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Several seam weld finite element idealizations challenged in fatigue within a French industrial collaborative workgroup.

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Abstract

In industry with fatigue strengthening concern, there are as many ways to make an FE seam weld idealization as factories involved in these types of simulations. Tens of ways to model a seam weld are available as soon as it is necessary to get out of direct connection assumption and if it is needed to take into account local weld stiffness and secondary bending effect of the fillet weld. Then it becomes mandatory to be able to discriminate such idealization processes while evaluating associated life prediction accuracy and industrial idealization efficiency (automation). This concern leads the idea to create a French industrial workgroup including CETIM and about 20 industrial companies, members of the "Mobile Machinery Program Committee". The main goal of this collaborative project being to challenge well documented and most promising seam weld idealization models (with associated methods) to converge at the end towards the most capacitive ones (with better results on given fatigue metrics).
As a first part of the multi-partner project work, a complete and precise technical review was achieved, giving a state of the art of the idealization models and methods available. Then the workgroup started extensive comparisons between well documented fatigue tests on seam welded components and associated FEA for some retained models and methods (Fayard, Lohr, IIW Hot Spot Stress, Notch Stress). Several components were considered, with more than 100 fatigue test results. This paper presents most of the obtained results, the Round Robin being still on-going. Preliminary results comparisons demonstrate general applicability of classical methods introduced in the standards or recommendations, these being also in most cases conservative. Results of this work aim to help choosing the right methodology, depending of the seam weld configuration and the in service loadings. It is also intended to try building partnerships with FE software editors to include most efficient methodologies in an automated way, making less tedious the seam weld modelling task on huge chassis frames. Small and medium factories should then reach an efficiency gain and improved accuracy level when building their virtual seam welded frames with new automated scripts integrated in their own FEA solutions.

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1. Introduction

1.1. Context

Industries from Public Works, Mining-Drilling, Handling-Lifting-Storage and Agricultural Machinery committees design and manufacture large and heavy metallic structures that must meet high requirement levels of reliability and safety. These structures often consist of components shaped and assembled mainly by welding. Although this process is widely used in industry, offering the possibility to produce structures with complex geometries, it appears however that the welded zones represent weak points with respect to fatigue resistance, due to geometric and/or structural discontinuities and residual stresses induced.

It is established that the fatigue strength of welded joints mainly depends on:

- The weld's geometry, introducing stress and strain concentrations at the toe and the root,

- Residual stresses associated with temperature gradients and phase changes due to the welding operation and overall average stress level in the structure,

Several fatigue design methodologies for welded joints are now available. They allow designers to evaluate the risk of fatigue failure by comparing different damage parameters, usually obtained by a structural calculation with an appropriate fatigue criterion. From a numerical point of view, these methods are implemented through different modeling strategies. Manufacturers need to have a panorama of these methods and techniques especially in finite element modeling and particularly in the case of shell models. This type of modeling is commonly used in industry, particularly for the advantage in model size and time calculations. However it relies on many simplifying assumptions that require a matching between the computed stresses and strains (not necessarily realistic) and associated analysis criteria in static and fatigue loading.

In addition, the designer rarely models the seam weld details (size, penetration, local geometry…), considering often the welded joint by coincident nodes, whereas this can have a significant influence on the transfer of loads.

1.2. Objectives of the study

The French multi-partner project aims to:

- Realize a state of the art on methods and techniques of modeling seam welded joints usable in an industrial context,
- Select some of these methods considered as most relevant and propose application cases to compare predicted fatigue lives to test ones,
- Allow, through the working group, exchanges between manufacturers on this theme of modeling welded joints and share experiences,
- Point out methodologies compatible with industrial constraints and tools, which can be recognized by the standard and legal authorities,
- Build partnership with FE and fatigue software editors in order to implement identified methodologies in an efficient industrial process.

This paper presents part of the results obtained during this still on-going project. It gives first a brief and non-exhaustive overview of the literature on seam welds modeling techniques and fatigue damage calculation methods, then comparisons of predicted lives to fatigue test lives are made for some selected approaches.

2. Fatigue assessment methods and welds idealization techniques

2.1. Background

Taking into account fatigue of materials in design of components or structures is a complex task, made even more complex when it comes to welded parts where the assembly process introduces factors influencing drastically the fatigue strength. Different approaches are available for fatigue analysis of welded joints, which are distinguished mainly by the damage parameter used. Among those are:

- The nominal stress approach, a global method, which uses the concept of nominal stress ranges $\Delta\sigma_n$, determined from loads and related cross section properties, and a set of S-N curves corresponding to the main constructive details representative of the welded connection. This is a basic and widely used design method, based on extensive laboratory fatigue test results aiming to provide probabilistic design S-N curves (representing 97.7% probability of survival), which is implemented in standard design codes or recommendations (e.g. EUROCODE [1], IIW[2]…).

- The structural stress (also called hot-spot stress or geometric stress) approaches, taking into account only the part of stress concentration related to the structural geometry (macro-geometry) but not the local discontinuities of the welded joint. This may be justified by the fact that exact and detailed geometry of the weld is not known accurately at the design phase. The non-linear notch effect is somehow included in the S-N curve through the test results scatter. Thus one S-N curve for a specific weld type can be used for any detail having similar properties to the weld type associated, unlike for the nominal stress approach where the designer needs finding the right S-N curve associated with the investigated detail.

- The use of local stresses or strains, taking into account local notch effect and possible occurrence of plasticity. These approaches allow considering the physical local aspects of fatigue damage, i.e. initiation of fatigue damage at a local level in critical locations. Damage calculations may be based on design curves given by standards or obtained by fatigue tests on parent material. Let's recall that usage of parent material data needs including in the model effects due to the welding process: material heterogeneity (base metal structure altered in the heat affected zone), residual stresses and distortions, welding defects and imperfections… witch is not a trivial task.

