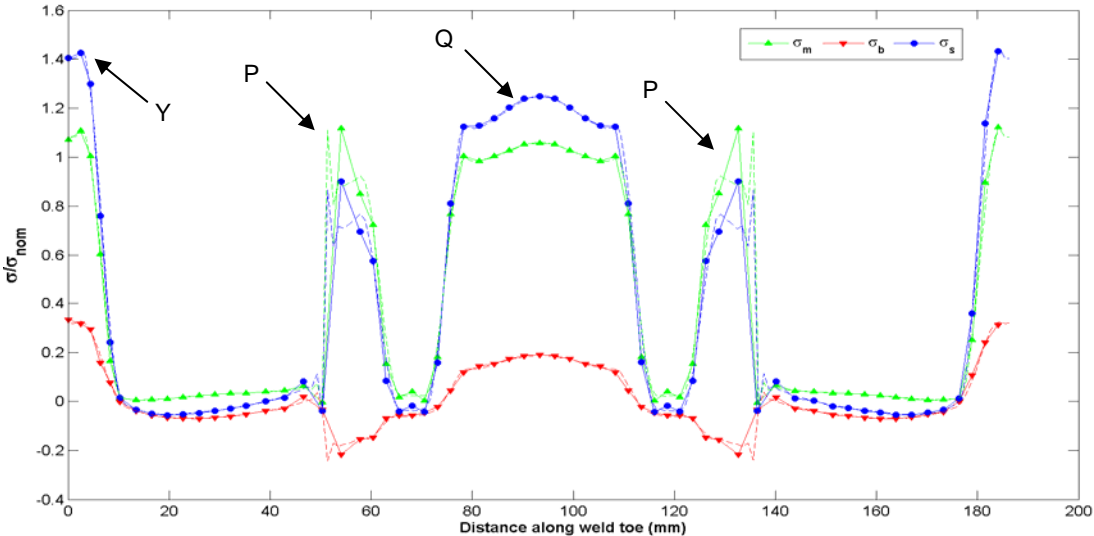
A detailed presentation of the results is provided only for specimen T. Figure 5 shows two finite element models constructed using shell elements. Because of the particular geometry, the T-shaped attachment is completely wrapped by the inclined elements forming the fillet. The simplified toe line in Figure 5b is continuous and smooth except in two points where there is an abrupt change in direction and a  $90^\circ$  angle. In these points (marked with a P in the figure) it has to be expected a disturb in the structural stress course.



**Figure 5:** a) Specimen with the T-shaped attachment; b)-c) shell models with different mesh size. The V.Mises stress is normalized by the nominal tensile stress.

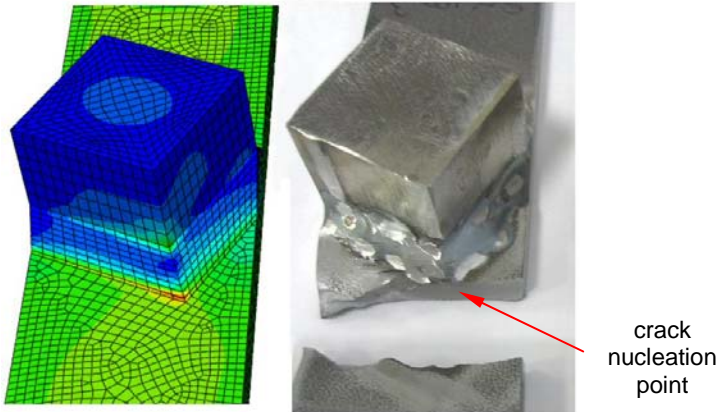
The results of the post-processed simulations are provided in Figure 6 for two mesh sizes. The graph has in the x-axis the position of the nodes on the toe line with respect to a curvilinear abscissa with origin on the tip of the T and running along the whole fillet. The continuous lines refer to the coarse mesh while the dashed lines correspond to the fine mesh. The structural stress values are normalized by the nominal stress in the section ( $\sigma_{nom} = F/A$ ,  $F$ : applied force,  $A$ : area of the transverse section). These stresses are to be intended as located on the upper surface of the plate and at the intersection of the weld elements and plate elements. The graph show that there is an obvious symmetry due to the geometry, and, above all, that the fine mesh and coarse mesh plots are consistently overlapped and the differences

are substantially negligible almost everywhere. There are some sensible discrepancies only close to the points P. It is evident from the graph that the membrane content is dominant as could have been expected since the specimen is loaded in tension. The macro-geometric notch effect caused by the attachment is responsible for the bending component. Peak values are reached at the tip of the T (point Y in Figure 5) and in the middle point of the top leg of the T (point Q in Figure 5). These are actually the locations of the fatigue failures, the choice between Y and Q being dictated by the level of grinding that influences the toe status. Along the sides of the T that are parallel to the load direction the stress is close to zero because only the stress component normal to the weld fillet is considered.



