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Experimental tests and fatigue strength assessment of a scotch yoke valve actuator

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Abstract

Aim of this work is the fatigue assessment of a main component, termed scotch yoke, of a valve actuator used for oil & gas, power and chemical industries, in order to comply with its heavy-duty applications. To do this, full-scale specimens of the scotch yoke made of structural steel have been fatigue tested under nominal axial loading. All specimens have been tested under stress-relieved conditions by adopting a nominal load ratio R=-1. After experimental tests, the fatigue crack paths have been analysed by means of liquid penetrant inspections. The fatigue strength class of the considered scotch yoke has been determined by statistically re-analysing the experimental results, expressed in terms of range of the nominal applied load, and it has been compared with the design condition required by the relevant European Standard, EN 15714-3/4. Finally, two methodologies for fatigue strength assessment of the considered scotch yokes have been proposed, which are based on experimental fatigue data derived from smooth or sharp V-notched specimens, respectively, made of the same yoke material. The assessment capability of the proposed methodologies has been evaluated and discussed by comparing theoretical estimations with the experimental fatigue results of the scotch yokes.

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Keywords: Valve actuator; structural steel; Experimental test; Fatigue assessment; Strain energy density (SED)

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1. Introduction

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

The LPS actuator series, see an example in Fig. 1, are mainly adopted in the Oil&Gas field and are able to provide a very high torque output. They are currently designed by FLOWSERVE – Limitorque according to the relevant European Standard, i.e. EN 15714-3/4 [1], which requires actuators to operate for a minimum number of cycles as a function of the nominal output torque. The required minimum number of cycle could be very different, even by some orders of magnitude. As an example, dealing with the Limitorque actuator model LPS-15, which is designed for a nominal output torque of 6000 Nm, the minimum number of cycles required by the Standard [1] is equal to 10^5. However, in high-cycle applications sometimes required also from chemical sector, customers frequently ask for actuators operating at least for 2 million cycles.

Accordingly, the adoption of standard actuator series for a wide range of applications inevitably leads to an oversize and an increase of the cost of the project, which corresponds to a decrease of the competitiveness on the global market. This issue can be addressed by adopting a validated design and by optimizing the actuator construction features, in order to guarantee the number of cycles required by Standards while avoiding an oversizing of the actuator.

To do this, experimental fatigue tests have been carried out on one of the main components of a valve actuator, that is the scotch yoke shown in Fig. 1. More in detail, the scotch yoke is located in the center-body assembly of LPS heavy-duty actuator series and it is responsible for the transmission of the actuation load from the cylinder to the valve. The experimental activity was aimed at deriving the design curve and at investigating the damage evolution of scotch yokes under fatigue loadings.

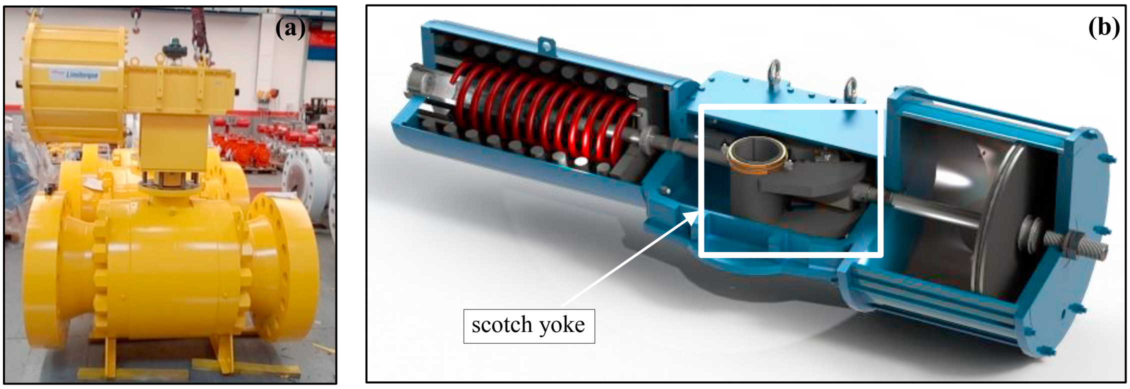

Figure 1: (a) LPS-30 actuator mounted on TMBCV Valbart valve, (b) Single acting fail close heavy-duty pneumatic actuator

Afterwards, methodologies for fatigue strength assessment of the considered scotch yokes have been proposed on the basis of experimental results obtained by fatigue testing simple specimen geometries made of the same material as the yoke. The adoption of either smooth or sharp V-notched specimens in the experimental tests aimed at defining the fatigue design curve has been investigated by comparing theoretical predictions with the experimental fatigue results of the scotch yokes.

The aims of the present contribution are:

- to fatigue test scotch yokes of valve actuators made of steel under nominal axial loading;
- to derive the fatigue strength class of the considered yoke;
- to convert the experimental results from a nominal to a local approach by means of FE analyses;
- to propose and discuss two methodologies for fatigue strength assessment of the scotch yokes based on experimental fatigue data derived from smooth or sharp V-notched specimens, respectively, made of the same yoke material.