- The application of Fracture Mechanics concepts, allowing computation of propagation life from an initiated defect. Crack propagation approach uses special parameters such as the J-integral or the range of stress intensity ΔK to describe the crack propagation rate (increase of crack length per loading cycle).

As illustrated in Figure 1 according to Marquis and al. [3], the level of complexity and accuracy varies according to the method adopted. Search for maximum precision often requires a significant effort in terms of modeling.

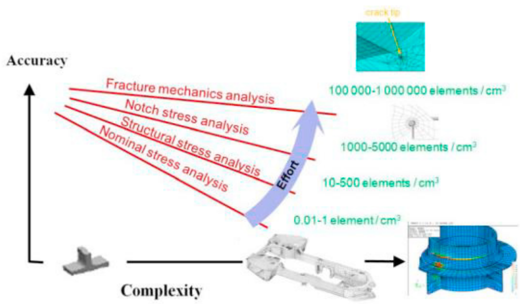


Fig. 1: Qualitative view of accuracy and modeling effort for different fatigue assessment methods dedicated to welded structures

Based on these different concepts, tens of methods have been developed and presented in extensive literature.
The following sections will concentrate on some structural and local approaches identified by the French working group as potential welds idealization and calculation techniques relevant for the participant's industry sectors.

2.2. Structural hot spot stress approach according to IIW [2]

This approach was initially used for welded tubular joints in the offshore sector and pressure vessels analysis since the 1960's; then extended to the case of plated welded structures. It seems that the "hot spot" denomination comes from the fact that the application of high level cyclic loads induces at the critical point (stress concentration zone) a heating of the material. In fact, the hot spot is simply the critical area where the initiation of the fatigue crack is likely to occur.

The hot spot stress corresponds to the maximum principal stress in the base metal at the weld toe, taking into account the effects of stress concentration due to the overall geometry of the considered detail, but excluding the local stress concentration effects due to weld local geometry and discontinuities. It is applicable to fatigue failure at weld toes. Traditional approach to derive this stress is based on linear or quadratic extrapolation of strains or stresses from two or three reference points at certain distances from the weld toe. The non-linear notch stress effect, not included in the structural hot spot stress definition, is considered to vanish within a distance 0.3t to 0.4t from the weld toe, t being the plate thickness. That's why IIW recommends making extrapolation from points witch distance from the weld toe is over 0.4t (e.g. 0.4t/1t or 0.4t/0.9t/1.4t). At plate edges, quadratic extrapolation is made from reference points at fixed distances from the weld toe (4, 8 and 12mm).
Figure 2 illustrates the structural hot spot stress definition from stress extrapolation at the plate surface.

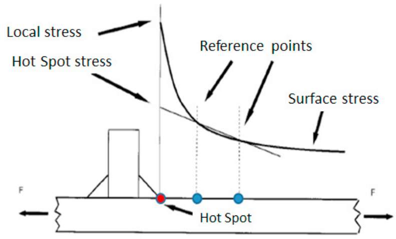


Fig. 2: Sample surface extrapolation of hot spot stress.

The structural hot spot stress may be calculated by finite element analysis either from surface extrapolation described above or alternatively by stress linearization over the thickness leading to the exclusion of the local stress peak in plate or shell structures. In that case the sum of membrane stress (σ_m) and bending stress (σ_b) components gives to the hot spot stress. Figure 3 illustrates the linearization approach.

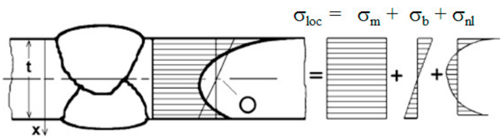


Fig. 3: Stress distribution over plate thickness

Detailed rules for finite element modelling and stress evaluation are given in the IIW Fatigue Recommendations publications [2]. In general, to limit modelling and computational efforts, simple models and relatively coarse meshes are preconized. Models with either thin plate or shell elements or with solid elements may be used.

When using plate or shell elements, these are arranged in the mid plane of the plates. In simplified models, the welds may be omitted except for cases where the results are affected by local bending, due for example to plate offsets or to interaction between welds close to each other. In such cases, the welds may be modelled by vertical or inclined plate elements having appropriate stiffness or by introducing constraint equations or rigid links to couple node displacements.

For complex cases, solid elements may be used, allowing the weld to be modelled with prismatic elements. If isoparametric 20 nodes elements are used, one element is sufficient in thickness direction due to the quadratic displacement function and linear stress distribution. By reduced integration, the linear part of the stresses can be directly evaluated at the surface and extrapolated to the weld toe.
It should be noted that when the weld is not modelled, extrapolation has to done to the intersection point.
Figure 4 illustrates typical finite element models and extrapolation paths.

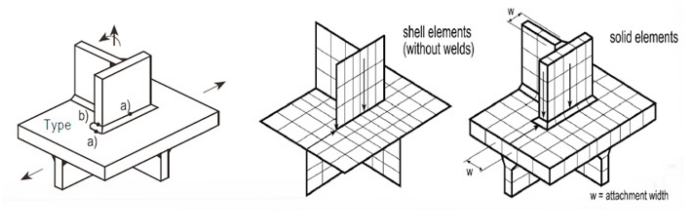


Fig. 4: Types of hot spots and sample finite element models according to IIW

Figure 5 summarizes recommendations on meshing and extrapolation.

