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Fatigue Design of Complex Welded Structures Using Finite Element Analysis in Hot Spot Approach and Notch-Stress Intensity Factor Approach

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Abstract: **In fatigue design of complex welded structures the consolidated approach of nominal stresses, as given by Design Codes, is seldom applicable. This is due to two main reasons, one connected with the complexity of the structure, causing a difficult definition and evaluation of a nominal stress, the second connected with the complexity of the welded joint itself, causing some difficulties to single out a similar joint among the detail categories included in the design classes of the Codes. To overcome this problem, the Eurocodes offer the possibility of applying the Hot Spot approach and Notch-Stress Intensity Factor (NSIF), which allows a better definition of a nominal stress (often called structural or geometrical stress in the literature), especially when FE analyses of the structure are employed. And authors found that the NSIF approach is much more reliable than the Hot Spot one.**

1. INTRODUCTION

Under fatigue loading the weld beads represent Achille's heel of welded structures. The fatigue crack will develop, usually, at the toe of the weld, in the base material, due to stress concentration effects, residual stresses and potential defects. Cracks starting at the root of the weld are also possible, for the same causes, but these are more dangerous in real structures for safety reasons, since the crack would become detectable only after emerging on the surface of the weld bead.

The design of real engineering welded structures is a difficult subject for several reasons. In our opinion the most important ones are the following:

- The choice of a representative time stress history;
- The high number of welds and the difficulty of selecting the weakest links on the structure;
- The difficulty of defining material properties, which vary throughout the weld and the heat affected zone (HAZ);
- The presence of high residual stresses, both local (due to the weld itself) and structural (due to the assembly process of the structure) which vary throughout the weld and the HAZ;
- The difficulty of defining precisely the weld bead geometry: bead size and shape and the radius at the toe of the weld will vary even in well-controlled manufacturing operations;
- The difficulty of taking into account secondary bending effects due to manufacturing imperfections;
- The difficulty of defining a model of the idealized weld geometry in a manner which is sufficiently precise for analysis purposes but sufficiently simple for industrial use;
- The difficulty to propose one single fatigue design criterion able to predict the whole fatigue life (i.e., from the first applied fatigue load cycle up to final failure) of a real structure. In fact in real structures crack propagation paths can be relatively long and even multiple, so that a distinction between initiation and propagation phases, even though qualitative, should be made. Accordingly, different criteria have to be used: concerning crack propagation phase the Linear Elastic Fracture Mechanics is commonly adopted, while concerning crack initiation phase (i.e. true initiation and early propagation of short cracks) we have put forward the Notch-Stress Intensity Factor approach. Anyway, as far as fatigue life of welded specimens and not of real structures is concerned, most of the fatigue life is spent in true initiation and short crack propagation, the NSIF approach has proven to be able to estimate the whole fatigue life, as it will be seen later.

Current methods for fatigue assessment of weldments are based on the nominal stress applied to the joint, which should be compared with the fatigue strength of a similar joint to be chosen between a numbers of classes given by the design Standards. To overcome the difficulties deriving from the complexity of the geometry of the structures and of the applied loads, a ''structural stress" or ''hot spot stress" has been introduced for the design of offshore structures, and then extended to general structures, which is simply a more careful definition of nominal stress.

From a theoretical point of view it has been shown that the fatigue strength of welded joints depends on the state of stress and strain reached in a small control volume around the crack starting zone and that the limit values of these parameters for a given number of cycles are not linked to the geometry of the joint under tensile or bending loading.

The paper will focalize on some aspects where the Standards need to be integrated by theoretical considerations, if a sound design has to be performed, as in the case of complex structures under heavy fatigue loadings.

2. USE OF FINITE ELEMENTS FOR FATIGUE ANALYSIS

In Finite Element analyses of welded joints the usual assumptions are

1. Homogeneous material throughout the weld and the base material;
2. Weld toe radius equal to zero; alternatively a fictitious notch radius ρf equal to 1 mm is suggested, after Radaj [37], in IIW Recommendations [38] to determine the 'effective notchstress', the value of qf being consistent with the assumption of a real weld toe radius equal to zero;
3. Constant geometry of the weld bead (using the mean values of parameters expected in production);
4. Absence of manufacturing imperfections.

