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FULL-MODEL MULTIAXIAL FATIGUE LIFE CALCULATIONS WITH DIFFERENT CRITERIA

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Abstract

Fatigue design of structural components under multiaxial loading often relies on fatigue life calculations based on a post-processing of the full-field elastic stress/strain maps obtained from finite element (FE) analysis. In the present article, a post-processing tool is adopted to compute the fatigue life of a structural component, and multiaxial fatigue assessment is carried out by considering different criteria such as that of Smith-Watson-Topper and of Fatemi-Socie. The present paper focuses on a specific structural component related to a quarter-turn heavy-duty valve actuator, called scotch yoke, commonly used in many application sectors such as oil & gas, power and chemical industries. The fatigue assessment of the component is carried out by employing a full-model FE analysis, considering fillet-welded joints exposed to in-phase constant amplitude cyclic bending-torsion fatigue load with load ratio R=-1, with applied maximum load according to EN 15714-3, which is the standard in the valve actuator sector. The elastic stress/strain field extracted from the FE model is used to perform the fatigue assessment of the fillet-welded joints, where the potential fatigue crack initiation points (weld toe and weld root) are described by adopting an effective radius at the weld notches.

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Keywords: multiaxial fatigue, Smith-Watson-Topper, Brown-Miller, Fatemi-Socie, critical plane, valve actuator.

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1. Introduction

Pneumatic or hydraulic scotch yoke valve actuators are commonly used in many industrial sectors and have many different applications. Among the most challenging scenarios for heavy-duty actuation are high-cycle applications, which impose high frequency of operation often at relatively high operating speed.

In order to improve the performance of FLOWSERVE – Limitorque LPS/LHS heavy-duty actuator series, a fatigue assessment is carried out using numerical analysis supported by experimental evidence, to evaluate the high cycle application behavior of one of the main component of the actuator center body, the scotch yoke.

The LPS/LHS actuator series, developed by FLOWSERVE – Limitorque in Italy is currently in accordance with the European norm [1], which imposes a minimum number of cycles for a specific loading condition. However, the number of cycles required by high-cycles applications can be very different, sometimes by orders of magnitude. For example, for an actuator like Limitorque model LPS-15, with a design output torque of 6000Nm, the minimum number of cycles required by the norm is 10^5 with a stroking time of 30s, while in high-cycles applications, a minimum of 2×10^6 cycles are frequently requested.

The use of standard actuator series inevitably involves some uncertainties and oversizing, decreasing the competitiveness on the global market and increasing the cost of the machine. The dimensioning approach presented in this paper is to focus on a validated design, optimizing the construction features of the actuator in order to guarantee the required number of cycles during its service life.

The present paper explores the fatigue assessment of the welded scotch yoke employing a full-model finite element (FE) analysis, considering fillet-welded joints exposed to in-phase constant amplitude cyclic bending-torsion fatigue load with load ratio R = -1. A post-processing tool, named LIFING (www.lifing-fdt.com), is adopted to compute the fatigue life of the component under study on the basis of some multiaxial fatigue criteria available in the literature. The theoretical results are then critically compared and validated against the experimental ones determined through tests on full-scale components [2].

2. Description of the quarter-turn heavy-duty actuator

A quarter-turn valve actuator is a machine aiming at actuating an underlying valve; it has a duty similar to that of a rotary motor, typically mounted on ball or butterfly valves. The LPS/LHS actuator series, developed by FLOWSERVE – Limitorque, are mainly applied in the Oil&Gas field and are able to provide a very high output torque.

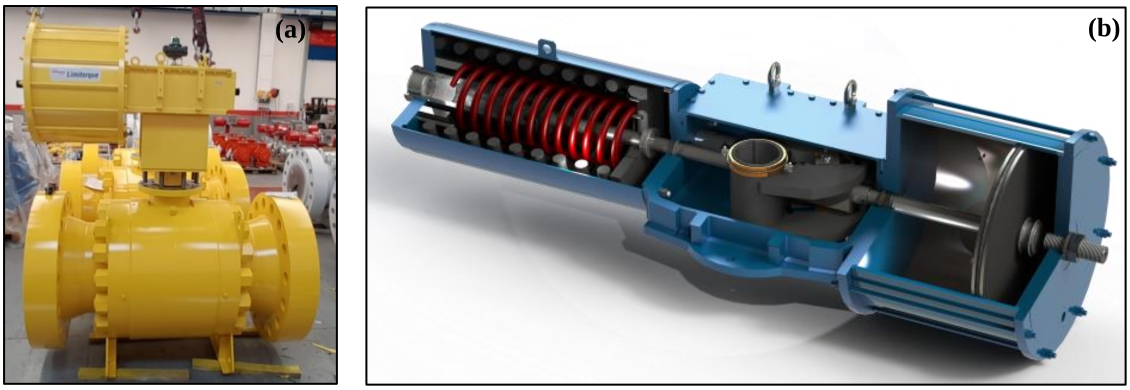

Figure 1: (a) LPS-30 actuator mounted on TMBCV Valbart valve, (b) Single acting fail close heavy-duty pneumatic actuator

Typically, a heavy-duty actuator is composed by two or three main components, depending on the type: in case of a double acting actuator, the machine is composed by a center–body assembly and an actuation cylinder that can be pneumatic or hydraulic, depending on the supply fluid. In case of a single acting actuator, one stroke is performed by spring action, where the spring is placed on the opposite side of the actuation cylinder.

