PAPER • **OPEN ACCESS**

Francis-99: Evaluation of the strain energy density value for welded joints typical of turbine runner blades

To cite this article: Pietro Foti and Filippo Berto 2019 *J. Phys.: Conf. Ser.* **1296** 012007

IOP Conf. Series: Journal of Physics: Conf. Series **1296** (2019) 012007 doi:10.1088/1742-6596/1296/1/012007

Francis-99: Evaluation of the strain energy density value for welded joints typical of turbine runner blades

Pietro Foti, Filippo Berto

Norwegian University of Science and Technology, Trondheim, Norway

E-mail: pietro.foti@ntnu.no

Abstract. The main aim of this work is to investigate the fatigue behaviour of welded joints through an energetic approach based on the Strain Energy Density failure criteria. The geometries, taken from the literature, are typical of turbine runner blades. The results of the fatigue tests on these details were summarised through the Strain Energy Density approach. The application of this method to these geometries is the first step of a wider research with the aim to provide a suitable tool in FEM code for the lifetime estimation of components characterised by complex geometries.

1. Introduction

Dealing with welded joints the design guides suggest the use of global methods [1–3] for the fatigue assessment even if the nature of fatigue phenomenon is local.

The most used method for the fatigue design is the nominal stress method that considers external loads or nominal stresses in the critical cross-section and compares them with the S-N curves that correlate the fatigue strength with the number of cycles.

In order to assess the fatigue life of a component, the standards define the permissible value of the nominal stress ranges at 2 million of cycles for the joint considered, the so-called FAT, whose value is defined on the base of the shape, the loading conditions and the most relevant geometrical parameters of the detail considered.

Even if this method does not allow to perform a precise fatigue assessment, resulting in an excessive conservative design, it represents, still nowadays, the base for fatigue assessment in almost all areas of mechanical and structural engineering due to its relative simplicity and statistical proof. However, its validation is based on tests carried out on geometry and conditions that are rarely encountered in practical applications whose assessment, in this way, lacks actually a statistical validation. A more precise evaluation of the expected fatigue life can be assessed through the local approaches [4]. The use of these approaches for welded joints, even if they generally consider some welding characteristics only in a statistical way, allows the user to perform a realistic evaluation of the fatigue strength with a relatively simple approach. On the other hand, a practicable application of the local approaches requires the determination of those parameters that have a decisive influence in the fatigue strength in order to avoid complicating, even more, the problem. In order to verify the feasibility of the local approaches to assess the fatigue behaviour of complex components, the present work summarises, through an energetic approach, some experimental fatigue tests from the literature carried out for the fatigue assessment of turbine runners.

2. Strain Energy Density Method

The Strain Energy Density (SED) method is an energetic local approach that has been validated as a method to investigate both fracture in static condition and fatigue failure [5–7].

Dealing with V-notches, the brittle fracture can be assumed to occur when the local SED, $\overline{W}$, evaluated in a given control volume, reaches a critical value $\overline{W} = W_C$ independent of the notch opening angle and of the loading type [6]. In 2D problems, the control volume has a circular shape with radius R_0 dependent only on the material and its value decreases with increasing brittleness. The averaged SED critical value, W_C, is evaluable in the case of an ideally brittle material through the following expression:

$$W_C = \frac{\sigma_t^2}{2E} \tag{1}$$

Where σ_t is the conventional ultimate tensile strength and E is Young's modulus.

The concepts stated above represent the basic idea of the SED method. More considerations about the analytic frame of this method are available in the literature [5–8].

In order to evaluate the averaged value of the strain energy density, a sector-shaped cylinder of radius R_0 along the notch tip line, called 'control volume', must be considered. In plane problems, both in mode I and mixed mode (I+II) loading, the control volume becomes a circle or a circular sector with radius R_0 respectively in the case of cracks and pointed V-notches, as shown in fig. 1.