2. Experimental fatigue tests on scotch yokes

The scotch yoke has been manufactured in structural steel. The weldments have been performed by using MIG/MAG process. Each yoke wing had a buttonhole being the seat of a slider blocks, which transmits the actuation load (see forces F in Fig. 2) to the yoke. Moreover, the tube had four keyways for the connection with a shaft. The main geometrical parameters of the scotch yoke produced by FLOWSERVE – Limitorque are reported in Fig. 2.

The details of the case cannot be revealed owing to considerations of confidentiality, but the exact nature and dimension of the component is not relevant here.

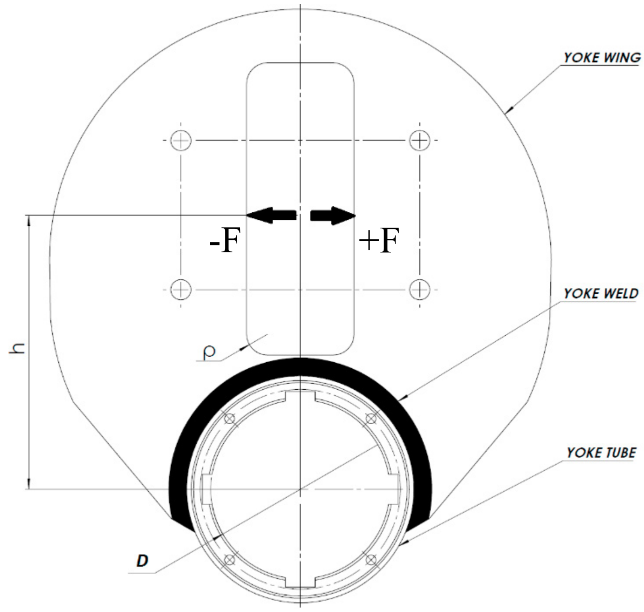

Figure 2: Geometry of the fatigue tested scotch yoke as produced by Flowserve – Limitorque s.r.l.

The experimental fatigue tests have been carried out by means of an axial servo-hydraulic MTS machine, with a load capacity of 250 kN and equipped with a MTS TestStar IIm digital controller (see Fig. 3). To reproduce the in-service loading conditions with the available axial testing machine, two yokes have been tested at the same time by taking advantage of an *ad hoc* experimental arrangement:

- each yoke has been connected to a shaft by means of four keys as in service conditions for high cycles and regulation applications;
- the ends of each shaft have been connected to steel plates by means of eight keys, four for each side (see Fig. 3b), to constrain both rigid displacements and rotations of the tested yokes;
- the axial load applied by the testing machine (see forces F in Fig. 3a) has been transferred to the yokes wings by means of slider blocks at a distance *h* from the tube axis, as reported in Fig. 2. Accordingly, an in-phase bending-torsion fatigue loading condition has been generated in the tube of each scotch yoke, the distance *h* being the torsion arm.

A sinusoidal load cycle, with a nominal load ratio equal to R = -1, was imposed under load control and with a frequency in the range 3÷6 Hz, depending on the applied load level. The tests were continued until failure of the specimen, which was defined by a stiffness drop approximately equal to 10%. All in all, 13 scotch yokes have been fatigue tested.

All tested yokes experienced crack initiation at the fillet-radius of the wing (see Fig. 4, point A), except in two cases where fatigue crack initiated at the tip of the keyway. The fatigue crack paths have been analysed by means of liquid penetrant inspections. Fig. 4a highlights that after an initial crack propagation along the fillet profile (A in

Fig. 4a), the fatigue crack propagated along the weldment (B in Fig. 4a) and in some cases also along the tube (C in Fig. 4a) until it reached the keyway. Eventually, the yoke wing has been separated from the tube on one side (D in Fig. 4b) due to crack propagation all along the weldment, while another propagating fatigue crack has been observed at the opposite side of the same yoke wing (E in Fig. 4b).