Type of model and weld toe		Relatively coase models		Relatively fine models	
		Type a	Type b	Type a	Type b
Element size	Shells	t x t max t x w/2*)	10 x 10 mm	≤0.4 t x t or ≤0.4 t x w/2	≤ 4 x 4 mm
	Solids	t x t max t x w	10 x 10 mm	≤0.4 t x t or ≤0.4 t x w/2	≤ 4 x 4 mm
Extra-polation points	Shells	0.5 t and 1.5 t mid-side points**)	5 and 15 mm mid-side points	0.4 t and 1.0 t nodal points	4, 8 and 12 mm nodal points
	Solids	0.5 and 1.5 t surface center	5 and 15 mm surface center	0.4 t and 1.0 t nodal points	4, 8 and 12 mm nodal points

*) w = longitudinal attachment thickness + 2 weld leg lengths
**) surface center at transverse welds, if the weld below the plate is not modelled

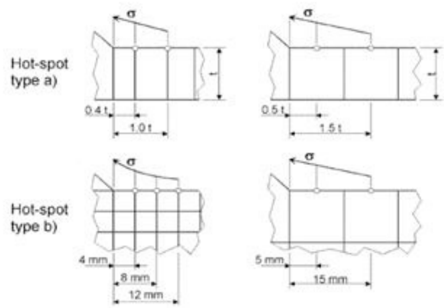


Fig. 5: Recommendations on meshing and extrapolation according to IIW

Once the structural hot spot stress is evaluated from an appropriate model, fatigue life calculations are carried out in the classical way thanks to a stress life approach. Each weld type is associated to a design S-N curve established from extensive fatigue tests. Fatigue class FAT 100 (range of allowable stress or fatigue strength reference value at 2.10^6 cycles in MPa) is recommended in normal cases for steel structures; except for load-carrying fillet welds or long attachments at plate edges (>100mm) for which FAT 90 applies.

2.3. Structural stress approach according to Fayard and al.

This approach was developed by Peugeot SA in partnership with Ecole Polytechnique for thin sheets [4]. The method uses a design structural stress calculated at the weld toe from a thin shell finite element model, where welds are modelled by rigid elements. This ensures the displacement compatibility in the joint zone, reproduces the local rigidity induced by the weld and also simulates the force flow from one sheet to the other through the weld.
The plates are represented by 4 nodes thin shell elements, placed on their mid planes. The thickness of the elements corresponds to those of the sheets. The size of the shell elements in the joint zone is defined such that the centers of gravity or Gauss points where the stresses are evaluated are located at the weld toe in order to avoid interpolations or extrapolations to the hot spot. The size and positioning of the shell elements have to take into account the weld leg length and the sheet thicknesses. Typically, the element size roughly equals the weld leg length.
Meshing rules are described in figure 6 for two seam-welded joints commonly encountered in automobile design. The stresses are evaluated at the center of gravity of each shaded element (post-processed elements).

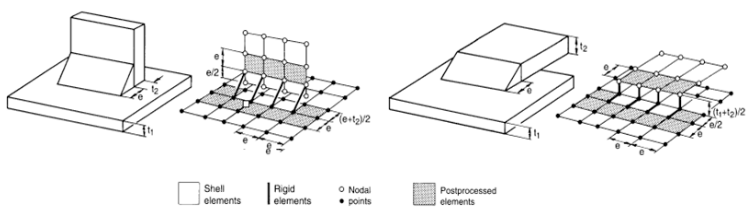


Fig. 6: Common welded joints idealization according to Fayard

The method uses a uniform hot spot structural stress *S–N* curve independent of the geometry of the structural member and the applied loading. The design stress is either defined as the maximum principal structural stress amplitude at the hot spot (like in the classical hot spot stress according to IIW) or as a multiaxial equivalent stress calculated according to the Dang Van criterion [5]. The latter is a linear combination of local shear stress and hydrostatic pressure, aiming to take into account complex multiaxial loadings.
As for the IIW structural host spot stress, the Fayard method considers only fatigue evaluation at weld toes.
Figure 7 shows the S-N curves used for each definition of the design stress, with differentiation between high and low strength steels (tension yield limit σ_y).

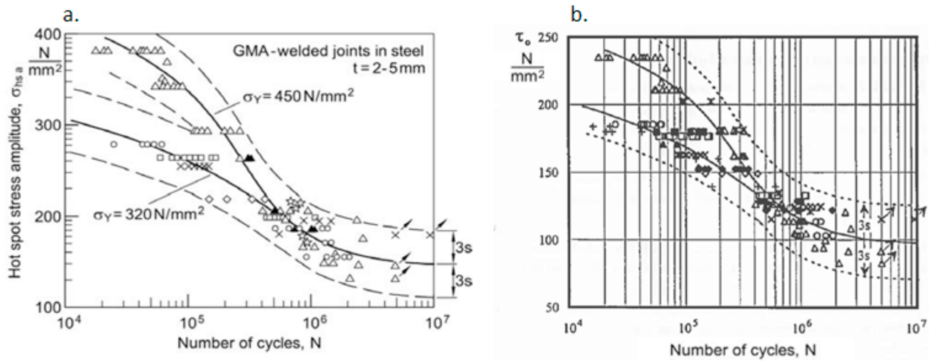


Fig. 7: Structural stress S-N curves with maximum principal stress amplitude (a) and Dang Van equivalent stress (b).