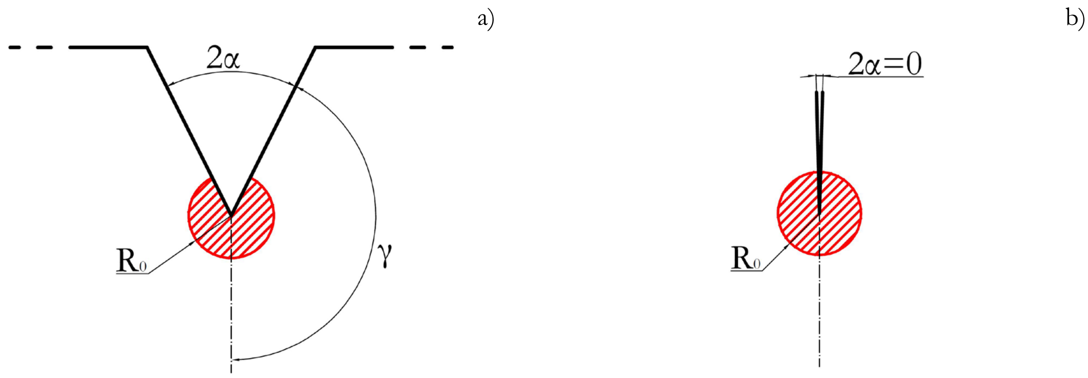

Figure 1: Control volume (area) for: a) sharp V-notch; b) crack.

The radius R_0 can be estimated for the crack case under plane strain and plane stress conditions as follows [9–11]:

$$R_0 = \frac{(1+\nu)(5-8\nu)}{4\pi}\left(\frac{K_{IC}}{\sigma_t}\right)^2 \qquad \textit{plane strain} \tag{2}$$

$$R_0 = \frac{(5-3\nu)}{4\pi}\left(\frac{K_{IC}}{\sigma_t}\right)^2 \qquad \textit{plane stress} \tag{3}$$

Where ν is Poisson's modulus and K_{IC} is the fracture toughness of the material analysed.

In the case of a pointed V-notch the critical radius can be assessed by [6]:

$$R_0 = \left[\frac{I_1 K_{1C}^2}{4\lambda_1(\pi-\alpha)EW_C} \right]^{\frac{1}{2(1-\lambda_1)}} = \left[\frac{I_1}{2\lambda_1(\pi-\alpha)} \left(\frac{K_{1C}}{\sigma_t} \right)^2 \right]^{\frac{1}{2(1-\lambda_1)}} \quad (4)$$

The values of λ, Williams' eigenvalues, and I_1, mode I function in the SED expression for sharp V-notches, depending on α, notch opening angle, and on the stresses field are available in the literature [8]. Dealing with blunt notches [9–13], as with the geometries analysed in the present work, it is important to do some considerations about the control volume that, under mode I loading, assumes a crescent shape, with R_0 being its maximum width along the notch bisector line [9,10], differently from pointed V notches. In this case, the control volume is given by the intersection between the component and a circle of radius $r + R_0$ centred on the notch bisector, between the notch edge and the notch-fitting radius centre, at a distance r from the notch edge.
Under mixed-mode loading, the maximum elastic stress is out of the notch bisector line and its position along the notch edge is a function of mode I and mode II stress distributions. In this case, the control volume is no longer centred with respect to the notch bisector, but rigidly rotated with respect to it and centred on the point where the SED reaches its maximum value [14–19], following, essentially, the mode I dominance concept.

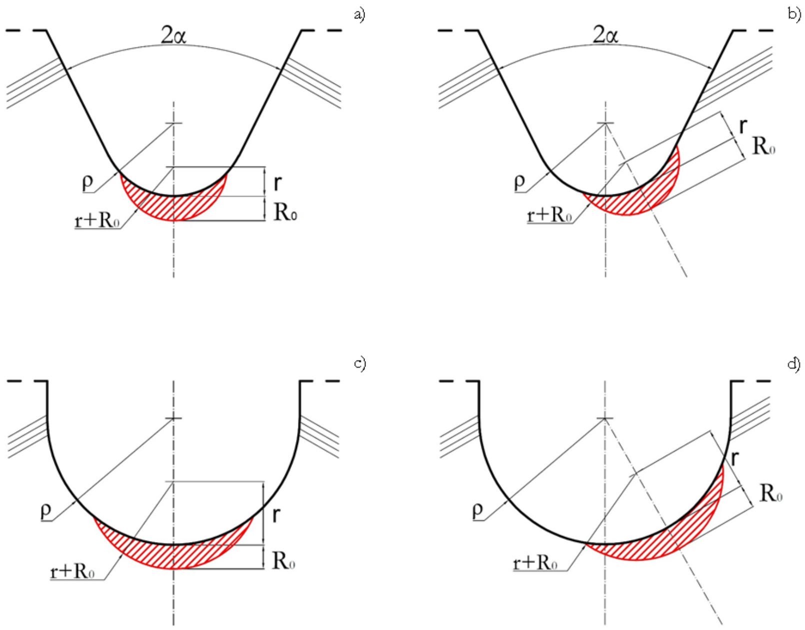

Figure 2:Control volume for: a) blunt V-notch under mode I loading ($r = \rho \cdot (\pi - 2\alpha)/(2\pi - 2\alpha)$); b) blunt V-notch under mixed mode loading ($r = \rho \cdot (\pi - 2\alpha)/(2\pi - 2\alpha)$); c) U-notch under mode I loading ($r = \rho/2$); d) U-notch under mixed mode loading ($r = \rho/2$)