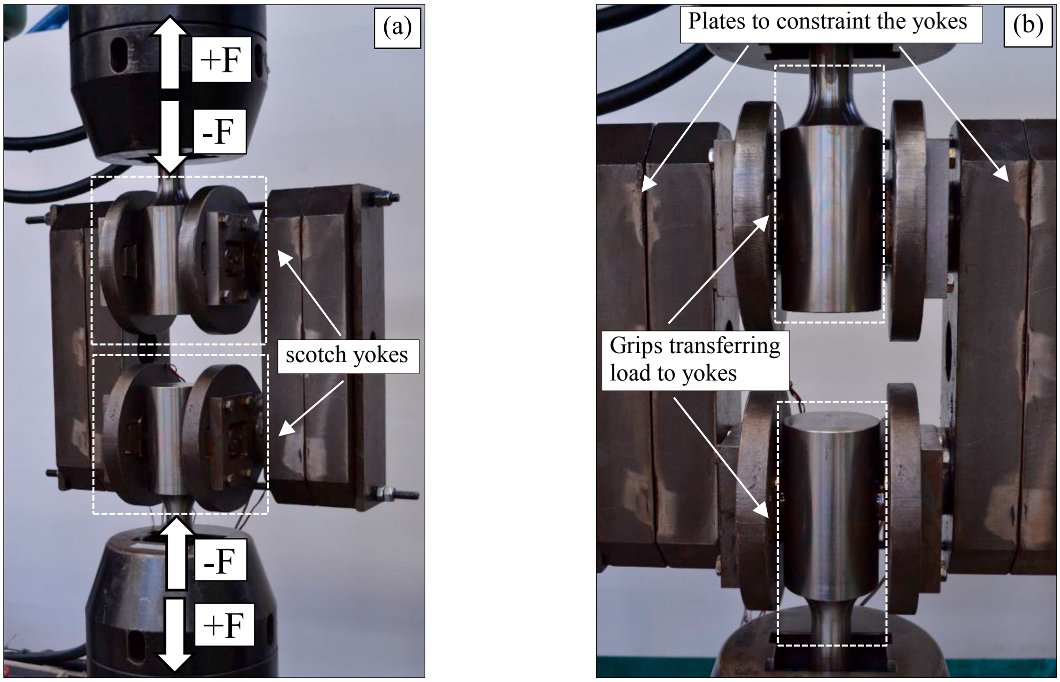

Figure 3: (a) Two scotch yokes tested simultaneously in the axial fatigue test machine. (b) Load and restraints applied to the scotch yokes by means of grips and lateral plates, respectively.

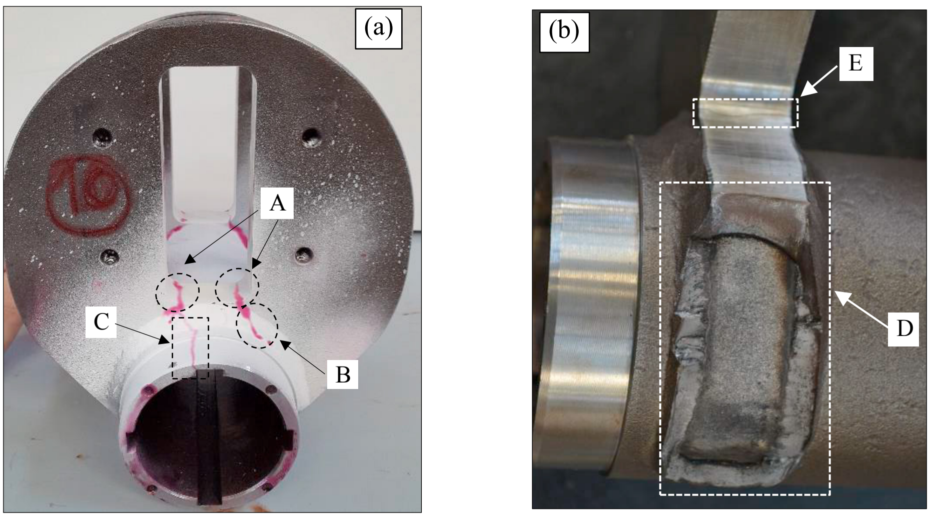

Figure 4: (a) Fatigue crack paths analysed by means of liquid penetrant inspections. (b) Yoke wing separated from the tube on one side due to crack propagation all along the weldment.

3. Fatigue test results

In order to obtain the endurable force ranges of the tested scotch yokes, a statistical re-analysis was carried out on the experimental results expressed in terms of range of the axial load applied by the testing machine, ΔF (i.e. maximum load minus minimum load).

The obtained scatter band, referred to probabilities of survival of 2.3÷97.7%, is reported in Fig. 5, along with the experimental results. The fatigue strength at a probability of survival of 97.7% and with reference to a number of cycles $N_A = 2 \cdot 10^6$ results to be 93 kN, while the inverse slope k is equal to 5.23.