Up to now these assumptions have demonstrated to be consistent with the experimental fatigue test results. In practical applications the quality control should evaluate the entity of manufacturing imperfections. Some limits for allowable secondary bending due to misalignments and for welding defects are given by design Standards and by IIW Recommendations [38]. When particular welding techniques or post-welding treatments smooth the toe radius, the fatigue life could be improved and the minimum value of the expected radius should be assumed for the numerical analyses, as will be clarified later on dealing with the size effect.

If these assumptions are accepted, the advantages and the drawbacks of a FE analysis for the fatigue design of a welded structure are in general the same that the ones for any other structure, but care should be taken for some peculiar aspects. They are simply a consequence of the fact that fatigue is a local phenomenon, and then each stress raising detail could be a weakest link for the structure and should be considered in the analysis. This can be accomplished either considering the detail as a black box (if data on its effect on fatigue life are available or could be estimated) or zooming the FE analysis on the detail. For many mechanical components this can be achieved directly by choosing 2-D (plane stress or plane strain) or 3-D (brick) elements for the analysis, with smaller elements around the critical detail. This is not possible in general for complex welded structures, since such a model would be not practical for industrial applications. In this case, the analysis of the structure is usually performed using beam or shell elements, depending on the overall geometry of the joint, which are suitable for modelling mid axis or mid surfaces of the structural component, respectively. In relation to the structural analysis, it has been evidenced that the stress field evaluated around the weld toe can be strongly underestimated if the bead stiffening effect is not accounted for. Several ways to do it have been proposed [39] and a very simple but efficient one is shown in Fig. 1, while the effects that an incorrect modelling technique can have are shown in Fig. 2 [40].

The results obtained from a main model of the structure with beam or shell elements can then be used to apply the hot spot approach. Some recent papers by Doerk et al. [41], Poutiainen et al. [42] and Hobbacher [38] are surely very useful for this purpose. Otherwise those results can be used as boundary conditions for 2-D or 3-D submodels of the critical details, as shown in Fig. 3 [40], if the local stress field caused by the weld itself has to be evaluated for application of NSIF approaches. Since the refinement of the mesh necessary to correctly evaluate the local stresses is difficult to obtain in 3D welded details, several procedures have been developed to obtain reliable results using relatively coarse meshes (on the order of 1 mm), e.g. by Taylor et al. [18], Meneghetti [40, 43], Tovo and Livieri [44], Lazzarin et al. [25, 26]. A simplified model is also available to estimate NSIF values directly from structural stresses evaluated by means of beams or thin shells at a distance from the weld toe equal to the main plate thickness [45]. The extension of the SED approach to multi-axial fatigue is presented in [46, 47], whereas the different slopes for tension and shear loading are justified in [25] on the basis of the different role played by local yielding under tension or torsion loadings.

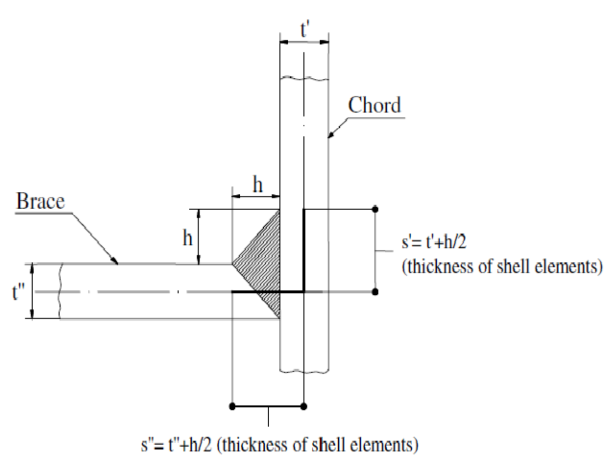

Fig 1. Adopted technique in order to account for the weld bead stiffness of tubular joints [39].

It should be pointed out and clarified that the hot spot approach and the NSIF approach are totally different each other, that they should be applied in different ways and the values of the stress ranges evaluated by the two methods should be compared with different values of fatigue strength. As far as stress analysis is concerned, the hot spot approach requires the evaluation of the stresses at a distance from the weld toe on the order of the main plate thickness, while for the NSIF approach the distance is on the order of 1/10 of the thickness or less. As far as the fatigue strength is concerned, only the NSIF approach is able to take into account different joint geometries and sizes and refers to a unique fatigue strength curve for fatigue failures starting at the weld toe for a given opening angle (e.g. 135°). Conversely, the hot spot approach takes into account only the structure complexity, and cannot take account of different joint geometries (requiring the appropriate fatigue strength curve for each geometry) nor of different sizes (requiring explicit rules on how to adapt the fatigue strength classes to joints of different sizes). Finally, only the approach based on strain energy density over a control volume is able to unify the fatigue strength data for various slopes of the weld bead or fillet and for cracks starting from the weld root, taking into account also the size effect.