In Figure 2(a), a sample curve of the torque against scotch yoke position produced by a single acting actuator is reported and compared with that required for the valve movement. With the continuum tract are highlighted the torque comparison between the valve and actuator torque during the opening movement, from 0° to 90° while, with the hatched lines are highlighted the torque comparison for the valve closing movement. It is evident from the geometric condition, how the most critical operation positions for the component occurs when the angular stroke is at 0° and 90° corresponding to angle of 45° with the vertical line.

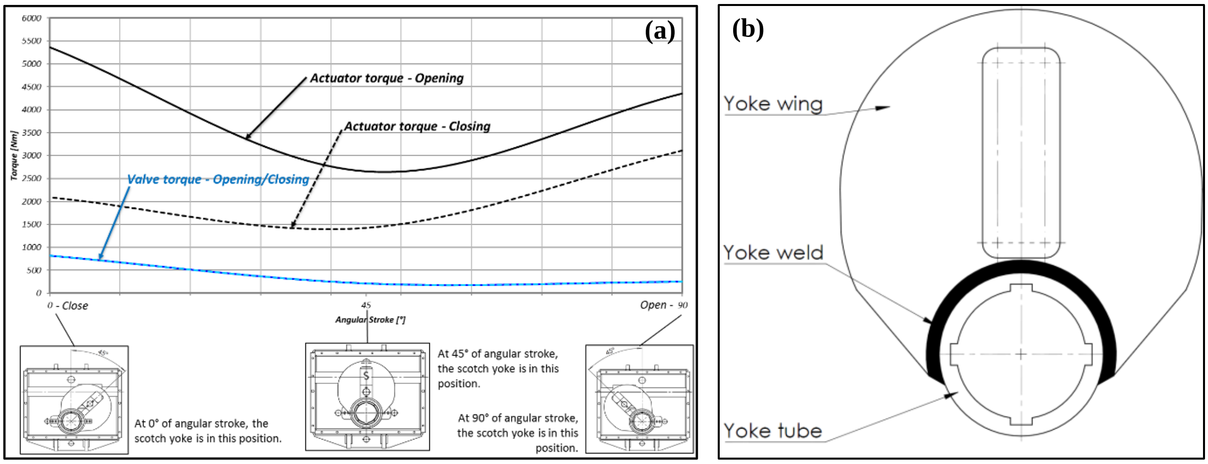

Figure 2: (a) torque comparison example graph between the valve required by the torque and that provided by the actuator, (b) scotch yoke drawing

The scotch yoke is located in the center-body assembly and it is the most important component of the machine because it has the duty to transmit the actuation load from the cylinder to the valve.

A typical scotch yoke for heavy-duty actuator, as those mounted in the LPS/LHS actuator series, is reported in Figure 2(b). It is made of structural steel and composed by a tube with two or four keyways for the connection with the valve joint and two round-shaped plates, called yoke wings, directly welded to the tube by means of two MIG/MAG welds. The two wings are machined in order to carry out a buttonhole for the upper and lower slider blocks. The two slider blocks, one for each wing, transmits the actuation load from the cylinder to the yoke, allowing the valve movement. The cyclic load applied to the component produces an in-phase stress field characterized by bending-torsion components along the welded joints. From a visual inspection of the yoke geometry, it is possible to identify three critical zones related to fatigue strength:

- Fillet-weld joints, on the toes and on the root;
- Edges of the four keyways;
- Final segment of the buttonhole, termed call slider surface

3. Fundamental of the adopted criteria for fatigue life assessment

The fatigue analysis is carried out with the support of a post-processing tool called LIFING. The software is able to calculate the fatigue life and the crack growth in metallic structures on the basis of the results of a FE analysis [3]. More in detail, the life module calculates the fatigue life of the component under investigation according to some well-established methods available in the technical literature, including also the critical plane-based multiaxial criterion co-authored by the last author of the present paper (e.g. see [4] for a review of the criterion). For many structural components, as for example the present scotch yoke, some plastic deformations tend to occur near notches, so that engineering methods for estimating local plastic strains at the notch are needed. In particular, based on the proportional nature of the multiaxial loading being considered, the procedure used for estimating notch plastic strains is based on the Hoffman-Seeger method, which is an extension of the Neuber uniaxial rule.