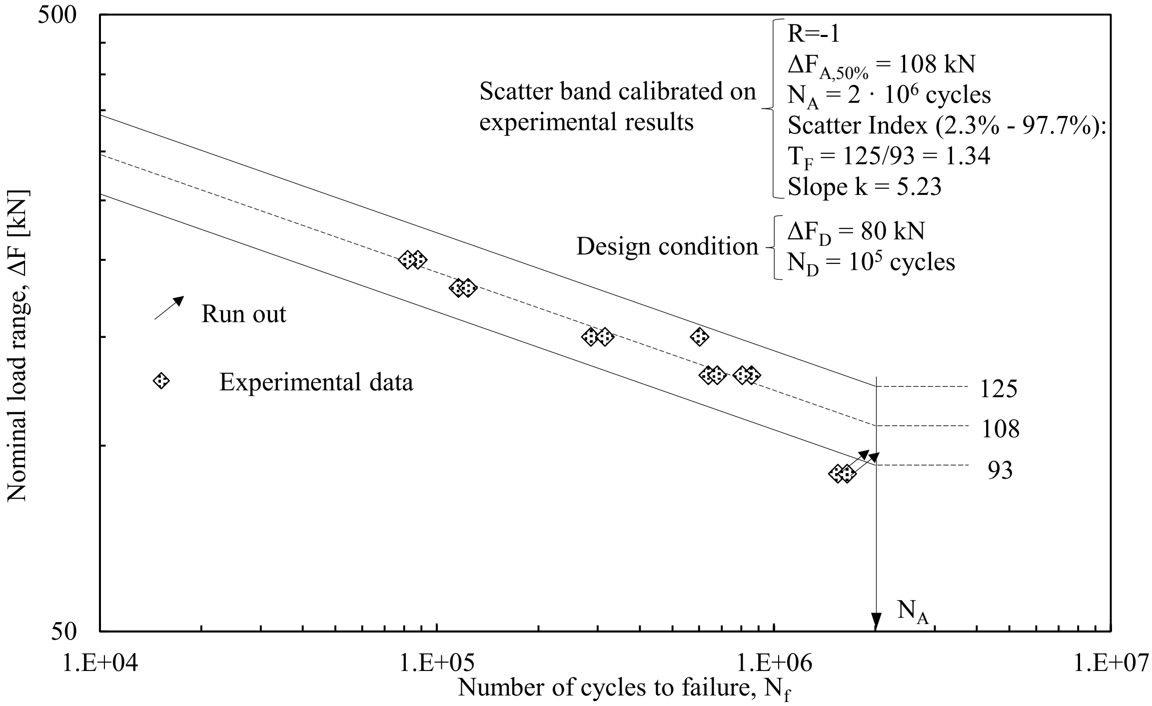

Figure 5: Comparison between the scatter band calibrated on experimental fatigue results and design point of the scotch yoke, as defined in the relevant European Standard [1].

4. FE stress analyses

To convert the experimental fatigue results of the scotch yokes from the nominal axial load to the local peak stress calculated at the point of crack initiation, i.e. the fillet radius, 3D linear elastic FE analyses have been performed by using Ansys FE code. A 3D model including the yoke geometry along with the shaft with four keys (see Fig. 6), has been analysed by employing second-order, ten-node tetrahedral elements (termed SOLID187 of the Ansys element library). First of all, a free mesh has been obtained by setting a global element size d = 3 mm, then, a mesh refinement has been performed in the region close to the fillet radius, i.e. the critical point of the yoke, in order to obtain a minimum FE size $d_{min} \approx 0.8$ mm. The obtained FE mesh is reported in Fig. 6 along with a zoom close to the fillet radius. It should be noted that only half of yoke geometry has been modelled, taking advantage of the symmetry on the YZ plane (see Fig. 6).

To properly simulate the loads applied to the yoke by the axial testing machine (+F and –F in Fig. 3), two different FE models have been solved as shown in Fig. 6: in model (a) the load has been simulated by applying a force (+F/2 due to symmetry) at one side of the yoke wing's buttonhole, while in model (b) the load has been simulated by applying the same force (-F/2) but at the opposite side of the yoke wing's buttonhole as compared to model (a). Indeed, during each fatigue loading cycle the load application point switches from one side to the other side of the buttonhole.

The constraint applied to the shaft by the side plates has been simulated by preventing the shaft end rotation, i.e. by imposing null displacements on the active flank of the key that sticks out of the tube, as shown in Fig. 6. Then, to simulate the connection between yoke and shaft, frictionless contacts have been applied between the flanks of the keys inside the tube and keyways. Both contact and constrained surfaces have been properly selected as a function of the load direction, as reported in Fig. 6 (see in comparison models (a) and (b)).

The FE models, shown in Fig. 6, have been verified by comparing the numerical results with those provided by a strain gauge fixed on the fillet radius of a fatigue tested scotch yoke. As an example, by applying a maximum load -F = 100 kN in both experimental test and FE model, the experimental strain resulted equal to -1500 με, while the numerical strain at the same position resulted equal to -1200 με, so that a fairly good agreement has been obtained.

