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Biaxial high-cycle fatigue life assessment of ductile aluminium cruciform specimens

R.A. Cláudio [a,*], L. Reis [b], M. Freitas [b]

[a] ICEMS, ESTSetúbal, Instituto Politécnico de Setúbal, Campus IPS, Estefanilha, 2910-761 Setúbal, Portugal
[b] ICEMS, Instituto Superior Técnico, ULisboa, Av. Rovisco Pais, 1049-001 Lisboa, Portugal

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ABSTRACT

Several results of experimental high-cycle fatigue tests carried out on a new biaxial in plane testing system are presented, covering a wide range of biaxial stress states, including proportional and non-proportional, performed on cruciform specimens machined from ductile aluminium (A1050-H14) sheet plate with 3 mm thickness. A specially designed cruciform specimen for crack initiation and low capacity test machines was used to perform fatigue tests. This specimen was optimized for the test machine in order to achieve the maximum uniform stress at the centre gauge area, while keeping the remaining stresses at least 20% lower than the stress at centre. Several of the most common multiaxial fatigue criteria were used to determine an equivalent uniaxial fatigue damage parameter for all the specimens that were tested experimentally and to determine which criteria provide a better correlation to the experimental results. From the results it was shown that most of the criteria provide non-conservative estimations. In overall, better results were found by the Minimum Circumscribed Ellipse criterion and also when using the damage parameter related with the shear stress computed by the Minimum Circumscribed Ellipse in the modified non-linear criterion proposed by Carpinteri and Spagnoli.

1. Introduction

Many machine components and structures are generally subjected to multiaxial fatigue loading conditions [1]. Fatigue life evaluation of mechanical components under complex loading conditions is of great importance in order to optimize structural design, and improve inspection and maintenance procedures. However fatigue experiments are much more easily performed under uniaxial loading and constant amplitude but most practical problems associated with metal fatigue in structural elements are associated with multi-axial loading. For example, rotor shafts in electric power plants, propeller shafts in ships, aircrafts fuselage and so on. The most common multiaxial fatigue specimens and testing fixture are therefore associated with bending-torsion or tension–torsion testing machines and in-phase and out-of-phase fatigue tests are available in the literature for a wide range of materials and loading paths [1,2].

Less attention has been paid to fatigue tests performed under biaxial loading such those present on pressure vessels or pressurized aircraft cabins [3–6]. These examples cover different circumstances of cyclic nature of loading and also variations in biaxiality including in-phase versus out-of-phase, different ratios of biaxiality, etc. The cost and availability of biaxial fatigue testing machines that can perform biaxial loading for example in cruciform specimens is certainly the cause. Fatigue under in-plane biaxial loading conditions remains very much an unexplored science, demanding appropriate testing technology. Test systems have been developed over the years to perform tests under static and cyclic biaxial loading using cruciform specimens, however the costs involved are usually the principal drawback. Some of these systems constitute simple and robust designs whose application using fewer actuators can perform a limited combination of biaxial loading conditions on sheet material.

2. Some multiaxial fatigue criteria

Several multiaxial criteria have been proposed in the literature to evaluate the fatigue strength of structural components [1,2]. For high-cycle fatigue the most accepted criteria are stress-based. Most of them reduces a given multiaxial stress state to an equivalent uniaxial stress state, which is compared with the uniaxial fatigue strength of the material.

First proposed models are based on traditional static yield criteria like the maximum normal stress theory, maximum shear stress

* Corresponding author.
E-mail address: ricardo.claudio@estsetutbal.ipt.pt (R.A. Cláudio).

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or the octahedral shear stress, this last one also called the von-Mises theory. It is nowadays known that these do not provide accurate approaches for most of the loading cases. However a modification of the von-Mises (v-M) criterion is still included in the ASME pressure vessel code for pressurized pipes [7], according to:

$$\Delta\sigma_{eq} = \frac{1}{\sqrt{2}}\sqrt{\Delta(\sigma_1-\sigma_2)^2+\Delta(\sigma_2-\sigma_3)^2+\Delta(\sigma_3-\sigma_1)^2} \quad (1)$$

where $\Delta\sigma_{eq}$ is the equivalent stress state range and σ_1, σ_2 and σ_3 are the principal stresses. Note that the value of $\Delta(\sigma_1-\sigma_2)$ must be computed for the whole loading cycle.

Sines [8] and Crossland [9], proposed a popular high cycle fatigue criterion based on the octahedral shear stress:

$$\sqrt{J_{2,a}}+k\sigma_H=\beta \quad (2)$$

where $\sqrt{J_{2,a}}=\Delta\sigma_{eq}/(2\sqrt{3})$ is the second deviatoric stress invariant, k is a material parameter, which takes into account the influence of hydrostatic stresses, σ_H and β is the equivalent uniaxial damage parameter that is related with the fatigue strength of the material at a specified number of loading cycles.