The Neuber rule, written in terms of equivalent quantities (Von Mises stress and strain) for multiaxial loading, is reported below:

$$\sigma_e^{eq}\varepsilon_e^{eq} = \sigma^{eq}\varepsilon^{eq} \tag{1}$$

where

$$\sigma_{eq} = \frac{\sigma_1}{|\sigma_1|}\sqrt{\sigma_1^{\ 2} + \sigma_2^{\ 2} - \sigma_1 \cdot \sigma_2} \tag{2}$$

being σ_1 and σ_2 the maximum and minimum principal stresses, respectively.

The stress-strain relationship in the elastic-plastic regime can be described by the cyclic Ramberg-Osgood equation applied to the equivalent Von Mises stress and strain:

$$\varepsilon^{eq} = \frac{\sigma^{eq}}{E} + \left(\frac{\sigma^{eq}}{K'}\right)^{\frac{1}{n'}} \tag{3}$$

Equations (1) and (2) are recasted in the typical Neuber format:

$$\sigma_e^{\ eq}\varepsilon_e^{\ eq} = \frac{\left(\sigma_e^{\ eq}\right)^2}{E} = \frac{\sigma_{eq}^{\ \ 2}}{E} + \sigma_{eq}\left(\frac{\sigma_{eq}}{K'}\right)^{\frac{1}{n'}} \tag{4}$$

and in its cyclic format

$$\frac{\left(\Delta\sigma_e^{\ eq}\right)^2}{E} = \frac{\Delta\sigma_{eq}^{\ \ 2}}{E} + 2\Delta\sigma_{eq}\left(\frac{\Delta\sigma_{eq}}{2K'}\right)^{\frac{1}{n'}} \tag{5}$$

By numerically solving equations (4) and (5), the local elastic-plastic stress and strain components can be obtained [5].

In order to compute the fatigue life under multiaxial proportional loading, different criteria, e.g. based on the critical plane approach, can be used. These criteria evolved from experimental tests on crack nucleation and growth in structural components subjected to general loading conditions, where, according to experimental observations, cracks tend to nucleate and grow on specific planes called critical planes. A critical plane criterion is able to incorporate the dominant parameters governing the type of crack growth, which, depending mainly on material, stress-strain state, is usually occurring along either shear or tensile planes. Critical plane criteria are able to predict both the fatigue life of the component and the critical plane orientation, described by the angle θ. Three different critical plane criteria are used for the fatigue assessment of the scotch yoke.

3.1. Brown-Miller criterion

Brown and Miller proposed that both cyclic shear and normal strain on the plane of maximum shear must be considered. The assumption is that cyclic shear strain contributes to nucleate the cracks while the normal strain assists in their growth. It is possible define a Brown-Miller parameter as [5]:

$$\Delta\gamma = \frac{\Delta\gamma_{\max}}{2} + S\Delta\varepsilon_N \tag{6}$$

where the first term is the equivalent shear strain amplitude, while S is a parameter which depends not only on the material but also on the fatigue life. In line with a Coffin-Manson approach, where elastic and plastic strains are considered separately, it is possible to write the strain-life equation reported below [5]:

$$\frac{\Delta\gamma_{\max}}{2} + S\Delta\varepsilon_N = A\frac{\sigma'_f}{E}\left(2N_f\right)^2 + B\varepsilon'_f\left(2N_f\right)^c \tag{7}$$

3.2. Fatemi-Socie criterion

Fatemi-Socie criterion, based on that of Brown-Miller, suggests that the normal strain term should be replaced with the normal stress. Moreover, the main difference with Brown-Miller criterion is that the fatigue parameter compounds shear and axial components in a multiplicative form, implying that whatever the axial component is, if no shear occurs, the crack does not nucleate.

The strain life equation is reported below [5]:

$$\frac{\Delta\gamma}{2}\left(1+k\frac{\sigma_{n,\max}}{\sigma_y}\right)=\frac{\tau'_f}{G}\left(2N_f\right)^{b\gamma}+\gamma'_f\left(2N_f\right)^{c\gamma} \tag{8}$$

3.3. Smith-Watson-Topper criterion

The two critical plane criteria illustrated above have been developed for materials for which the dominant failure mechanism is shear crack nucleation and growth. The damage model proposed by Smith, Watson and Topper (SWT) regards materials where the failure is mainly related to crack growth on planes of maximum tensile strain or stress. The crack nucleation remains related to shear but the fatigue life of the component is controlled by crack growth on planes perpendicular to the maximum principal stress or strain. SWT model can be used in both proportional or non-proportional loading conditions on a material that fails primarily due to Mode I cracks. The SWT parameter can be determined using equation (9) reported below [5]:

$$\sigma_{n,\max}\frac{\Delta\varepsilon_1}{2}=\frac{{\sigma'_f}^2}{E}\left(2N_f\right)^{2b}+\sigma'_f\,\varepsilon'_f\left(2N_f\right)^{b+c} \tag{9}$$

4. Numerical analysis

Figure 3(a) shows a sample of the welded scotch yoke prototype under investigation, without any surface treatment but with a stress relieving heat treatment. In order to evaluate the fatigue behavior of the component and also to device a predictive fatigue strength tool for different sizes of the component, a series of numerical analysis is carried out.