2.4. Structural stress approach according to Turlier and al.(Lohr method)

Turlier and al. [6] developed, an approach dedicated to heavy transportation vehicles (Lohr Industry), allowing calculations of two types of structural stresses depending on the fatigue evaluation location, i.e. weld toe or weld root.
Assembly modelling is made through a hybrid idealization technique combining shell elements, rigid elements and multi-points constraints.
The parent material parts are represented by shell elements placed at their mid planes. The thickness of these elements corresponds to the plates ones.
The weld is modelled by inclined shell elements, linked to the plate by rigid elements and multi-point constraints. The connection of the weld leg lines to the metal plates is performed using the FEA gluing technique without additional rigidity. Figure 8 illustrates this modelling: the weld leg node (A) is connected by a rigid 1D element (RBE) to a projected node on the plate (B) and a multi-point constraint (MPC) connects (B) to the element of the metal plate. The weld element thickness must correspond to the weld throat size to ensure that the model stiffness is equivalent to the real seam weld one.
This technique may be applied to different connections types.

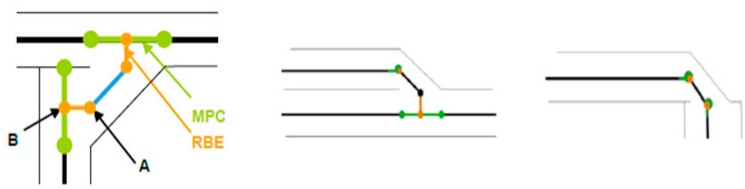
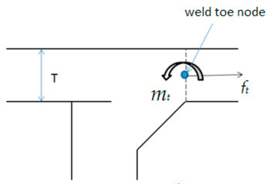


Fig. 8: Weld joint idealization according to Turlier and al.

The design structural stress is calculated at each location of interest (weld toe or root) from nodal line forces and moments.
The structural stress $\sigma_{s,t}$ at the weld toe is calculated from the line force normal to the weld toe line f_t and the line moment m_t tangent to the weld toe line, as described in Figure 9. T being the plate thickness.



$$\sigma_{s,t} = \sigma_m + \sigma_b = \frac{f_t}{T} + \frac{6 \cdot m_t}{T^2}$$

Fig. 9: Structural stress at weld toe line.

The structural stress calculation by combination of the membrane normal stress and the bending normal stress has been extended to the weld leg section in the continuation of the root face by Fricke [7] for weld root fatigue assessment when the weld throat is subjected to bending. This approach is based on the linearization of the normal stress σx(z) within the weld leg section, as illustrated figure 10 a. The root structural stress is calculated at node A from nodal forces and moments like for the weld toe, l being the weld leg length, see figure 10 b.

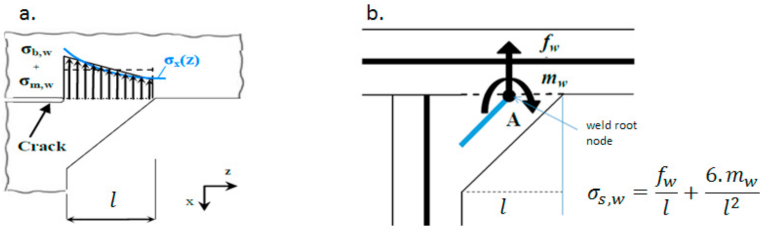


Fig. 10: Stress distribution at weld leg (a) and structural stress at weld root (b).

Once the structural stresses calculated, fatigue evaluation is done thanks to a stress life approach. Based on the work of Fricke and al., authors recommend to use FAT 90 or FAT 100 S-N curves at the weld toe, depending if the weld is carrying load (FAT 90) or not (FAT 100). For fatigue evaluation at weld root, FAT 80 is used.

2.5. Effective Notch stress approach

The effective notch stress concept first introduced by Radaj [8], considers the increase in local stress at the notch formed by the weld toe or the weld root, based on theory of elasticity (elastic material behavior). This approach is mainly used in the form of fictitious notch rounding, the basic idea behind it being that the stress reduction in a notch due to averaging the stress over a certain depth can alternatively be achieved by a fictitious enlargement of the notch radius.
For structural steels and aluminum alloys an effective notch root radius, also called reference radius, of r = 1 mm has been verified to give consistent results for sheet thickness greater than 5mm. For thinner plates, a reference radius of 0.05 mm is recommended.

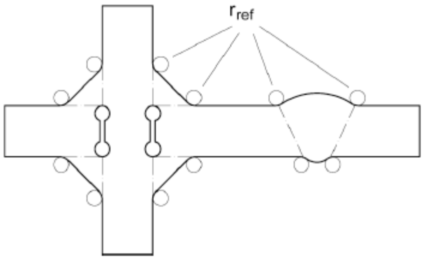


Fig. 11: Fictitious notch rounding at weld toes and roots, according to Hobbacher [2].

Usually, the notch stress is computed from a finite element model. Stresses may be solved by 3D or a 2D analysis, the latter being restricted to cases where variations of the geometry and loading in the 3rd direction can be neglected. In this case, plane strain conditions are usually assumed.

The discretization of the structure is normally performed with a relatively coarse overall mesh, which is locally refined near the notches under consideration. The mesh should be gradually refined towards the notched area, avoiding large steps in element size and excessive element distortion. Figure 12 shows a typical example for a weld root modelled with a keyhole notch [2].

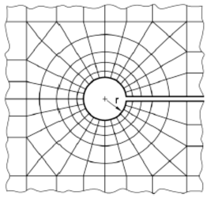


Fig. 12: Typical mesh example for the notch stress at a weld root.

For the determination of effective notch stress by FEA, element sizes of not more than 1/6 of the radius are recommended in case of linear elements and 1/4 of the radius in case of higher order elements. These sizes have to be observed in the curved parts as well as in the beginning of the straight part of the notch surfaces in both directions, tangential and normal to the surface. Notch stresses are evaluated on the rounded surface of the notch, using the tangential and normal stresses in the section and the shear stresses on the notch surface. From these, principal or equivalent stresses can be derived. More details are available in [2].

