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Prediction of fatigue life in aircraft double lap bolted joints using several multiaxial fatigue criteria

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ABSTRACT

In this research, the effects of torque tightening on the fatigue strength of 2024-T3 aluminium alloy double lap bolted joints have been studied via experimental and multiaxial fatigue analysis. To do so, three sets of the specimens were prepared and each subjected to different levels of torque i.e. 1, 2.5 and 5 N m and then fatigue tests were carried out at various cyclic longitudinal load levels. A non-linear finite element ANSYS code was used to obtain stress and strain distribution in the joint plates due to torque tightening of bolt and longitudinal applied loads. Fatigue lives of the specimens were estimated with six different multiaxial fatigue criteria by means of local stress and strain distribution obtained from finite element analysis. Multiaxial fatigue analysis and experimental results revealed that the fatigue life of double lap bolted joints were improved by increasing the clamping force due to compressive stresses which appeared around the hole.

1. Introduction

Removable joints such as bolts, rivets or pins are frequently used in aerospace industry for connecting various parts. Among the mentioned removable joints, bolted joints are the most important elements in aerospace structures. Nevertheless, the existence of geometrical discontinuity as a consequence of essential hole drilling in bolted joints results in stress concentration and thus increases the tendency of fatigue crack to initiate and grow under cyclic loading [1–4]. Conversely, due to ease of assembling and possibility of dismantling, bolted joints have excessive use in aerospace constructions. Therefore, it is of great importance to reduce the effect of the stress concentration and attain enhanced fatigue life [5]. According to results of previous researches, bolted joints have higher fatigue strength than welded, riveted, and pinned joints [6,7].

When a nut and bolt are used to join mechanical members together, the nut is tightened by applying a torque and a force is exerted on the nut and then the bolt and nut are pulled towards each other. This force creates tension in the bolt, which clamps the assembled parts of the joint together. Preload is the technical term for the tension caused by torque tightening the nut that holds the assembled part together [8–13].

It is evident that, most of machines and structures, such as aircraft structures, in their service lives are subjected to multiaxial stresses in which two or three principal stresses vary with time; i.e., the corresponding principal stresses are out-of-phase or the principal directions change during a cycle of loading. Therefore, multiaxial fatigue analysis becomes an important tool for estimating the fatigue strength of these components. The origin of multiaxiality in stress is dependent on various parameters such as type of loading, complex geometry of the mechanical parts, residual stresses or pre-stresses, etc. In order to estimate fatigue strength of the components, many multiaxial fatigue criteria have been proposed in the literature for metals.

From technical point of view, multiaxial criteria for prediction of fatigue strength of mechanical components may be categorized into three main groups [14], namely stress based criteria, strain based criteria and energy based approaches. As for the stress criteria proposed by Susmel and Lazzarin [15], McDiarmid [16], Crossland [17] and so on, are based only on stress and are suitable for high cycle fatigue when the deformation is elastic or the plastic strain is small. The strain criteria such as the Brown–Miller model [18], Fatemi–Socie [19], Li–Zhang [20] and Wang–Brown model [21], are appropriate for such cases in which there is significant plasticity. The energy based multiaxial criteria such as the Smith–Watson–Topper model [22], Glinka et al. [23] and Varani-Farahani [24] include both stress and strain terms.

Multiaxial fatigue criteria, from another point of view, can be categorized as those criteria which use the critical plane concept

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and those do not use this concept. In the multiaxial fatigue criteria based on critical plane, initially a material plane on which a combination of some stress components has the maximum value has been determined. Then, fatigue parameters as combinations of the maximum shear strain or stress and normal strain or stress on the critical plane have been calculated [19]. The criteria will be checked then on this plane. Like to classical models, critical plane models can be stress-, strain- or energy-based.

The concept of critical planes was primarily proposed by Brown and Miller [18]. Brown and Miller proposed a strain-based parameter that considers that fatigue life to be a non-linear function of strain. The critical plane of this parameter is the plane of maximum shear strain. Firstly, the history of strain on the critical plane is analyzed; and then strain parameters are used to quantify the damage parameters on the critical plane. Various terms have been proposed for different materials and experimental results [25,26].

Socie [25] proposed that the fatigue life criterion should be based on a physical mechanism. Socie modified the SWT [22] parameter for a material with tensile-type failures, by taking the view that crack growth is perpendicular to the maximum tensile stress.

The energy criteria have been used in conjunction with the critical plane approach, as proposed by Liu [27], and Glinka et al. [23]. Varani-Farahani (VF) [24] proposed critical plane based energy parameters that are weighted by axial and shear fatigue properties of the material.