The two loading cases corresponding to opening and closing position of the actuator, where the stroke is positioned, respectively, in counterclockwise and clockwise direction, are considered in a linear elastic static analysis.

After de-featuring operations on the 3D parametric model, the geometry is imported in ANSYS Workbench 17.1, specifying the material properties reported in Table 1. Exploiting the symmetric geometry along YZ plane of the component, as clearly shown in Figure 4(b), only its upper part is analyzed.

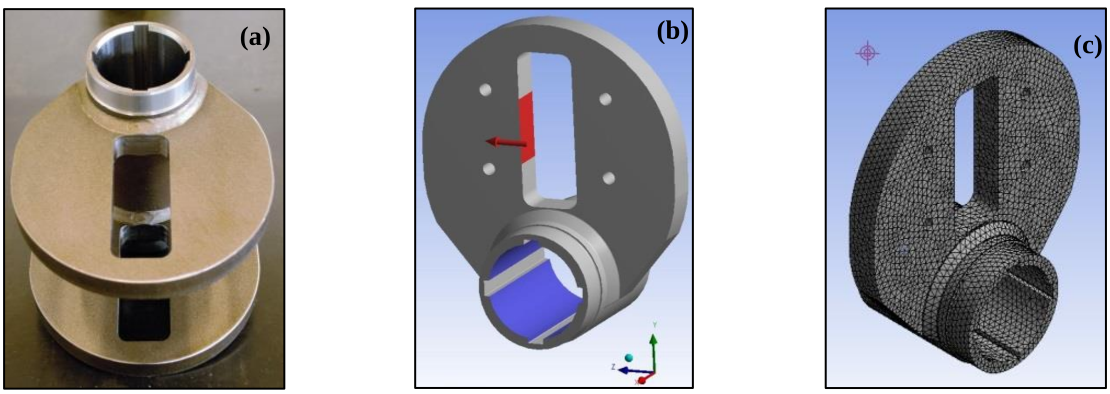

Figure 3: (a) example of scotch yoke, (b) main model, (c) FE mesh

A fictitious round notch with a reference radius of r_{ref} = 1mm [6] is used at the toes and root of the non-penetrant fillet weld. The component is loaded along the slider block position with a $\Delta F = 130\ kN$. Two constrains are applied to the model: the first one in the inner surface of the yoke tube along radial direction in order to simulate the presence of the joint, the second one in the faces of keyways in contact with keys as shown in Figure 3 (b). Two main-models, meshed with 66432 tetrahedral 10-node elements, termed SOLID187 in ANSYS, are analyzed, one for each loading conditions, as shown in Figure 3 (c).

From the observation of FE results, the hot spots are identified and one sub-model with the two loading conditions imported, shown in Figure 4(a), is generated in order to obtain a detailed description of the stress/strain field in the most critical regions of the component. Such critical regions, identified in the two main-models previous analyzed, are meshed with an average element dimension, $d = 0.20$ mm (see Figure 4(b)), which fulfills the recommended dimension reported in [6].

Boundary conditions are applied along with the loading conditions on the meshed sub-model geometries, so that the output torque generated by the scotch yoke is equal to the maximum structural torque (MST). Note that, according to the reference standard [1], the actuator must be able to guarantee a minimum number of cycles at 60% of MST, so that a more demanding loading scenario, than that strictly prescribed by the standard, is considered in the present analysis.

The FE analysis being performed identifies the following most critical zones: weld notch, weld roots, edge of the keyways, radius of the final segment of the slider surface.