2.1. *Fatigue assessment of steel welded joints*

The high-cycle fatigue failure usually happens in the linear elastic regime and shows a brittle nature. These two conditions allow the use of the SED method, in terms of the cyclic averaged SED $\Delta\overline{W}$ of the pointed weld notch, dealing with welded joints made of steel or aluminium alloy under different loading conditions [20–23]. In this case, the radius of the control volume R_0 can be estimated by means of the expression (5):

$$R_0 = \left(\frac{\sqrt{2e_1}\,\Delta K_{1A}^N}{\Delta\sigma_A} \right)^{\frac{1}{1-\lambda_i}} \tag{5}$$

Where e_1 is the mode 1 function of the notch opening angle in the SED expressions for sharp V-notches, $\Delta\sigma_A$ is the fatigue strength of the butt ground welded joint and ΔK_{1A}^N is the N-SIF-based fatigue strength of welded joints. Dealing with welded joints the value of the control volume radius is 0.28 mm. A large amount of experimental fatigue data is summarised through the Strain Energy Density [8] in a double logarithmic diagram (see fig. 3) providing a robust statistical validation for the application of the method to high-cycle fatigue failure when the welded plate thickness is equal to or greater than 6 mm.

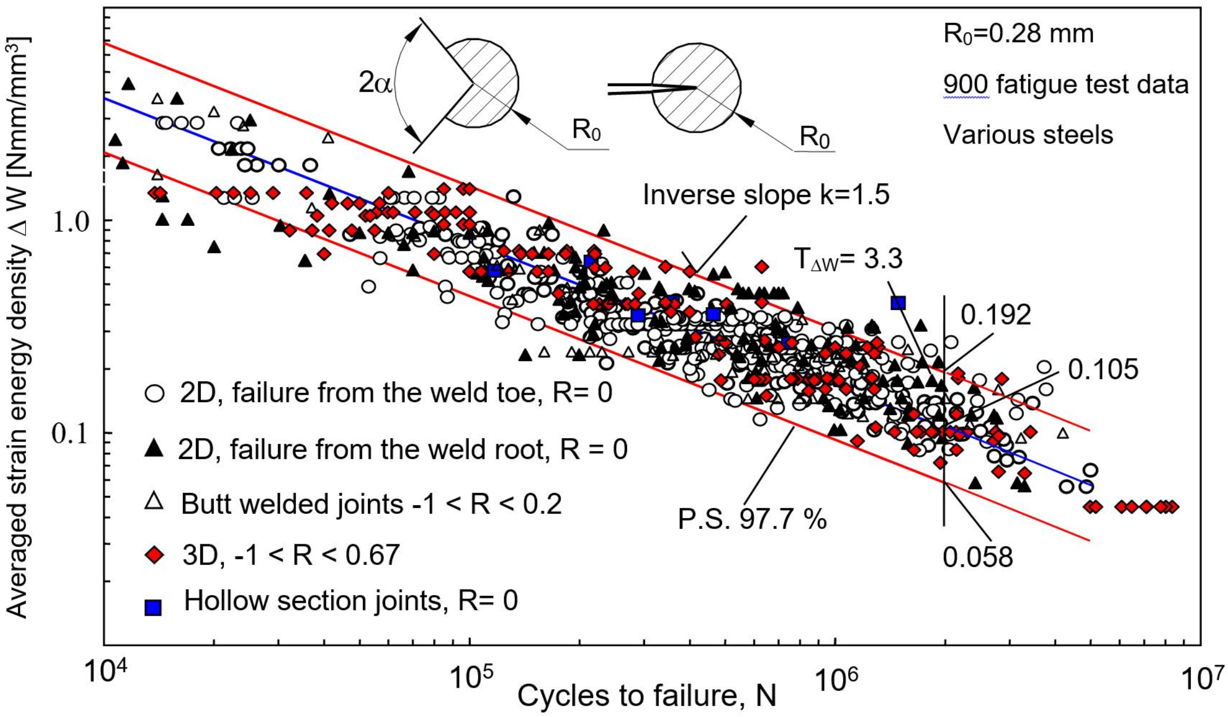

Figure 3:Fatigue strength of steel welded joints as a function of the averaged local strain energy density.