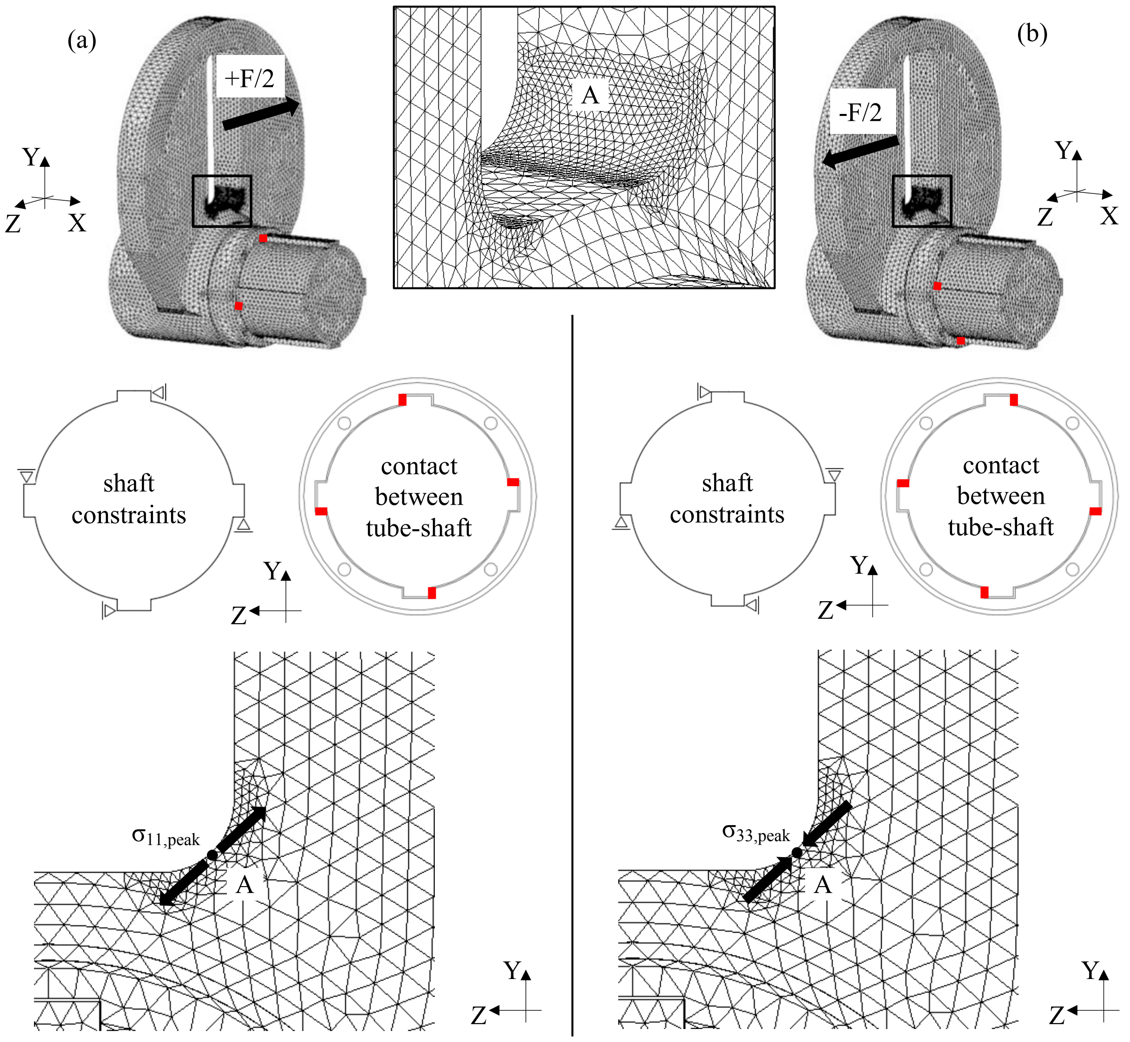

Figure 6: 3D linear elastic FE analyses performed by adopting second-order, 10-node tetrahedral elements (SOLID 187 of the Ansys element library). Models (a) and (b) simulate the loading conditions and the constraints relevant to loads +F and -F, applied during the experimental fatigue tests, as shown in Fig. 3. Point (A) represents the fillet radius, from which peak stresses have been extracted.

Since full notch sensitivity is expected with the adopted fillet radius of the yoke wing (Fig. 2), the maximum peak stress values at the notch tip control the fatigue crack initiation, so that they have been extracted from FE analyses. More in detail, due to different load directions and load application points, the considered fillet (see point A in Fig. 6) is subjected to tensile stress in model (a) and compressive stress in model (b). Accordingly, the maximum principal stress value ($\sigma_{11,peak}$) and the minimum principal stress value ($\sigma_{33,peak}$) have been extracted at the same node from model (a) and (b), respectively. Then, the range of the peak stress value at fillet, normalized by the range of the nominal load, has been evaluated as follows:

$$\frac{\Delta\sigma_{peak}}{\Delta F} = \frac{\sigma_{11,peak} - \sigma_{33,peak}}{\Delta F} = \frac{717\ MPa\ -\ \left(-272\ MPa\right)}{200\ kN} = 4.95\frac{MPa}{kN} \tag{1}$$

The stress values reported in Eq. (1) are referred to a nominal load range ΔF equal to 200 kN, so that a load F/2 = 100kN/2 = 50kN has been applied to the yoke wing in both models (a) and (b). On the basis of the obtained results, it has been possible to estimate also the local load ratio at fillet, which resulted:

$$R_{fillet} = \frac{\sigma_{33,peak}}{\sigma_{11,peak}} = \frac{-272\ MPa}{717\ MPa} \cong -0.4 \tag{2}$$

It should be noted that the fillet is subjected to a load ratio different from the nominal one, i.e. R = -1. This is due to the fact that the load is applied directly above the fillet during a half-cycle, while it switches to the opposite side of the buttonhole during the subsequent half-cycle, so that the resulting cyclic stress cycle at the fillet radius is not symmetrical.

5. Methodologies for fatigue strength assessment of scotch yokes

Methodologies for fatigue strength assessment of the tested scotch yokes can be proposed on the basis of experimental results obtained by fatigue testing simple specimen geometries made of structural steel as the yoke: typically, either smooth or sharp V-notched specimens can be adopted in the experimental tests. Accordingly, two different approaches are presented and discussed in the following by comparing theoretical predictions with the experimental fatigue results of the scotch yokes.

5.1. Fatigue assessment based on results from smooth specimens

The Wohler curve relevant to a given material and load ratio R can be derived on experimental basis, by testing smooth specimens at load ratio R, alternatively it can be estimated by adopting approximate expressions.