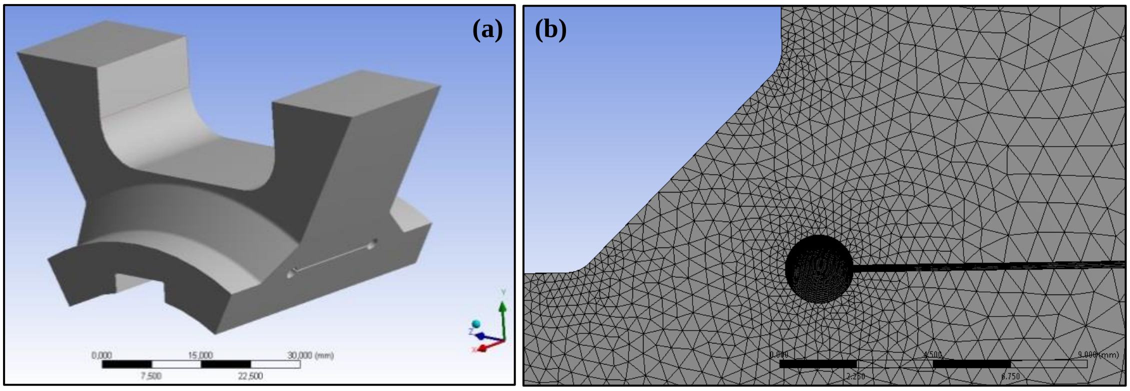

Figure 4: (a) sub-model, (b) mesh detail at weld root and weld toes

The results of the two sub-model analyses are exported from ANSYS and, after some manual operations on the files, the stress tensors related to the surface points are imported in LIFING together with the mesh data. In Figure 5 (a) and (b) the two maps of the equivalent Von-Mises stress for both loading conditions are showed.

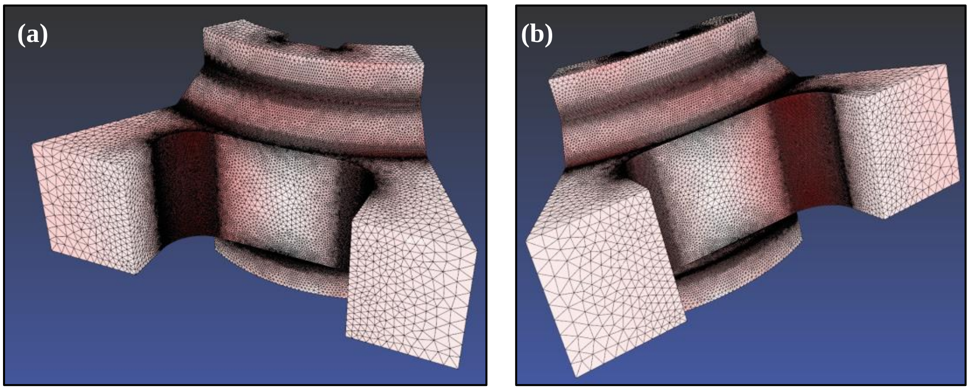

Figure 5: (a) qualitative Von Mises equivalent stress map, load case 1, (b) qualitative Von Mises equivalent stress map, load case 2. For both the pictures, the red colour highlights the highest stress

The materials parameters required in the fatigue with LIFING are taken from the technical literature [7] and are reported in Table 1.

5. Results and comparison

A discussion of the results obtained with LIFING is presented. At first glance, the numerical results confirm that the critical regions for the fatigue strength of the component are coincident with those identified by an experimental campaign conducted on 12 scotch yoke specimens, in collaboration with the Department of Industrial Engineering of Padova University [8], [2]. The fatigue life maps of the scotch yoke are reported in Figure 6(a), for the SWT criterion, in Figure 6(b) for the FS criterion and in Figure 6(c) for the BM criterion. Each critical region is subsequently analyzed in details, by considering fatigue life estimation along a critical path of length of about 4mm. The fatigue life results of the three criteria being considered are compared and validated against experimental results.

Table 1. Material parameters adopted in FE analysis and fatigue life calculation.

Structural steel mechanical properties		
Elastic Modulus	306	GPa
Poisson Modulus	0.3	/
σ_y	346.00	MPa
K'	595.85	MPa
n'	0.0757	-
σ'_f	952.20	MPa
b	-0.089	-
ε'_f	0.7371	-
c	-0.664	-
S – Brown-Miller model	0.40	-
S – Fatemi-Socie model	0.36	-

The fatigue life calculation shows that the critical points of the yoke component are in the fillet-weld between the tube and the plate, and that the weld root is more critical than the lower toes. As far as keyway edges are concerned, it should be noted that fatigue life is calculated by considering a uniform load distribution between the four keys of the joint, which is a difficult condition to be obtain in the experiments; therefore, the FE comparison for these regions with experimental results should be taken with caution.