To evaluate the fatigue strength through the SED method, it is enough to calculate the mean-SED at the weld toe or root for a remote tensile load $\Delta\sigma_1$ through a static FE simulation. By means of Eq. (6), valid only under the hypothesis of linear elastic behaviour, it is possible to evaluate the remote tensile load $\Delta\sigma_L$ that represents the fatigue limit of the component:

$$\Delta\sigma_L = \Delta\sigma_i \left(\frac{\Delta W_L}{\Delta W_i} \right)^{\frac{1}{2}} \tag{6}$$

Being ΔW_L the critical value of the averaged SED that corresponds to the fatigue limit, $\Delta\sigma_i$ the remote applied stress in the numerical simulation and ΔW_i the SED value acquired through the numerical

IOP Conf. Series: Journal of Physics: Conf. Series **1296** (2019) 012007 doi:10.1088/1742-6596/1296/1/012007

simulation. As regards steel welded joints ΔW_L is $0.058\ Nmm/mm^3$ with a probability of survival of $P_S = 97{,}7\%$

3. FE analysis

Dealing with turbine runners, the welded joint between blade and crown can be idealised by a simple rounded T-joint subject to bending load condition as shown in fig. 4. In the present work three categories of fatigue specimens, found in literature [23], have been considered: A 2D bending load specimen, called in the present work 2D B. circular, that describes the T-joint between Francis runner blade and ring or crown through a simply rounded joint; a 2D bending load specimen, called in the present work 2D B. optimised, that describes the T-joint between Francis runner blade and ring or crown through an optimized shape for the joint; a 3D specimen for tensile fatigue loading, called in this work 3D T. circular. The specimens are made of typical steel for turbine runner applications.

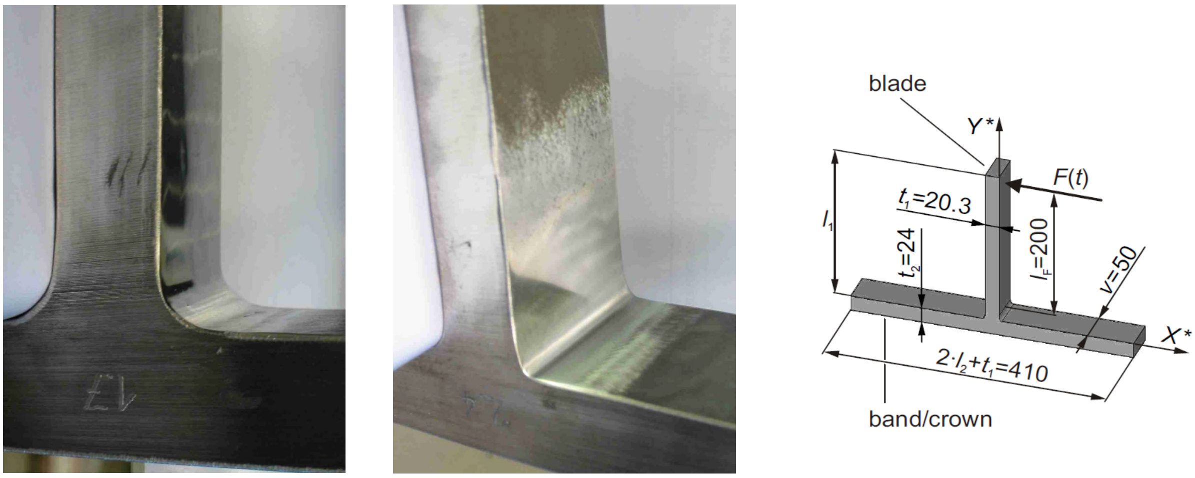

Figure 4: Specimens for bending tests under fatigue loading conditions from the literature [23]