The fatigue limit or the high-cycle fatigue strength at N_A = 2 million cycles expressed in terms of stress amplitude, $\sigma_{A,R=-1}$, relevant to smooth specimens made of steel and tested under rotating bending (R = -1) can be estimated as half of the tensile strength, σ_R [2]. Then, the high-cycle fatigue strength $\sigma_{A,R}$ referred to a load ratio R different from -1 can be estimated by using the Goodman-Smith expression, Eq. (3).

$$\sigma_{A,R} = \frac{\sigma_R \cdot \sigma_{A,R=-1}}{\sigma_R + \sigma_{A,R=-1} \cdot \left(\frac{1+R}{1-R}\right)} \cong \frac{\sigma_R \cdot \frac{\sigma_R}{2}}{\sigma_R + \frac{\sigma_R}{2} \cdot \left(\frac{1+R}{1-R}\right)} \tag{3}$$

The low-cycle fatigue strength at 10^3 cycles, $\sigma_{a,MAX,R}$, can be estimated from Eq. (4), which assumes a static failure regime. So doing, the Wohler curve for a given load ratio R can be estimated and adopted for fatigue strength assessment.

$$\sigma_{a,MAX,R} = \sigma_R \cdot \left(\frac{1-R}{2}\right) \tag{4}$$

This approximate procedure has been verified in the present contribution by comparing the Wohler curve estimated by means of Eqs (3) and (4), with experimental fatigue results obtained by testing smooth specimens (see geometry inside Fig. 7) made of the same structural steel, and subjected to axial loading with a nominal load ratio R = 0. A very good agreement between theoretical estimations and experimental results of smooth specimens can be observed from Fig. 7. Therefore, it is believed that the above procedure could be reliable also in estimating the Wohler curve for a load ratio R = -0.4, which is relevant to failures starting from the fillet radius of the scotch yokes under investigation (see Eq. (2)).

Accordingly, Fig. 7 reports the experimental fatigue results of the scotch yokes expressed in terms of range of the peak stress evaluated at the fillet $\Delta\sigma_{peak}$, Eq. (1), along with the estimated Wohler curve referred to R = -0.4 and expressed in terms of stress range. Figure 7 shows a great conservatism between theoretical estimations and experimental results. The reason for this is that all experimental results of the scotch yokes expressed in terms of $\Delta\sigma_{peak}$, evaluated from FE analyses under linear elastic conditions, are well above the yielding stress level (see the red line in Fig. 7), so that more complicated and time-consuming elastic-plastic FE analyses should be carried out to take into account the presence of plastic deformations at fillet radius.

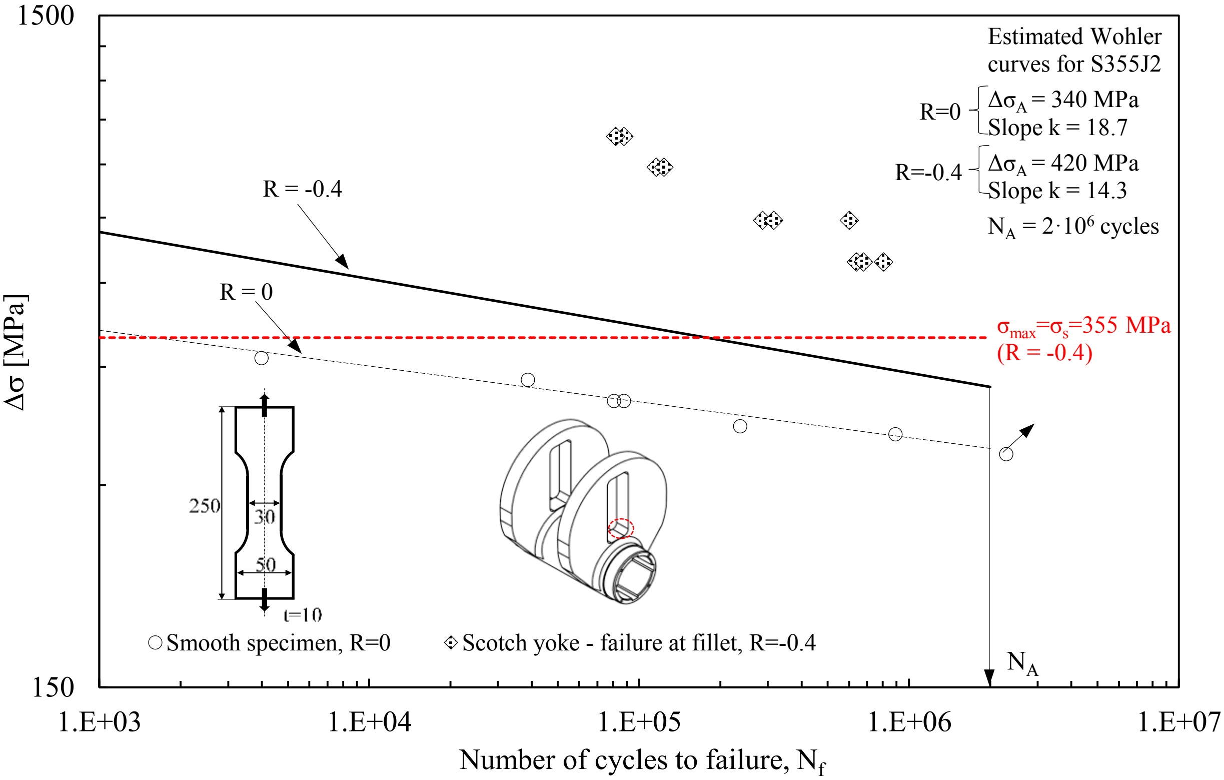

Figure 7: Comparison between experimental results obtained from smooth specimens tested under axial loading with R = 0 and those obtained from scotch yokes with the relevant Wohler curves estimated by using Eqs. (3) and (4), for R = 0 and R = -0.4, respectively.