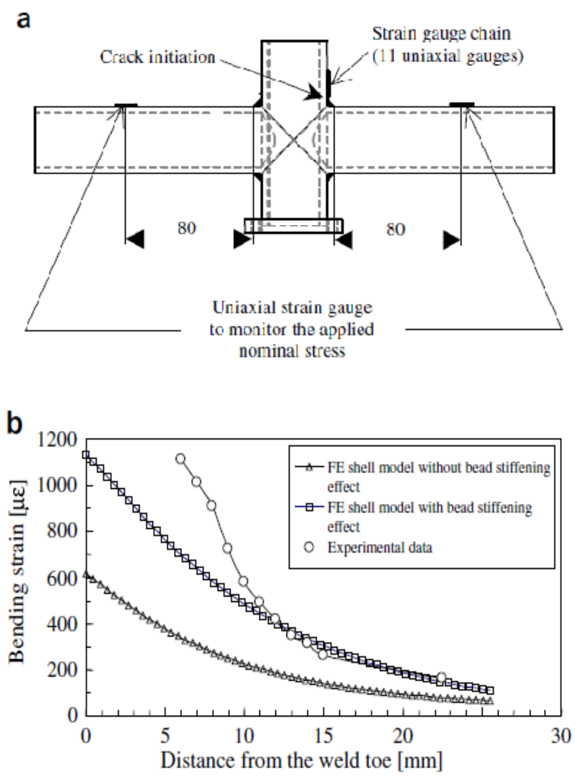

Fig.2. Position of the strain gauge chain for measuring the structural stress field close to the crack initiation point. The first gauge of the chain is located at 6 mm from the weld toe (a); comparison between experimental strains as measured by the strain gauge chain reported in Fig. 11(a) and numerical FE results obtained from the shell 'main model' (b) [40].

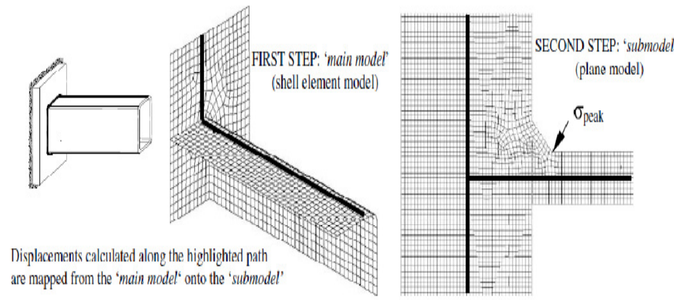

Fig 3. Adopted analysis technique for the tube-to-flange type geometries [40].

Table 1. NSIF range ΔK1, A evaluated at NA = 2 x10^6 cycles and PS = 50%, inverse slope k and scatter index TK of fatigue strength curve for steel and aluminium welded joints

Material	ΔK_{1A} (MPa mm$^{0.326}$)	k	T_K
Steel [19]	286	3	1.80
Aluminium [19]	124	4	1.85

Table 2: Range of the averaged SED Δw^{k}A evaluated at NSIF range ΔK1, A evaluated at NA = 2 x10^6 cycles and PS = 50%, critical radius Rc, inverse slope k and scatter index TW_ of fatigue strength curve for steel and aluminium welded joints

Material	$\Delta\bar{W}_A$ (N mm/mm^3)	R_c (mm)	k	T_W
Steel [23]	0.105	0.28	1.5	3.3
Aluminium [23]	0.103	0.12	2	3.3

On the basis of the data published up to now [19], the NSIF fatigue strength curve for cracks starting at the toe of fillet welds ribed by the following equation for both materials:

$$\Delta K^k_{1,N} \cdot N = \Delta K^k_{1,A} \cdot 2\times10^6$$

Where ΔK1, N and ΔK1, A are the NSIF ranges evaluated at N and NA = 2 x10^6 cycles, respectively, with PS = 50%. The values of ΔK1, A, of the inverse slope k and of the scatter index Tk (ratio of the fatigue strength at PS = 2.3% and PS = 97.7%) are listed in Table 1.