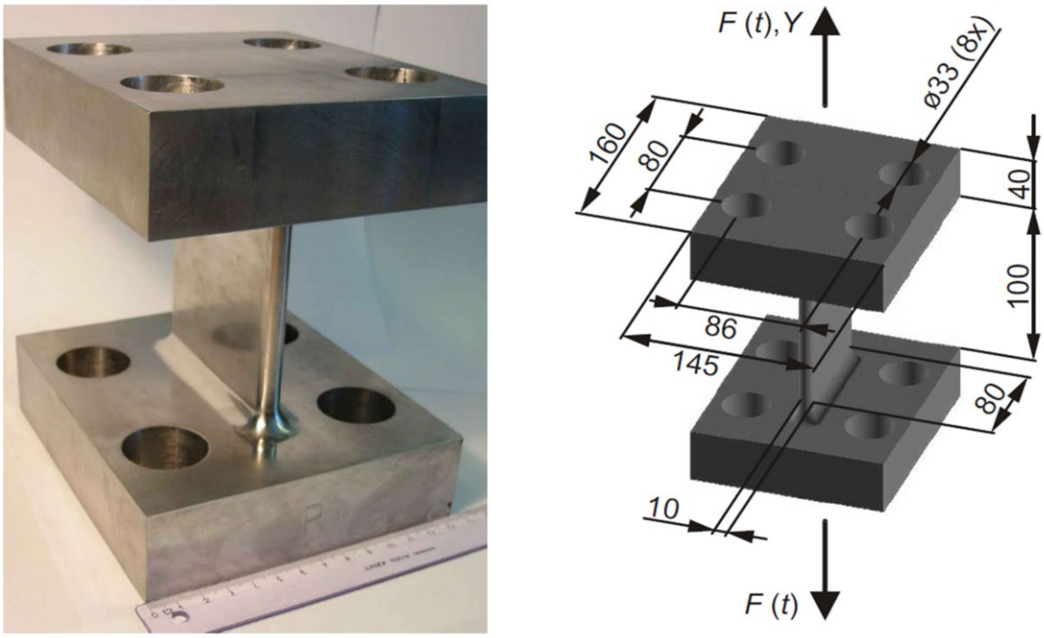

Figure 5: Specimens for tensile tests under fatigue loading conditions from the literature [23]

Table 1 reports the results of the fatigue tests, found in the literature [23], analysed in the present work in terms of averaged SED.

Table 1: Fatigue tests data from literature [23]

Model	Loading	σ_{max} [MPa]	σ_{an} [MPa]	Number of Cycles to failure
2D B. circular	Bending	888	383	80500
		888	383	33600
		892	410	28800
		892	410	52000
		892	410	42000
2D B. optimised	Bending	856	383	148000
		856	383	132000
		856	383	108800
		856	383	110300
		875	410	95000
		875	410	113000
		875	410	115600
3D T. circular	Tensile	689	420	113500
		689	420	135300
		701	461	99100
		701	461	67300
		655	430	43251
		655	430	45800
		608	368	88200
		673	461	12300

IOP Conf. Series: Journal of Physics: Conf. Series **1296** (2019) 012007 doi:10.1088/1742-6596/1296/1/012007

3.1. FE modelling

In order to summarise fatigue tests taken from the literature in terms of averaged SED, the geometry of the specimens was modelled through the software Ansys APDL. To obtain more efficient analyses and minimise the computational time, the symmetries of the details considered were exploited, using the appropriate symmetry conditions in the FE modelling.

Exploiting their geometry and the loading condition, the specimens tested under bending loading were analysed in the present work with a 2D model while, as regards the tensile tests they were analysed through a 3D model. Only for the tensile tests was possible to exploit the symmetry of the geometry modelling only an eighth of the specimen as shown in fig. 7.