**Figure 6:** Structural stress and its components along the fillet weld of specimen T.

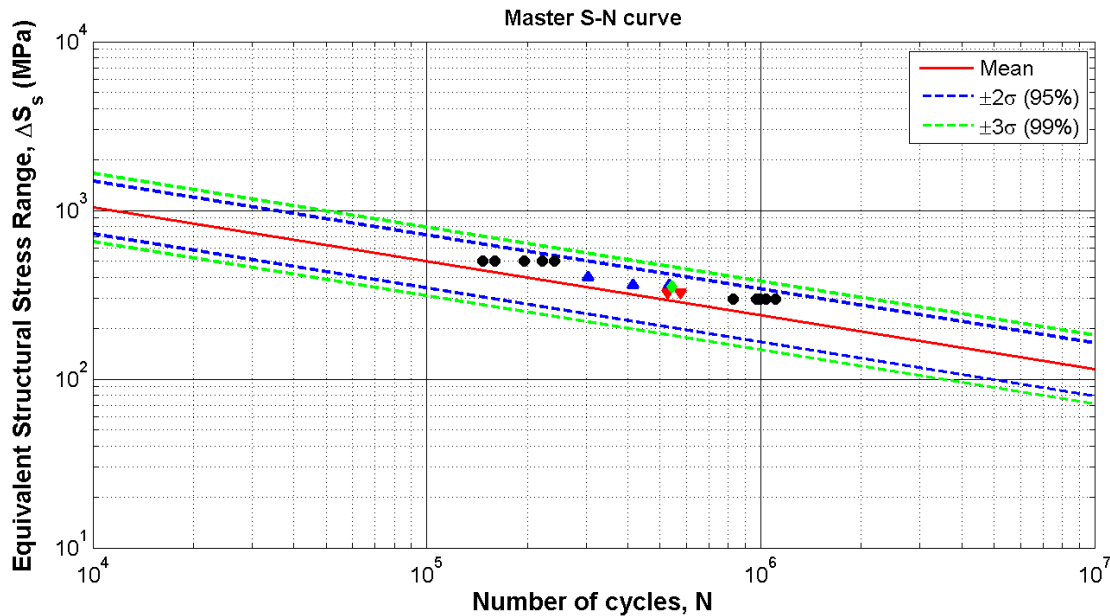
Analogous plots are obtained for specimens H and for sake of brevity are not reported. The presence of the stocky block in specimens B prevents the use of shells so for that case solid 3D elements are employed, Figure 7. It is important to note that the failure location is always in correspondence with the highest value of the structural stress so these tests confirm the ability of the method to take into account the main features of fatigue failure in a simplified but correct way.



**Figure 7:** Finite element model using 3D elements and real specimen with the solid block.



The full set of data is reported in Figure 8, it can be seen that all the tested specimens, with their different geometries, fall between the mean master S-N curve and the  $+2\sigma$  curve. The prediction made using the presented calculation approach is therefore fairly accurate and conservative.



**Figure 8:** Master S-N curve and experimental results (●: specimens H, ▲: specimens T, ▼: specimens B).

## 6. CONCLUSIONS

The work presented a structural stress approach to fatigue assessment of welded joints that integrates well with finite element modelling. The implementation in a post-processor tool was successful and showed the potentiality for becoming a useful tool for the design of welded structures subjected to fatigue.

The mesh-insensitivity was assessed: even coarse meshes provide adequate structural stress estimates so the method can be used for the modeling of complex structures.

The procedure was applied to three different specimens subjected to constant amplitude loading and gave correct location of the fatigue cracks. Finally, the use of the ASME master S-N curve proved to give satisfactory results in terms of accuracy of cycles to failure.

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