The available data for strength of steel joints failing from the toe or from the root of the weld, in terms of local strain energy density [22, 23] are described by the following equation:

$$\Delta\bar{W}^k_N \cdot N = \Delta\bar{W}^k_A \cdot 2\times10^6$$

where ΔW^{k}N and Δw^{k}A are the ranges of the averaged SED evaluated at N and NA = 2 x10^6 cycles, respectively, with PS 50% and critical radius Rc. The values of Δw^{k}A, of the critical radius, of the inverse slope k and of the scatter index Tk related to PS = 2.3% and PS = 97.7% are listed in Table 2.

3. USE OF STRAIN GAUGES FOR FATIGUE ANALYSIS

It is obvious, but not always clear, that strain gauge measurements are not an alternative but a complementary method for fatigue design of complex welded structures.

There are three different ways to locate strain gauges on the structures:

1. Far away from weldments or, in general, from stress raisers, in zones with no- or very low-stress gradient. This location is useful especially to evaluate in-service load histories and to calibrate finite element analyses;
2. At a distance from the weld toe on the order of main plate thickness, where low-stress gradients exist. This location is needed to apply the hot spot approach (by extrapolation to the weld toe of the measurements of two strain gauges) and to evaluate the time-histories of the analyzed detail;
3. At a distance from the weld toe of two-three millimeters, in zones where high stress gradients exist. This is the location suggested by Haibach to unify the fatigue strength of welded joints of different geometries on the basis of the measured strain.

While for the first two applications the exact location and the length of the strain gauge grid is not important, for the third application the measured strain is strongly influenced by both the distance from the weld toe and the grid length [12].

For fatigue strength assessments of welded joints with the hot spot approach, a reference fatigue curve is needed, although expressed in different ways for the two Standards. When a small strain gauge is located near to the weld toe, at a fixed distance from the weld toe and for a given grid length, the strain range for a given fatigue life is in general assumed to be independent on the joint geometry [1].

Published data [1, 48–51] for a life of 2 x10^6 cycles and for a 3 mm grid length located with its mid axis at 2.5 mm from the weld toe are summarized in Table 3.

Table 3: Published data for a life of NA = 2 x10^6 cycles, for 3 mm grid length located with its mid axis at 2.5 mm from the weld toe

Material	Thickness of the main plate (mm)	R	$\Delta\varepsilon_A$ (με)
Steel stress relieved [1]	10–50	−1	1050
Steel as welded [48]	10–17	0.1	752
Steel as welded [49]	10–25	−1	1140
		−0.25	830
		0.5	652
Steel as welded [49]	3–5	−1	1394
		0.1	898
		0.5	782
Aluminium as welded	10–20 [50]	−1	1388
		0.5	880
	5 [51]	−1	1832
		0.5	1260

4. SIZE EFFECT AND IMPROVEMENT OF FATIGUE STRENGTH

It has been shown by several authors that a size effect is present in fatigue strength of welded joints, due to the sharp V-notch at the toe of the weld. For fillet welds with a 135° open V-notch describing the weld toe profile, the stress field very close to the V-notch tip will be of the type

$\sigma_{xx} r^{0.326}$ = constant and the size effect, for similar welds of different size, could be expressed by the ratio of the values of whichever geometric parameter, to the power of the same exponent 0.326 describing the mode I stress field intensity. Since welded joints of the same geometry but of different sizes usually are not similar, with a lower ratio h/t for thicker plates, Atzori et al. [4, 5] linked the size effect to the weld leg length h, due to the strong effect of the local V-notch size on fatigue strength.

Fig.4 represents the curves proposed on this basis for evaluation of the variation of fatigue strength at 2 x10^6 cycles due to size effect on fillet welds in steel [4] as a function of the geometric stress **σG** applied to the joint. As already widely evidenced for cracks and for V-notches, the size effect phenomenon has two cut-off limits: the upper one for small sizes given by the strength of the unnotched material **ΔσG.AO= Δσg.AO** (in the case of welded joints, the fatigue strength of butt joints machined to remove the weld caps) and the lower one for large sizes, when a sufficiently large radius is present at the toe of the weld, given by the same strength reduced by the full sensitivity notch coefficient KtV. Since the weld toe radius depends on the manufacturing technology and not on the size of the weld, the lower bound will not be reached for conventional welding technologies, for which, as a consequence, the size effect is expected to be always fully effective. When welding or post-welding treatments able to increase the toe radius are adopted, the size effect can decrease or disappear, but only if the increase of the radius ρ is high enough to lower the theoretical stress concentration KtV below a given value. It has been found [5] that for fillet welds this limit can be expressed as a function of the weld bead size h:

$$K_{tV} \leqslant \left(\frac{h}{h_0}\right)^{0.326}$$

When this condition is satisfied, the fatigue strength at 2 x10^6 cycles of the welded joint can be estimated as $\frac{\Delta\sigma_{g.AO}}{K_{tV}}$

(being **Δσg.AO** the fatigue strength at 2 x10^6 cycles of the butt weld with bead removed). When this condition is not satisfied, smoothing the weld toe radius will have no effect on the fatigue strength of the weld.