For high cycle fatigue is generally accepted that the fatigue strength of the material at a specified number of loading cycles N is related by:

$$\lambda(N)=A(N)^b \quad (3)$$

where A and b are defined as the fatigue strength coefficient and the fatigue strength exponent respectively. The value of A must be calculated for each fatigue criteria because the way how the normal and shear stresses are accounted is different. Therefore in the following criteria that will be used in this study, the β parameter will be replaced by $\lambda(N)$, as presented in Eq. (3).

The value of k can be calculated by knowing the reversed bending fatigue strength σ_{-1} and the reversed torsion strength τ_{-1} of the material as follows:

$$k=\frac{3\tau_{-1}}{\sigma_{-1}}-\sqrt{3} \quad (4)$$

The difference between Sines and Crossland criteria is in the value of the hydrostatic stress. While Crossland suggests to use the maximum value of the hydrostatic stress $\sigma_{H,\max}$, Sines uses the mean value of the hydrostatic stress $\sigma_{H,\text{mean}}$.

Findley [10] proposes the shear stress amplitude at the maximum value of the normal stress on the critical plane as damage parameter. This can be computed by identifying the maximum combination of the shear stress and normal stress over all the planes:

$$\left\{\frac{\Delta\tau}{2}+k\sigma_n\right\}_{\max}=\lambda(N) \quad (5)$$

Latter Fameti and Socie (F–S) [11], proposed a parameter in which the shear cyclic strain $\Delta\gamma$ is modified by the normal stress to include crack closure effects:

$$\frac{\Delta\gamma}{2}\left(1+k_{FS}\frac{\sigma_{n\max}}{\sigma_y}\right)=\lambda(N) \quad (6)$$

k_{FS}/σ_y represents the sensitivity to the normal stress and as first approach this parameter can be replaced by $1/\sigma_{-1}$ [1].

The stress state in a plane can be decomposed in a normal and shear component. During a load cycle the normal component remains perpendicular to the critical plane but the shear stress describes a closed curve as explained by Papadopoulos [12]. This curve can be circumscribed by a circle of radius R_a, being this approach usually designated by the Minimum Circumscribed Circle (MCC), see Fig. 1. The shear stress amplitude is defined as:

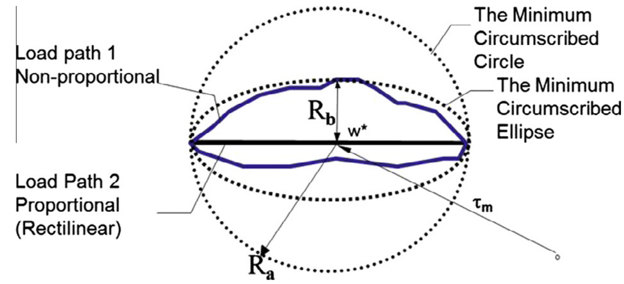

Fig. 1. Comparison of the MCE and MCC approaches for evaluating shear stress amplitude.

$$\sqrt{J_{2,a}}=R_a \quad (7)$$

In order to take into account the non-proportional loading effect the Minimum Circumscribed Ellipse (MCE) approach was proposed by Freitas et al. [13]. In this case the closed curve described by the shear stress vector can be enclosed by an ellipse with major radius R_a the major radius (equal to MCC) and miner radius R_b. The MCE defines the shear stress amplitude as:

$$\sqrt{J_{2,a}}=\sqrt{R_a^2+R_b^2} \quad (8)$$

To facilitate the computation of the values of R_a and R_b the stress components of can be transformed into a 5-dimensional deviatoric stress space denoted as E_5 [14].

For both MCC and MCE criteria the mean stress effects can be included in the same way as proposed by Sines or Crossland.

Carpinter et al. [15] (modified C–S) proposed a non-linear damage parameter that is a modified version of the original C–S criterion [16]:

$$\sqrt{(\sigma_a)^2+(\tau_a)^2\left(\frac{\sigma_{-1}}{\tau_{-1}}\right)^2}=\lambda(N) \quad (9)$$

in which σ_a and τ_a are the applied normal and shear stress amplitudes. The shear stress amplitude is determined by applying the same procedure proposed by Papadopoulos [12].

In this work another modification to the modified C–S criterion is proposed by considering as shear stress amplitude the parameter defined by Eq. (8) (MCE) instead of Eq. (7) (MCC). MCE includes as addition a non-proportional load effect factor which may provide better results as it what was proven for tension–torsion loading conditions in previous works made by some of the authors [13].

Estimations will be made with the aforementioned multiaxial fatigue criteria to find which ones provides better results for the particular experimental loading conditions, in tension–tension, presented in this work.

3. Experimental procedure

3.1. Test machine

The experimental tests were conducted on a new in plane biaxial fatigue testing machine designed and constructed by IPS and IST institutions [17]. The machine is all electrical and was developed with four iron-core linear motors (one of the most powerful models available on the market), one for each arm of the specimen. This machine is an alternative to the servo-hydraulic actuation used for decades because of their versatility, fast response and force capacity. The servo-hydraulic machines have several disadvantages like the amount of power that is lost in the form of heat and the quantity of oil needed, resulting in enormous running and maintenance costs. The solution proposed, by using electrical motors instead of