5.2. Fatigue assessment based on results from sharp V-notched specimens

As alternative option, experimental results obtained from sharp V-notched specimens can be adopted to assess the fatigue strength of the considered scotch yokes. Accordingly, plates made of structural steel and weakened by sharp V-notches with opening angle equal to 90 degrees, see geometry inside Fig. 8, have been fatigue tested under axial loading with nominal load ratios R = 0 and -1.

Then, the strain energy density (SED) averaged over a structural volume having circular shape with radius R_0 and surrounding the fatigue crack initiation point, as proposed by Lazzarin and co-workers [3], has been adopted as fatigue criterion. Dealing with sharp V-notches under mode I fatigue loading, the SED averaged over the control volume can be expressed as follows [3]:

$$\Delta\overline{W} = c_w \frac{e_1}{E}\left[\frac{\Delta K_1}{R_0^{1-\lambda_1}}\right]^2 \tag{5a}$$

Dealing, instead, with blunt V-notches under mode I fatigue loading, in the case of large notch tip radii the SED averaged inside the structural volume is practically coincident with its maximum value [4]. This is the case of the fillet of the yoke wing (Fig. 2), so that the averaged SED can be evaluated as follows:

$$\Delta\overline{W} = c_w \frac{\Delta\sigma_{peak}^2}{2E} \tag{5b}$$

In previous expressions, E is the Young's modulus, e_1 is a known parameter which depends on the notch opening angle 2α and on the Poisson's ratio ν, ΔK_1 is the range of the Notch Stress Intensity Factor (NSIF) relevant to mode I, while $\Delta\sigma_{peak}$ is the range of the peak stress value calculated at the fillet radius, Eq. (1). Finally, the coefficient c_w in previous Eqs (5a) and (5b) depends on the nominal load ratio R [5]: in particular, c_w equals 0.5 for $R = -1$ and 1 for $R = 0$, while it equals 0.6 for $R = -0.40$, i.e. the local load ratio at the fillet radius, Eq. (2).

The control radius R_0 for fatigue strength assessment of notched components made of structural steel has been estimated here by equating the averaged SED in two situations, i.e. the fatigue limit of un-notched and notched specimens, respectively, as proposed by Lazzarin et al. [3]. Therefore, R_0 combines two material properties: the high-cycle fatigue strength of smooth specimens referred to $N_A = 2\cdot10^6$ cycles, $\Delta\sigma_A$, and the value of the Notch Stress Intensity Factor (NSIF) range for sharp V-notches with opening angle equal to 90 degrees referred to the same number of cycles N_A, ΔK_{1A}, according to Eq. (6) below derived in [3].

In the present case, $\Delta\sigma_A$ has been derived by fatigue testing smooth specimens under axial loading with R = 0 (see Fig. 7), while ΔK_{1A} has been calculated by converting the experimental results of V-notched specimens tested again under axial loading with R = 0 in terms of NSIF range from linear elastic FE analyses. Accordingly, the following value of R_0 is obtained:

$$R_0 = \left(\sqrt{2e_1}\cdot\frac{\Delta K_{I,A}}{\Delta\sigma_A}\right)^{\frac{1}{1-\lambda_1}} = \left(\sqrt{2\cdot 0.145}\cdot\frac{252\ MPa\cdot mm^{0.455}}{338\ MPa}\right)^{\frac{1}{1-0.545}} \Longrightarrow 0.135mm \tag{6}$$

Then, taking advantage of the equality $W = (1-\nu^2)\cdot\sigma_{eq,peak}^2/2E$ valid under plane strain conditions, an equivalent peak stress, $\sigma_{eq,peak}$, can be derived according to Eqs (7a) and (7b). This approach has been adopted in the fatigue strength assessment of welded joints according to the Peak Stress Method (PSM) [6].

$$\Delta\sigma_{eq,peak} = \sqrt{c_w}\sqrt{\frac{2e_1}{1-\nu^2}\frac{\Delta K_1}{R_0^{1-\lambda_1}}} \tag{7a}$$

$$\Delta\sigma_{eq,peak} = \sqrt{c_w}\sqrt{\frac{1}{1-\nu^2}}\Delta\sigma_{peak} \tag{7b}$$

The fatigue experimental results relevant to both sharp V-notched specimens and scotch yokes have been reconverted in terms of equivalent peak stress, Eqs (7a) and (7b) respectively, evaluated at the point of crack initiation as observed experimentally: that is the notch tip for V-notched specimens and the fillet radius for scotch yokes. Figure 8 shows the comparison between the experimental data and the scatter band calibrated only on results relevant to sharp V-notched specimens. It can be observed that all test data relevant to scotch yokes fall inside the scatter band, showing a good agreement between theoretical estimations and experimental results.