Fatigue evaluation uses an S-N curve dependent of the material and reference radius used. This is summarized in table 1.

Table 1: FAT class to use in conjunction with notch stress approach

Material	Characteristic fatigue strength for r_{ref} = 1 mm	Characteristic fatigue strength for r_{ref} = 0.05 mm
Steel	FAT 225	FAT 630
Aluminium alloys	FAT 71	FAT 180

3. Demonstration cases

In this paper are presented applications of the idealization techniques and fatigue evaluation methods described in section 2 to five types of welded components for which fatigue test results are available. Some of the analyzed results are from tests at CETIM and others are from well documented literature. In this part of the multi-partner project no stress or life correction has been applied attached to the methods used (R; thickness, gradient, analytical correction…).

3.1. K type welded connection

The component considered is a common welded joint encountered in crane industry. It is a K type welded joint, where two inclined rectangular tubes of thickness 6mm are welded to a 1500mm length frame hollow section of 15mm thickness and squared section. The component is made of general construction steel grade, S355. The weld throat dimension is 6mm. The component is loaded in bending (4 point bending). Figure 13 shows the geometry characteristics, loading conditions and test bench.

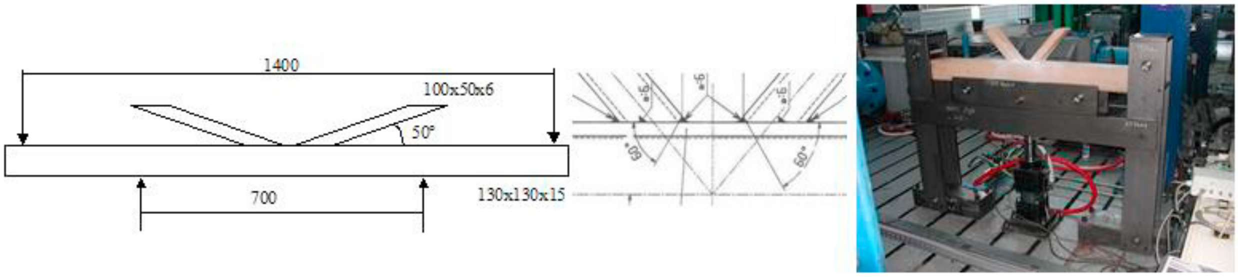


Fig. 13: geometry characteristics of the component and test bench

21 components have been fatigue tested by CETIM at constant amplitude loading with stress ratio R = 0.1. Three load levels (7 tests per level) allowing covering the LCF and HCF fatigue domains are applied, corresponding to maximum loads of 644 kN, 245 kN and 190 kN . Figure 14 presents the S-N curve obtained, in terms of nominal stress range in the main frame (squared hollow section) versus number of cycles to failure.

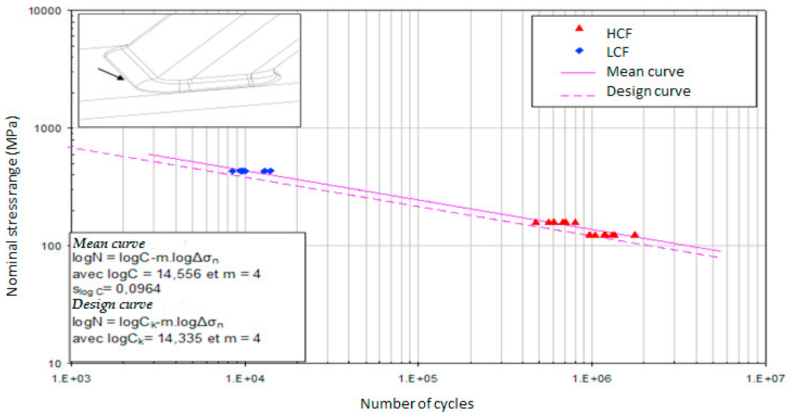


Fig. 14: Fatigue tests results

All failures occur at the weld toe, in the main frame as indicated by the arrow figure 14.
All methods presented in section 2 may be used as they all consider weld toe failure. As it may be too long to present all modelling details for all compared methods, only views of the models and comparison of results are given.

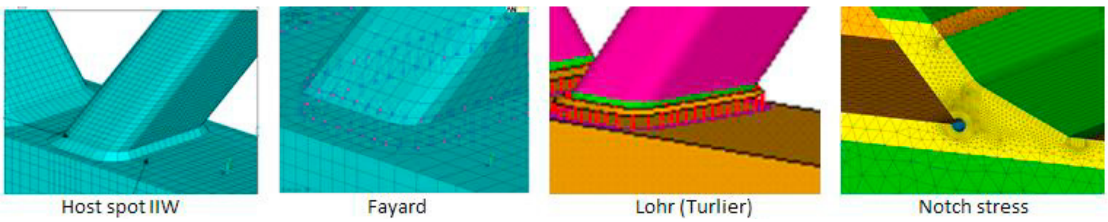


Fig. 15: view of the models.

Calculated stress ranges and corresponding lives are given in table 2. Comparison with test life is made through a life ratio LR = (calculated life/test life).