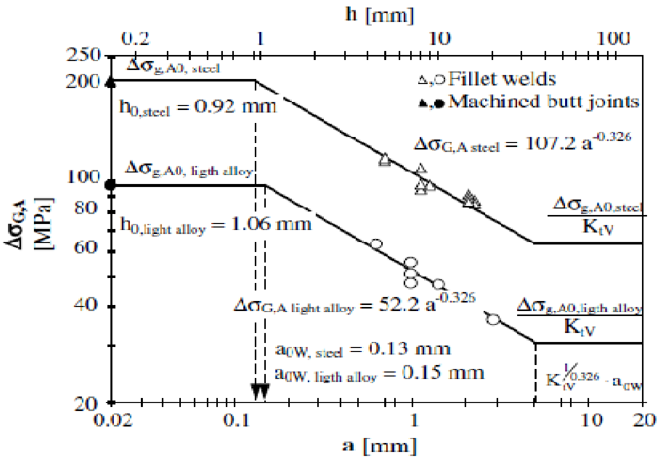

Fig. 4 Variation of the fatigue strength of fillet-welded joints.

According to the curves proposed in Fig. 4, the geometric fatigue strength ΔσG can be estimated as a function of the depth ''a'' of the local V-notch by means of the following relations:

For steel [4]:

$$\Delta\sigma_{G,A} = \frac{107.2}{a^{0.326}}\ \text{MPa}(N_A = 2\times10^6;\ P_s = 50\%;\ \text{depth } a \text{ in mm})$$

$$\Delta\sigma_{G,N} = \left(\frac{N_A = 2\times10^6}{N}\right)^{1/3}\Delta\sigma_{G,A}$$

For aluminium alloys [5]:

$$\Delta\sigma_{G,A} = \frac{52.2}{a^{0.326}}\text{MPa}(N_A = 2\times10^6;\ P_s = 50\%;\text{depth } a \text{ in mm})$$

$$\Delta\sigma_{G,N} = \left(\frac{N_A = 2\times10^6}{N}\right)^{1/4}\Delta\sigma_{G,A}$$

where N is the number of cycles to failure for PS = 50% under the stress range $\Delta\sigma_{G,N}$.

5. CONCLUSION

In fatigue design of complex welded structures the consolidated approach of nominal stresses, as given by Design Codes, is seldom applicable. This is due to two main reasons, one connected with the complexity of the structure, causing a difficult definition and evaluation of a nominal stress, the second connected with the complexity of the welded joint itself, causing some difficulties to single out a similar joint among the detail categories included in the design classes of the Codes.

To overcome this problem, the Eurocodes offer the possibility of applying the Hot Spot approach, which allows a better definition of a nominal stress (often called structural or geometrical stress in the literature), especially when FE analyses of the structure are employed. While this approach is satisfactory to overcome the problem connected with the complexity of the structure, it is not useful for the problem connected with the complexity of the joint. Also in the case of a joint which can be assimilated to one of the simple geometries considered by the Design Codes, care should be taken in defining the fatigue strength of the joint, since both the relative and absolute dimensions can have a strong effect.

In the last 20 years a new approach has been developed, today known as Notch-Stress Intensity Factor (NSIF) approach, which is not yet included in Design Codes, but is more powerful and reliable that more usual approaches. This method is based on the local stress and strain field close to the critical point, it can be applied also considering the stresses or the strains either at a fixed distance from the critical point or at the critical point, and several procedures have been developed which make its application not more difficult than the Hot Spot approach. Moreover the NSIF approach is much more reliable than the Hot Spot one, since it overcomes also the problem connected with the complexity of the joint itself, because the fatigue strength can be defined in such a way to be independent on relative and absolute dimensions.

The authors strongly believe that the NSIF approach is simpler and more reliable than the Hot Spot one and they hope that this synthetic presentation of the state-of-the-art could contribute to clarify the differences between the two approaches and a future inclusion of the NSIF approach in Design Codes.

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