Figure 7(a) show the results of fatigue life calculation for the weld root, which is the most critical zone of the component. The results obtained with FS and BM criteria are very similar, as shown in Figure 7(b), where the fatigue life estimation is reported along the selected path. On the other hand, fatigue life estimation with SWT criterion slightly diverges from the previously mentioned ones, although it provides conservative results. A comparison with the experimental tests can be made but with some preliminary observations, that is, the numerical analysis identifies the crack nucleation in the component, while the experimental tests are carried out up to a certain crack growth in the specimen. In particular, for the loading condition being considered, the experimental test showed a fatigue life of 6.61×10^5 cycles, with a crack already propagating in the component [2]. By comparing the experimental results with those obtained from FE analysis it is evident that, considering the crack growth phase, the SWT criterion is able to better describe the real behavior of the yoke. The three multiaxial criteria, employed for fatigue life evaluation, identify two different critical elements in the weld root, as reported in Table 2. The two critical elements, located in the left side of the lower root, are close to each other. Also, the critical planes identified in the two elements are slightly different. In Figure 7(c), the critical plane, highlighted in green and obtained applying the SWT criterion (the angle θ defines the orientation of the critical plane with respect to the normal to the specimen surface [3]), is reported for the critical element.

The same elaboration of the results illustrated above is carried out for the other three critical zones identified by the FE model. The results are reported in Table 2 and Figure 8.

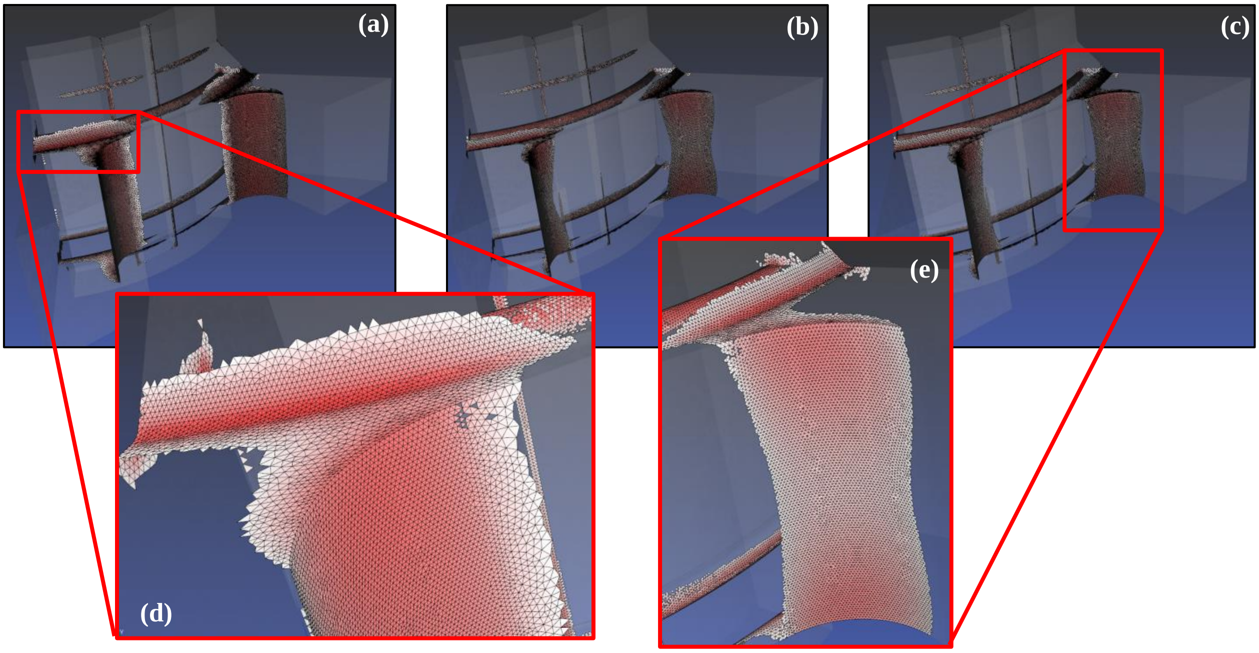

Figure 6: (a) qualitative life map obtained with Smith-Watson-Topper criterion, (b) qualitative life map obtained with Fatemi-Socie criterion, (c) qualitative life map obtained with Brown-Miller criterion, (d) critical zone enlargement from the SWT qualitative life map, (e) critical zone enlargement from the BM qualitative life map. For all the pictures reported, the red colour highlights the shortest estimated life.

All the models used agree in identifying as the most critical region of the components the weld root, followed by the lower weld toe, in proximity of the yoke wing. The critical planes calculated at weld toe are similar for the three criteria being considered, and comparable with the experimental evidence. For the other two critical zones, i.e. the final segment of the slider surface and the edges of the keyways, the results of the three criteria in terms of fatigue life are slightly different. In particular, the SWT criterion estimates a similar fatigue life for both the two zones, while the BM and FS criteria estimate that the slider surfaces have a fatigue life one order of magnitude than that of the edge keyways. For the edge of the keyways a similar orientation of the critical plane is found by the three criteria, while for the slider surfaces, a slight difference of critical plane orientation between SWT criterion and the two others is detected; the criterion is the best comparison with experiments is that of SWT.