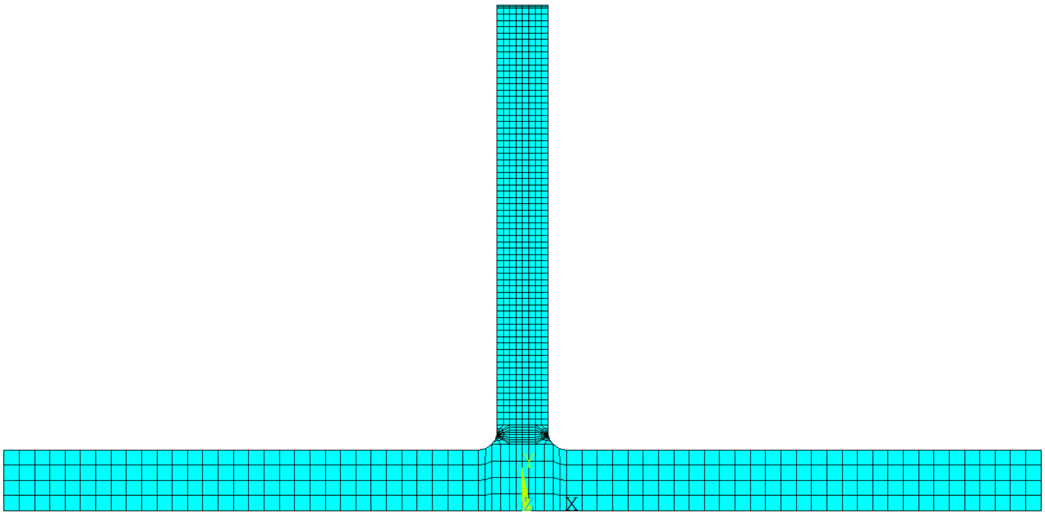

Figure 6:2D FE model for specimens for bending tests

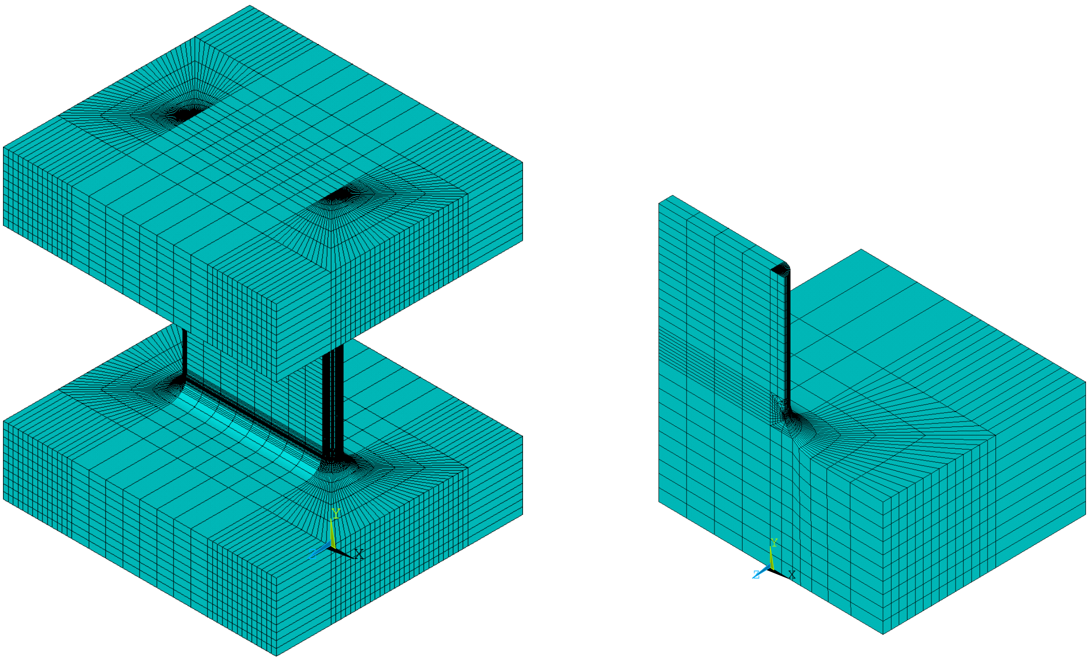

Figure 7:3D FE model for specimens for tensile tests: complete model (left) and model analysed exploiting the symmetry of the component (right)

3.2. Sensitivity analysis

The SED low sensibility to the mesh refinement [24] makes this method feasible to be used for the fatigue assessment of very complex geometries.

The SED low sensibility to the mesh refinement was already hugely proved for 2D models, but few numerical data are available regarding 3D models. In order to understand its sensibility also for this type of model, different analysis was carried out considering as elements both the linear brick (solid185) and the quadratic one (solid186). The results of the analysis, intended as error in percentage (error %) in the evaluation of the SED value and as number of elements (N.E.) for the entire model, are reported in table 2 showing that, using a quadratic element, a good evaluation of the averaged SED value with an error of 0.27% is possible already with a mesh size of $R_0/4$ at the control volume, that correspond for the entire model to a number of 9205 elements. The model with this particular value of the mesh size is shown in fig. 7. From the results, it is also possible to state that the use of quadratic elements (solid186) leads to reach the convergence of the solution with less elements in the model.

Table 2: Sensitivity analysis for the 3D FE model

Element type	Mesh refinement							
	$R_0/2$		$R_0/4$		$R_0/8$		$R_0/16$	
	N.E.	Error %	N.E.	Error %	N.E.	Error %	N.E.	Error %
Solid 185	1130	26.13	9167	3.47	29739	1.07	68656	0.00
Solid 186	1089	5.95	9205	0.27	29708	0.00	68613	0.00

Also for the 2D model different analysis was carried out considering as elements both the linear one (plane182) and the quadratic one (plane183). The results of the analysis are reported in table 3 showing that, using a quadratic element, a good evaluation of the averaged SED, with an error of almost 2%, is already possible with a mesh size of $R_0/4$ at the control volume, that correspond for the entire model to a number of 676 elements.