In fact, many efforts have been made to evaluate the performance of multiaxial fatigue criteria for various materials, notched and unnotched components, different loading conditions and stress (and strain) states by several authors such as Papadopoulos et al. [28], Sonsino [29], Brown and Miller [30], You and Lee [31], Macha and Sonsino [32], Wang and Yao [33], Chakherlou and Abazadeh [34], Varvani-Farahani et al. [35], and Jiang et al. [36]. However, due to the complexity of this challenging problem (fatigue life estimation for multiaxial loading condition) and its practical application, much additional studies are still needed to evaluate the accuracy and reliability of the multiaxial fatigue criteria in design, life estimation, and failure assessment particularly for practical specimens (components).

In order to estimate fatigue life in the double lap bolted joints, for a reliable design, using multiaxial fatigue criteria is needed due to the complexity of stress distribution as a result of the tightening torque and longitudinal remote load. Therefore, the current investigation has sought to improve the existing body of knowledge about the performance of these multiaxial fatigue criteria in bolted joints in general and the effect of torque tightening on the fatigue life of double shear lap joints in particular.

In the present study, the effects of torque tightening on the fatigue life of 2024-T3 aluminium alloy double lap bolted joints was investigated via experimental and multiaxial fatigue analysis. To do so, three batches of specimens were prepared and each subjected to torque of 1 N m, 2.5 N m and 5 N m and then fatigue tests were carried out at different cyclic longitudinal load levels. A nonlinear finite element ANSYS code was used to obtain stress and strain distribution in the joint plates due to torque tightening of bolt and longitudinal applied loads. Fatigue lives of specimens were estimated with six different multiaxial fatigue criteria, i.e. SWT, Glinka, KBM, FS, Crossland and VF by means of local stress and strain distribution obtained from finite element analysis.

2. Experimental procedures

Aluminium alloy 2024-T3 with thickness of 2 mm, was selected for manufacturing of the specimens employed in this investigation. This type of Aluminium alloy has been extensively used in aerospace structures. The mechanical properties of this material have been illustrated in Tables 1 which obtained from tension (static) tests in accordance with ASTM: E8/E8M-13a. In addition, configuration and dimensions of test specimens were designed and manufactured to conform as nearly as possible to ASTM: E466-07, and have been illustrated in Fig. 1.

Table 1
Mechanical properties of 2024-T3 aluminium alloy.

Young's modulus (GPa)	Yield stress (MPa)	Tensile strength (MPa)	Poisson's ratio	Elongation (%)
72	315	550	0.33	18

Fastener holes of 5 mm diameter were drilled and reamed. Hex head M5 (class 10.9) bolts along with suitable types of steel washers and nuts were used to prepare the joint as illustrated in Fig. 1.

In order to use the bolt and nut within the elastic region, some primary experimental tests were carried out and the obtained results indicated that initial plastic deformations started at approximately 8 N m at threads [37]. To measure the clamp force (bolt axial tension) at different applied torques, a special experimental method was designed using a steel bush that was placed between the nut and the plate. At the bush outer surface, two strain gauges were stuck to measure the compressive axial strain and so the stress in the bush using Hooke's stress–strain law. Having the bush cross-sectional area at hand and the axial stress, the axial force in the bush and then the clamp force has been determined. The used method and the bush dimensions were shown in Fig. 2.

To calibrate the applied torque and clamping force, torques were applied in 1 N m increments from 1 to 7 N m to the nut using a torque wrench, and then the axial strains were recorded for each value of the torques. This test was repeated three times for each case to obtain the mean value of compressive strains (ε_m), and determine the corresponding clamping forces using Eq. (1). The elastic modulus for the bush material (E_{bush}) was also experimentally determined in order to obtain the accurate values for the mean axial clamping force.

$$F_{cl} = E_{bush}A_{bush}\varepsilon_m = 204,188 \times \frac{\pi}{4}\left(9^2 - 5^2\right)\varepsilon_m = 89.8 \times 10^5\varepsilon_m(N) \tag{1}$$

where A_{bush} is the area of the bush cross section. The relation between the measured clamping forces and the applied torques for the specimen is shown in Fig. 3. As the figure shows, the relationship between the clamping force and the applied torque is linear in the range of applied torques. This indicates that the bush material is still in its elastic region, even under the maximum applied torque.

In the next step, three batches of specimens were prepared in which with using a torque-wrench the bolts were tightened up to required amounts of torques, i.e. 1 N m, 2.5 N m and 5 N m which created clamping forces equal to F_{cl} = 976, 2440 and 4880 N respectively, according to the linear equation obtained from Fig. 3. Fatigue tests have been carried out for stress ratio of 0.1 and frequency of 10 Hz using servo-hydraulic 250 kN Zwick/Roell fatigue testing machine (Fig. 4). In each case, five fatigue tests were performed with different maximum remote longitudinal loads.

The resulting average life from the constant amplitude fatigue tests was plotted in accordance with ASTM: E468-11, and displayed in Fig. 5. As it can be seen from the figure, increasing the clamping force leads to a considerable enhancement in the fatigue life as expected. The fractured specimen under fatigue tests with clamping force F_{cl} = 976 N that subjected to maximum remote longitudinal load equal to 7.2 kN is shown in Fig. 6.