Table 2. Fatigue life calculation

Life value		SWT	FS	BM
Roots	Average fatigue life of the most critical elements	2.03×10^5 cycles	9.71×10^5 cycles	1.17×10^6 cycles
	Critical element fatigue life	1.20×10^5 cycles	3.91×10^5 cycles	4.64×10^5 cycles
Toes	Average fatigue life of the most critical elements	3.43×10^5 cycles	3.76×10^6 cycles	3.96×10^6 cycles
	Critical element fatigue life	1.62×10^5 cycles	9.71×10^5 cycles	1.04×10^6 cycles
Slider surface	Average fatigue life of the most critical elements	9.93×10^6 cycles	6.77×10^7 cycles	7.23×10^7 cycles
	Critical element fatigue life	2.75×10^6 cycles	1.11×10^7 cycles	1.35×10^7 cycles
Keyway	Average fatigue life of the most critical elements	1.71×10^7 cycles	1.16×10^9 cycles	7.47×10^8 cycles
	Critical element fatigue life	2.28×10^6 cycles	1.15×10^8 cycles	1.04×10^8 cycles

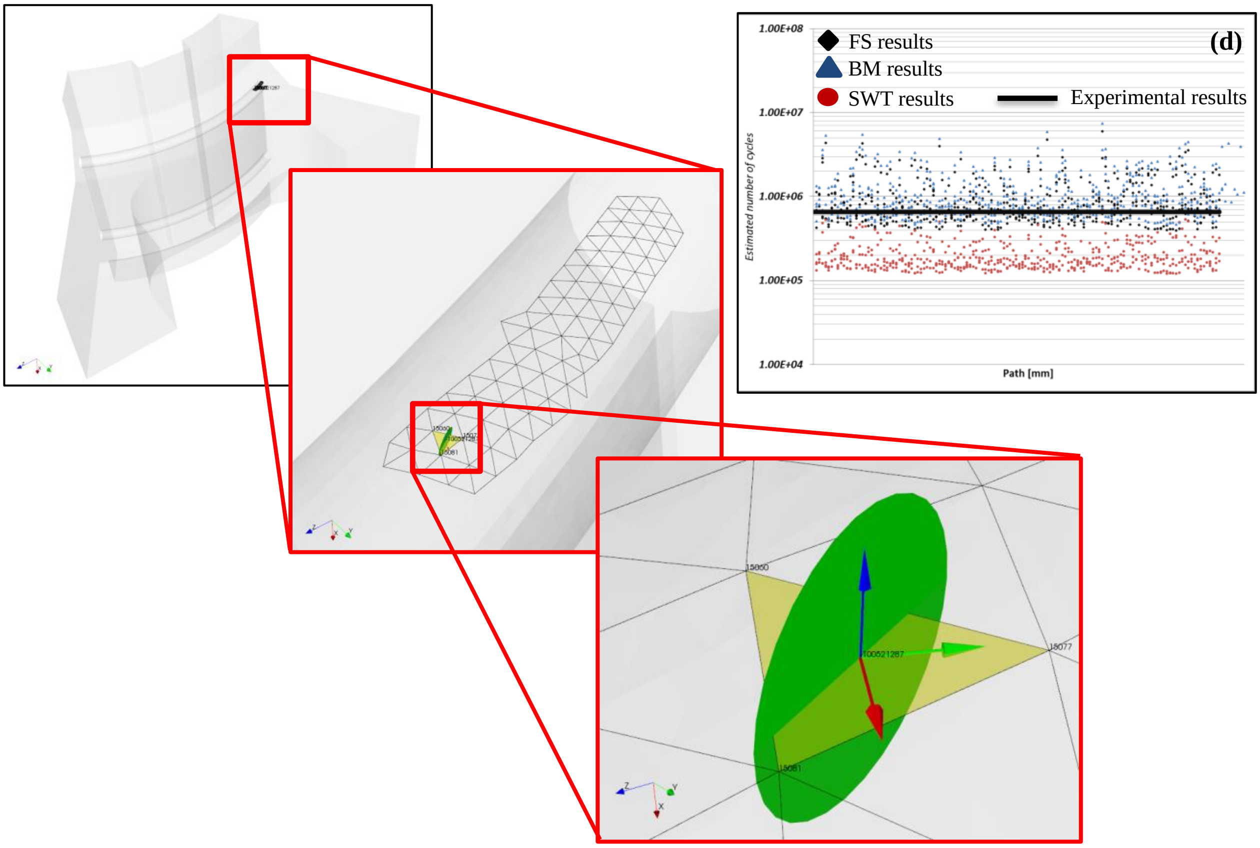

Figure 7: (a) analysed element of the weld root, (b) estimated life for a 4mm path along weld root, (c) critical element of the weld root with the critical plane, (d) fatigue life according with the three criteria exanimated

The experimental tests conducted in the laboratory of Padova University can be used for validating the FE results discussed above [8], [2]. However, a relevant difference in the location of the most critical zone is recorded between experiments and FE analysis, as the experimental most critical zone is shown to be the final segment of the slider surface [8], while the most critical zone in the FE models, for all the three analysis model used, is the weld root. Moreover, the experimental tests showed a complex and not clearly understandable situation with multiple crack initiation sites, where in some cases the crack nucleation site is not clearly discernible. An unambiguous identification of the crack nucleation point is made particularly difficult due to the vicinity, apart from the keyway edges, of the critical points in the component.