Table 2: Comparison between tests and predictions

Nominal stress range from test	Method	Structural stress range (MPa)	Life ratio LR
$\Delta\sigma_n$ = 125 MPa	Hot Spot (IIW)	175	0.31
	Fayard	173	>10
	Lohr	178	0.29
	Notch stress	419	0.35
$\Delta\sigma_n$ = 160 MPa	Hot Spot (IIW)	226	0.39
	Fayard	222	>10
	Lohr	229	0.38
	Notch stress	540	0.45
$\Delta\sigma_n$ = 430 MPa	Hot Spot (IIW)	594	1.13
	Fayard	586	>5
	Lohr	603	1.08
	Notch stress	1420	1.30

In the HCF regime, results show that all methods apart Fayard are on the conservative side. Nearest results to the tests are obtained with the notch stress approach. In the LCF regime, all methods are on the unsafe domain. Life ratios calculated are not really relevant as these methods are more dedicated to middle or high cycle fatigue. Tests at LCF level were done to allow investigation of local strain method, not presented in this paper.
Let's recall that in this case, the numerical model has been compared to strain gages measurements for the hot spot stress approach, showing less than 15% difference between calculation and measure.

3.2. Rectangular hollow section welded to an intermediate plate

This example is from literature and completely described by Fricke in [9].Two rectangular hollow sections of dimensions 120 mm x 80 mm x 6 mm are connected to a 15mm thick intermediate plate by a single-sided fillet weld. The weld throat dimension is 4.5mm. . The component is made of mild steel S235JRG2.
Two load cases are investigated: tension and 3 points bending of the rectangular section. Figures 16 and 17 show the geometry and loading characteristics.

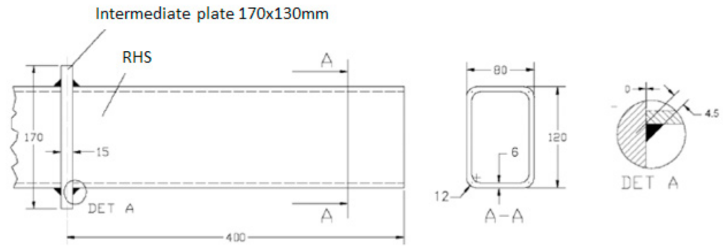


Fig. 16: component geometry

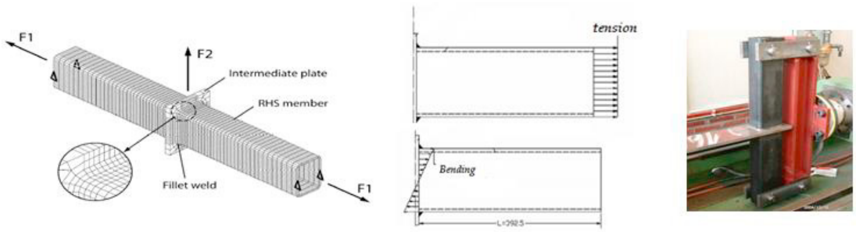


Fig. 17: loading conditions

Figure 18 show the fatigue tests results in constant amplitude tension and bending at a stress ratio R=0.5. A total of 21 tests are reported, 8 in tension and 13 in bending. The S-N curves give the allowable nominal stress range in the hollow section versus number of cycles to failure.

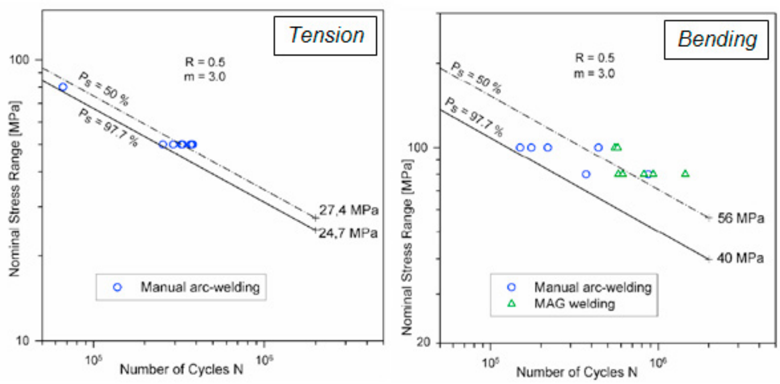


Fig. 18: Fatigue tests results

In this case all failures occur at the weld root. The relevant methods dealing with such failure types are the Notch Stress and Lohr approach. Figure 19 and table 3 give respectively views of the models and comparison of calculated and test lives.

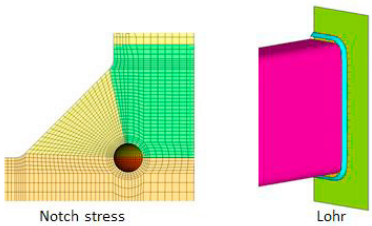


Fig. 19: view of the models

Table 3: Comparison between tests and predictions

Nominal stress range from test	Method	Structural stress range (MPa)	Life ratio LR
Tension $\Delta\sigma_n$ = 50 MPa	Lohr	189	0.63
	Notch stress	459	0.98
Bending $\Delta\sigma_n$ = 77 MPa	Lohr	240	0.36
	Notch stress	507	0.58
Bending $\Delta\sigma_n$ = 97 MPa	Lohr	280	0.36
	Notch stress	658	0.62

Both methods give conservative results in all cases, with the notch stress approach being more accurate.

3.3. Fillet weld around attachment end

This example, from literature, is completely described by Fricke in [9]. The specimen, made from S355 steel, has been designed for the investigation of weld toe and root failure at attachment ends. The plate's thicknesses are 12mm and the weld throat is 4mm. The fatigue-critical detail is the soft nose of the right attachment as illustrated in figure 20, where the geometry and boundary conditions are described.