Table 3: Sensitivity analysis for the 2D FE model

Element type	Mesh refinement							
	$R_0/2$		$R_0/4$		$R_0/8$		$R_0/16$	
	N.E.	Error %	N.E.	Error %	N.E.	Error %	N.E.	Error %
Plane 182	158	-19.94	680	-2.31	1560	0.71	2808	0.00
Plane 183	156	23.89	676	2.04	1554	0.14	2800	0.00

4. Conclusions

For the details analysed, the averaged SED value at the control volume was evaluated using structural analysis and applying to the model the maximum value of the load reached during the fatigue tests considered, $\sigma_{\max}$ in table 1. The results of the analysis are summarised in fig. 8, in a double logarithmic diagram, reporting the value of the averaged SED evaluated through the numerical analysis versus the cycles to failure taken from the experimental tests data.

The diagram reports also the SED-N curve (SED-N curve butt ground welds) obtained considering the fatigue strength of the butt ground weld, available in the literature [4] and equals to 155 MPa at $5 \cdot 10^6$ cycles that results in 210 MPa at $2 \cdot 10^6$ cycles considering an inverse slope for the S-N curve equal to 3. This value could be converted in terms of SED applying the eq. (1) and considering a Young modulus of $200000\ MPa$. Considering this value of SED as the critical value at $2 \cdot 10^6$ cycles and an inverse slope for the SED-N curve equals to 1.5, it is possible to obtain a design curve for the details considered.

The diagram reports also the SED-N curve for a probability of survival of 97,7% (SED-N curve 97,7%), available in the literature [8], obtained from fatigue strength data from welded joints with failure both from weld toe and weld root.
From fig. 8 it is possible to notice that this method is able to summarise in a unique fatigue design curve different type of joints and loading conditions with the advantage of requiring numerical analysis with low computational effort due to the SED low sensibility to mesh refinement. Besides, making a comparison between the fatigue data analysed in the present work and the fatigue curves already available in literature [4,8], a conservative evaluation of the fatigue life for this type of joints is already possible exploiting the SED-N curve for the butt ground welds or the SED-N curve with a probability of survival of 97,7% for welded joints.

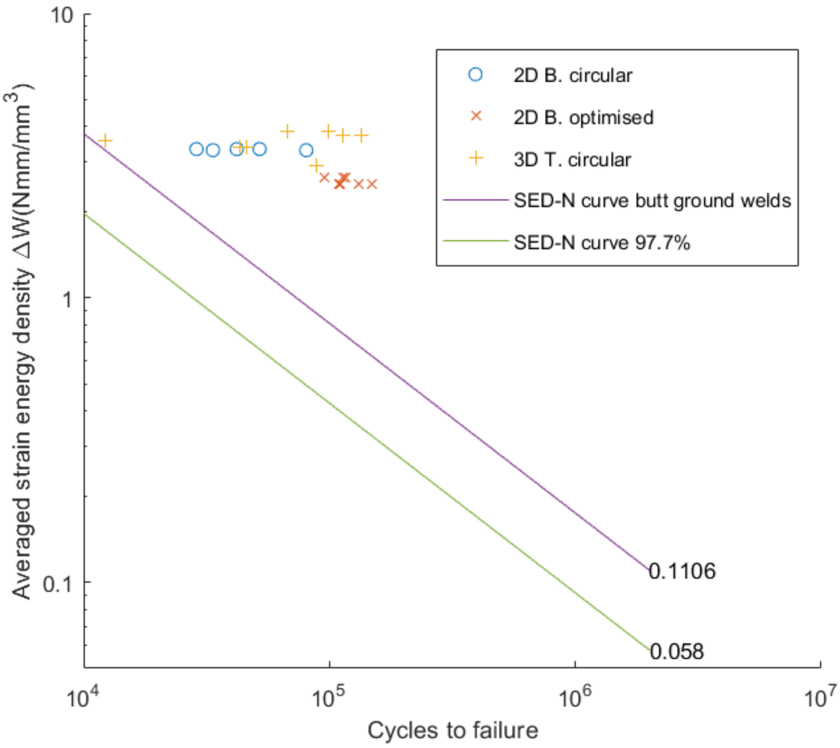

Figure 8: Averaged strain energy density vs cycles to failure for the fatigue tests from literature [23]

5. Acknowledgement

The work presented in this paper has received funding from the European Union’s Horizon 2020 research and innovation programme under grant agreement No 764011.