Some experimental specimen of the yoke showed a root failure as reported in Figure 8(a), where the detail (b) indicates the presence of multiple crack nucleation sites, along different orientation planes, which tend to coalesce during growth phase. It can be conjectured that the different crack positions highlighted at the end of the experimental tests could have originated in the weld root or toes.

6. Conclusions and research developments

The following conclusions can be drawn:

- the FE analysis carried out is able to identify all the critical areas in terms of fatigue resistance of the component, offering to designer good information about the behavior of the scotch yoke under cyclic loading;

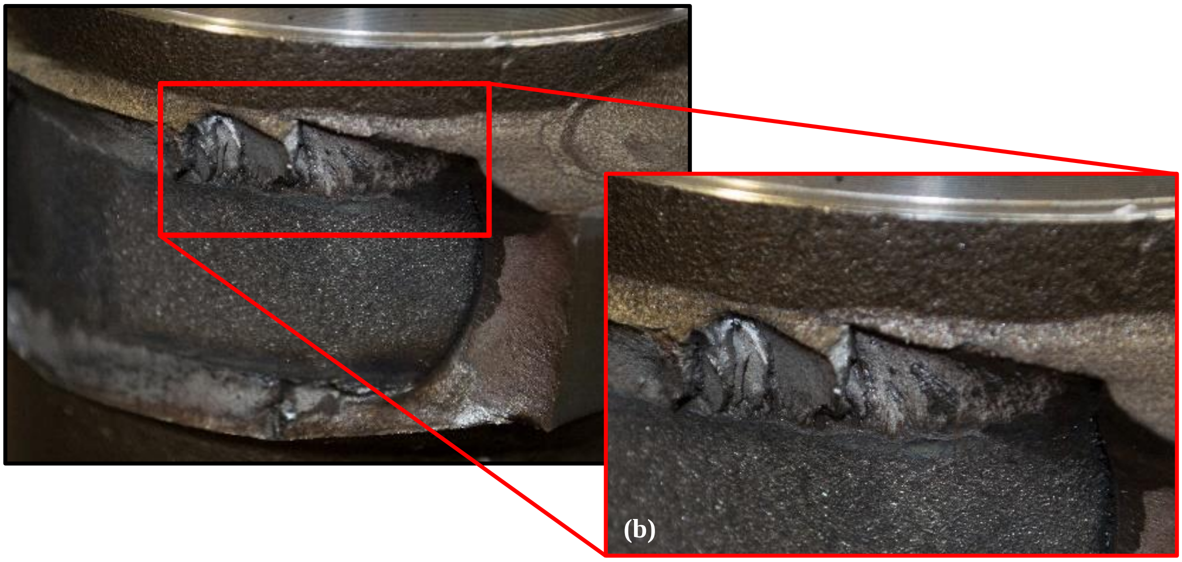

Figure 8: (a) crack of the weld in the experimental test, (b) detail of the crack planes.

- the time required for the FE analysis is compatible with the working time typically required in industrial design environment, ensuring a quick response with an acceptable reliability of the results;
- the component under study has met the fatigue requirement of the relevant norm by a factor of about 1.6;
- from the fatigue life estimation obtained with three popular multiaxial criteria (SWT, FS and BM criteria), the SWT criterion appears to be the most conservative;
- according to FE analysis, the most critical point of the component appears to be the weld root. This finding is partially confirmed by experiments, as in the latter a certain scatter in the initiation site of cracks is recorded (some cracks started from the weld root, others from the weld toes and in the final segment of the slider surface);
- the FE results related to the keyways of the component are not comparable with the experimental tests, possibly because, conversely to FE models, in the tests the load was not uniformly distributed between the four keys.

The future developments of the present work are as follows:

- thorough investigation on the experimental crack nucleating sites in the component in order to carry out a more detailed comparison between experimental and numerical results;
- use of material parameters in the FE model, obtained from experimental tests on the actual material of the scotch yoke;
- experimental testing of the scotch yoke directly in the application condition in order to verify if the adopted boundary and loading conditions are representative of the actual ones.

7. Acknowledgments

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