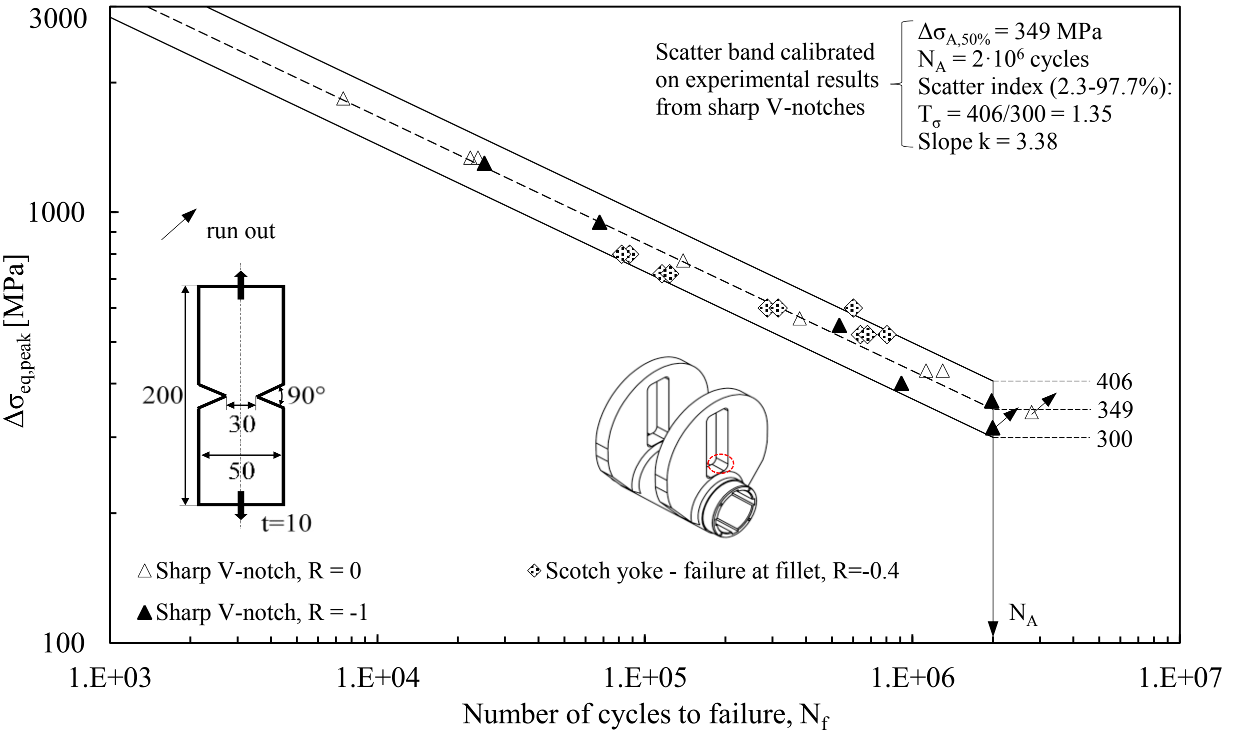

Figure 8: Experimental results obtained from sharp V-notched specimens tested under axial loading with R = 0, -1 and those obtained from scotch yokes, expressed in terms of equivalent peak stress, Eqs (7a) and (7b), respectively. Comparison with the scatter band calibrated only on test data relevant to V-notched specimens.

From the comparison of Figs 7 and 8, it seems that the methodology for fatigue strength assessment of scotch yokes based on experimental results from sharp V-notched specimens and on the application of the averaged SED criterion, provide the best results without the excessive conservatism shown in Fig. 7. Moreover, according to the SED approach, linear elastic FE analyses, which are more rapid and easy than elastic-plastic ones, can be kept valid in the presence of small scale yielding conditions [7].

6. Conclusions

A full-scale welded component of valve actuators, the scotch yoke, made of steel has been fatigue tested under nominal axial loading. Residual mean stresses have been eliminated by stress relieving, so that a nominal load ratio R=-1 has been applied . Despite the presence of the stress concentration due to weldment**,** all tested yokes experienced crack initiation likely located at the fillet radius of the wing, except in two case where fatigue crack initiates at the tip of the keyway. The fatigue crack paths have been analysed by means of liquid penetrant inspections.

The fatigue strength class of the yoke has been determined and it has been shown that the design point, based on the relevant European Standard, is on the safe side. Then, 3D linear elastic FE analyses have been adopted to convert the experimental fatigue results from the nominal axial load to the local peak stress at the point of crack initiation.

Finally, two methodologies for fatigue strength assessment of the considered scotch yokes have been proposed on the basis of experimental fatigue data relevant to smooth and sharp V-notched specimens subjected to axial loading. By comparing theoretical predictions with the experimental results of scotch yokes, it has been shown that the methodology based on sharp V-notched specimens provide the best agreement avoiding an excessive conservatism and keeping valid linear elastic FE analyses.

7. Acknowledgments

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