References

[1] Fricke W. IIW guideline for the assessment of weld root fatigue. Weld World. 2013;57(6):753–91.
[2] 1993-1-3:2009, Eurocode 3: Design of steel structures - Part 1-9: Fatigue (2011)
[3] A. Hobbacher. IIW document IIW-1823-07 FATIGUE DESIGN OF WELDED. 2008;

[4] Radaj D, Sonsino CM, Fricke W. Fatigue Assessment of Welded Joints by Local Approaches: Second Edition. Fatigue Assessment of Welded Joints by Local Approaches: Second Edition. 2006.
[5] Lazzarin P, Livieri P, Berto F, Zappalorto M. Local strain energy density and fatigue strength of welded joints under uniaxial and multiaxial loading. Eng Fract Mech [Internet]. 2008 May 1 [cited 2018 Oct 21];75(7):1875–89.
[6] Lazzarin P, Zambardi R. A finite-volume-energy based approach to predict the static and fatigue behavior of components with sharp V-shaped notches. Int J Fract. 2001;112(3):275–98.
[7] Lazzarin P, Zambardi R. The equivalent strain energy density approach re-formulated and applied to sharp V-shaped notches under localized and generalized plasticity. Fatigue Fract Eng Mater Struct. 2002;25(10):917–28.
[8] Berto F, Lazzarin P. Recent developments in brittle and quasi-brittle failure assessment of engineering materials by means of local approaches. Mater Sci Eng R Reports. 2014;75(1):1–48.
[9] Lazzarin P, Berto F. Some expressions for the strain energy in a finite volume surrounding the root of blunt V-notches. Int J Fract. 2005;135(1–4):161–85.
[10] Lazzarin P, Berto F. From Neuber's elementary volume to Kitagawa and Atzori's diagrams: An interpretation based on local energy. Int J Fract. 2005;135(1–4):33–8.
[11] Yosibash Z, Bussiba A, Gilad I. Failure criteria for brittle elastic materials. Int J Fract. 2004;125(1957):307–33.
[12] Filippi S, Lazzarin P, Tovo R. Developments of some explicit formulas useful to describe elastic stress fields ahead of notches in plates. Int J Solids Struct. 2002;39(17):4543–65.
[13] Gómez FJ, Elices M, Berto F, Lazzarin P. Fracture of U-notched specimens under mixed mode: Experimental results and numerical predictions. Eng Fract Mech. 2009;76(2):236–49.
[14] Berto F, Lazzarin P, Marangon C. Brittle fracture of U-notched graphite plates under mixed mode loading. Mater Des. 2012;41(2012):421–32.
[15] Lazzarin P, Berto F, Ayatollahi MR. Brittle failure of inclined key-hole notches in isostatic graphite under in-plane mixed mode loading. Fatigue Fract Eng Mater Struct. 2013;36(9):942–55.
[16] Gómez FJ, Elices M, Berto F, Lazzarin P. Local strain energy to assess the static failure of U-notches in plates under mixed mode loading. Int J Fract. 2007;145(1):29–45.
[17] Berto F, Lazzarin P, Gómez FJ, Elices M. Fracture assessment of U-notches under mixed mode loading: Two procedures based on the "equivalent local mode I" concept. Int J Fract. 2007;148(4):415–33.
[18] Gómez FJ, Elices M, Berto F, Lazzarin P. A generalised notch stress intensity factor for U-notched components loaded under mixed mode. Eng Fract Mech. 2008;75(16):4819–33.
[19] Lazzarin P, Lassen T, Livieri P. A notch stress intensity approach applied to fatigue life predictions of welded joints with different local toe geometry. Fatigue Fract Eng Mater Struct. 2003;26(1):49–58.
[20] Livieri P, Lazzarin P. Fatigue strength of steel and aluminium welded joints based on generalised stress intensity factors and local strain energy values. Int J Fract. 2005;133(3):247–76.
[21] Atzori B, Berto F, Lazzarin P, Quaresimin M. Multi-axial fatigue behaviour of a severely notched carbon steel. Int J Fatigue. 2006;28(5–6):485–93.
[22] Berto F, Lazzarin P. Fatigue strength of structural components under multi-axial loading in terms of local energy density averaged on a control volume. Int J Fatigue. 2011;33(8):1055–65.
[23] HUTH H-J. Fatigue design of hydraulic turbine runners. 2005. 178 p.
[24] Berto F. A criterion based on the local strain energy density for the fracture assessment of cracked and V-notched components made of incompressible hyperelastic materials. Theor Appl Fract Mech. 2015;76:17–26.