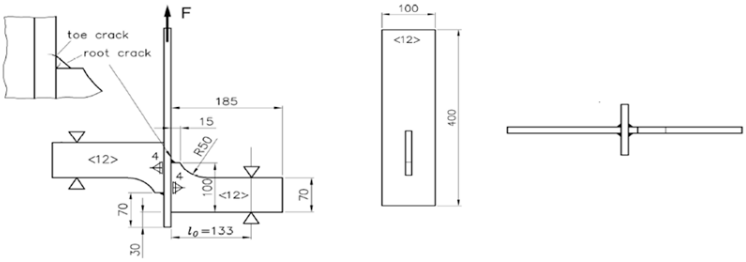


Fig. 20: Geometry and loading characteristics of the analyzed component

40 specimens have been tested at three different constant amplitude levels and two stress ratios R=0 and R=0.5. Figure 21 summarizes the test results in terms of nominal weld stress range in the central plate versus life.

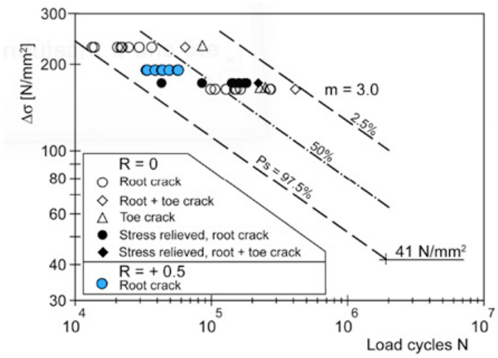


Fig. 21: Fatigue tests results

Weld root failure was observed for almost all specimens. Then the Notch stress and Lohr methods will be compared for this case. Views of models and results comparison are reported Figure 22 and table 4.

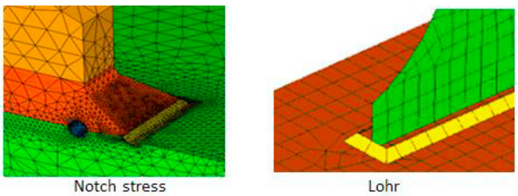


Fig. 22: view of the models

Table 4: Comparison between tests and predictions

Nominal stress range in plate	Method	Structural stress range (MPa)	Life ratio LR
$\Delta\sigma_n$ = 20 MPa, R=0	Lohr	131	>10
	Notch stress	690	1.57
$\Delta\sigma_n$ = 28 MPa, R=0	Lohr	184	>10
	Notch stress	965	3.60
$\Delta\sigma_n$ = 23.5 MPa, R=0.5	Lohr	154	>10
	Notch stress	810	2.02

Both methods give non conservative results in all cases.

3.4. Lap joints and cover plates examples

These cases are from literature and completely described by Fricke and Feltz in [10]. Types of joints investigated are cover plates, having partial-load carrying fillet welds, and lap joints having full-load carrying fillet welds. The thickness of all plates is 12 mm, while two throat thicknesses with 3 and 7 mm were analyzed. Material is an S355 steel. Figure 23 shows the components geometry and loading.

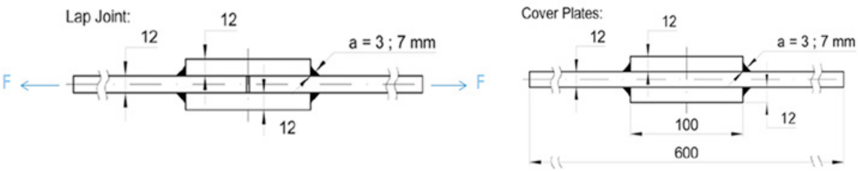


Fig. 23: Geometry and loading of the lap joints and cover plates

40 specimens have been tested in fatigue at stress ratio R=0 and constant amplitude levels corresponding to nominal stress ranges in the central plate ranging from 90 MPa to 210 MPa. Figure 24 shows the obtained S-N curves.

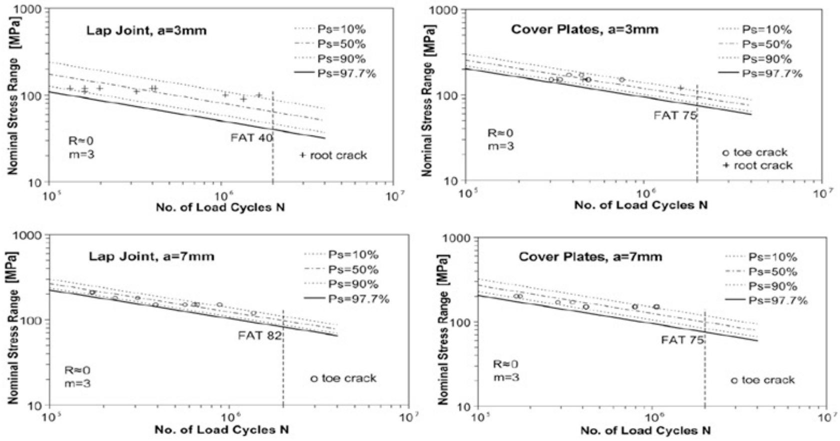


Fig. 24: Fatigue tests results for lap joints and cover plates

For the lap joint with 3mm weld throat (L3), all failures occur at the weld root. For all other configurations (L7, C3 and C7) fatigue cracks at toes were observed. Table 5 summarizes the results for the relevant models.

Table 5: Comparison between tests and predictions

Nominal stress range from test	Method	Structural stress range (MPa)	Life ratio LR
L3, $\Delta\sigma_n$ = 120 MPa	Lohr	271	0.69
	Notch stress	950	0.36
C3, $\Delta\sigma_n$ = 150 MPa	Hot Spot (IIW)	154	1.60
	Fayard	152	>10
	Lohr	150	1.73
	Notch stress	560	0.52
L7, $\Delta\sigma_n$ = 150 MPa	Hot Spot (IIW)	160	1.09
	Fayard	152	>10
	Lohr	152	1.27
	Notch stress	514	0.51
C7, $\Delta\sigma_n$ = 150 MPa	Hot Spot (IIW)	160	1.42
	Fayard	152	>10
	Lohr	152	1.66
	Notch stress	430	1.15

For weld root cracking on the L3 configuration, the Lohr and Notch stress approach lead to conservative results.

In all other configurations, the Hot Spot stress, Fayard and Lohr are in the unsafe domain, whereas the Notch stress approach remains mainly in the conservative domain.
All structural stress approaches (hot spot, Lohr, Fayard) are unable to catch the stress gradient close to the weld toe in a pure tensile flange configuration and are unsafe. Only notch and nominal stress method are able to succeed in this particular case.

4. Conclusion

Several seam welds idealization techniques and fatigue damage calculation methods have been presented and compared to physical fatigue tests. Figure 25 synthetizes the overall results of comparison between test and predicted lives.

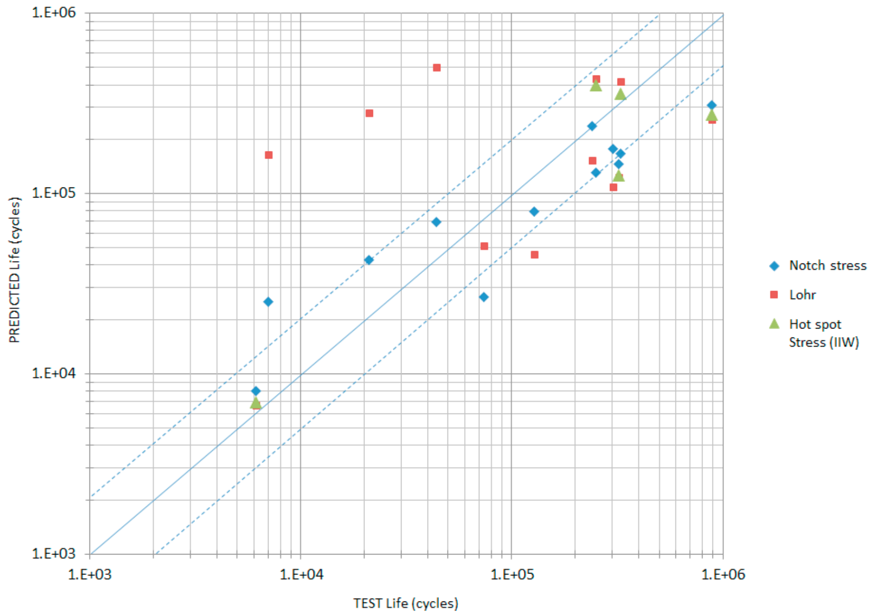


Fig. 25: Overall view of results with no additional correction (R, thickness, gradient…)

It appears that nearly all results for the Notch stress and Hot spot stress according to IIW are in a scatter band of factor 2 on tested and predicted lives, with most results in the conservative side. The Lohr method results are mitigated, some being in this scatter band and some being non conservative, particularly for the fillet weld around attachment end where no method is really able to predict safe side root life cracking. The Fayard method, not included in this graph, showed very non conservative results for the analyzed components. At its discharge the method has been developed and validated for thin welded sheets, which is not the case of all considered examples. The variant of the method using a Dang Van stress parameter has not been investigated.

Future work aims to complete these comparisons with other demonstration examples and also include methods not investigated such as "in-house" industrial methods or other available approaches like the Battelle Dong method [11].

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References

[1] Eurocode 3: Design of steel structures - Part 1-8: Design of joints; NF EN 1993-1-9, 2005
[2] Recommendations for fatigue design of welded joints and components; A. Hobbacher, IIW-1823-07/XIII-2151r4-07/XV-1254r4-07, 2008.
[3] Recent developments in the fatigue design and assessment of welded structures; G. Marquis, Fatigue Design 2009 November 25 & 26 – Cetim, Senlis – France.
[4] Assessment of welded structures by a structural multiaxial fatigue approach; K. Dang Van, A. Bignonnet, JL. Fayard, Biaxial/Multiaxial Fatigue and Fracture, ESIS Publication 31.
[5] On a new fatigue criterion. Theory and application. K. DANG VAN, B. GRIVEAU, 0. MESSAGE, Biaxial and multiaxial fatigue, EGF3, Mechanical Engineering Publications, 1989.
[6] New FEA shell element model for seam weld static and fatigue structural assessment; D. Turlier, P. Klein, L. De Nonia, D. Lawrjaniec, Fatigue design 2011 – November 23 & 24 – Cetim, Senlis – France.
[7] Fatigue assessment of root cracking of fillet welds subject to throat bending using the structural stress approach; W. Fricke, A. Kahl, H. Paetzold, IIW Doc. XIII-2072-05 / XV-1198-05.
[8] Design and analysis of fatigue resistant welded structures, D. Radaj, Abington Publishing, Abington Cambridge, U.K. 1990.
[9] IIW Guideline for the Assessment of Weld Root Fatigue, W. Fricke, IIW-Doc. XIII-2380r3-11/XV-1383r3-11, 2012.
[10] Fatigue Tests and Numerical Analyses of Partial-Load and Full-Load Carrying Fillet Welds at Cover Plates and Lap Joints; W. Fricke, O. Feltz, IIW Doc. XIII-2278-09/XV-1320-09.
[11] A structural stress definition and numerical implementation for fatigue analysis of welded joints, P. Dong, International Journal of Fatigue, vol.23 Number 10 Nov. 2001.