FREQUENCY-DOMAIN FATIGUE ANALYSIS OF BONDED JOINT UNDER RANDOM LOADS

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Abstract. *The use of adhesively bonded structures has increased over the years, together with the development of composite materials. This work investigates a procedure for fatigue life prediction of this type of structures under random loads, in particular, the cohesive failure of the adhesive. Dirlik's method is employed to predict the stress response in the adhesive layer, from which the fatigue life is obtained. The effect of damping is investigated, and it is observed that an increase in damping by 50% increases the fatigue life by more than 14 times.*

1. INTRODUCTION

The aircraft industry is always looking for new technologies to improve aircraft performance and safety, meeting several regulatory requirements. Also, accurate information on the fatigue life of aircraft structures is increasingly required to improve aircraft safety and dispatchability. The last decades have also seen an increase in the use of lightweight components made of composite materials for secondary and even primary structures. Adhesive joining has attractive intrinsic advantages for composite components compared to the classical joining techniques using riveting fasteners, for two reasons. Rivets require that holes be drilled in the component, which is highly disadvantageous for continuous fiber-reinforced structure. In addition, rivets are very sensitive to fatigue, as they introduce stress concentrations which may lead to complete failure of an assembled component. Thus, it becomes important to characterize the fatigue life of the bonded joints.

Adhesively bonded structures are subjected to two types of failure scenarios: cohesive failure or adhesive failure. Cohesive failure is the one that occurs in the bulk layer of the adhesive, while adhesive failure happens at the interface between the adhesive and the adherent. It should be noted that cohesive failure is usually the preferred failure mode. In this work, the fatigue life of a skin-to-stiffener bonded joint under spectrum loads will be evaluated, with the assumption of cohesive failure of the adhesive.

2. FATIGUE ANALYSIS FOR RANDOM LOADS

Random loads can be characterized in the frequency-domain by their Power Spectrum Density (PSD). Due to their random nature, these loads are better described by a Probability Density Function (PDF), rather than by a time history. The PDF gives the probability of occurrence of a certain range of load amplitudes. For wideband, Gaussian process, Dirlik [1] proposed a semi-empirical method for the estimation of the stress PDF based on its PSD. The method is based on the spectral moments of the PSD, which, for a one-sided PSD, $PSD_X(f)$, are defined by:

$$m_i = \int_0^\infty f^i \times PSD_X(f)df \tag{1}$$

where m_i is the spectral moment of i-th order, and f denotes frequency. The PDF of the stress amplitudes, $PDF(S)$, can then be calculated as:

$$PDF(S) = \frac{1}{\sqrt{m_0}}\left[\frac{G_1}{Q}e^{-\frac{Z}{Q}} + \frac{G_2 Z}{R^2}e^{-\frac{Z^2}{2R^2}} + G_3 Z e^{-\frac{Z^2}{2}}\right] \tag{2}$$

where Z is the normalized amplitude, and x_m is the mean frequency, which are defined as:

$$Z = \frac{S}{\sqrt{m_0}} \tag{3}$$

and

$$x_m = \frac{m_1}{m_0}\left(\frac{m_2}{m_4}\right)^{\frac{1}{2}} \tag{4}$$

Another important parameter is the spectral width, which takes the general form of:

$$\alpha_i = \frac{m_i}{\sqrt{m_0 m_{2i}}} \tag{5}$$

The parameters G_1, G_2, G_3, R and Q are defined as:

$$G_1 = \frac{2(x_m - \alpha_2^2)}{1 + \alpha_2^2}, \qquad G_2 = \frac{1 - \alpha_2 - G_1 + G_1^2}{1 - R}, \qquad G_3 = 1 - G_1 - G_2 \tag{6}$$

$$R = \frac{\alpha_2 - x_m - G_1^2}{1 - \alpha_2 - G_1 + G_1^2}, \qquad Q = \frac{1.25(\alpha_2 - G_3 - G_2 R)}{G_1} \tag{7}$$

Knowing the PDF of the stress, and assuming a linear combination of damage (Palmgren-Miner's rule), the expected damage per second, $E[D]$, can be calculated as:

$$E[D] = \frac{freq}{C}\int_0^\infty S^b \times PDF(S)dS \tag{8}$$

where C and b are the material coefficient and the Basquin exponent of the S-N curve of the material, and $freq$ is the expected peak occurrence per second, which can be calculated as:

$$freq = \sqrt{\frac{m_4}{m_2}} \tag{9}$$

3. GEOMETRY AND NUMERICAL MODEL

The component used in this work consists of an aluminum stiffener bonded to a Fiber Metal Laminate (FML) skin, which is commonly used in aircraft structures such as in fuselage panels. The geometry chosen is the one used in Stiffener Pull-Off Tests (SPOT), which are used to characterize the performance and failure mode of this type of bonding. The geometry and material properties used in this work were taken from Freitas and Sinke [2], who used SPOT to characterize the behavior of two different types of adhesives.

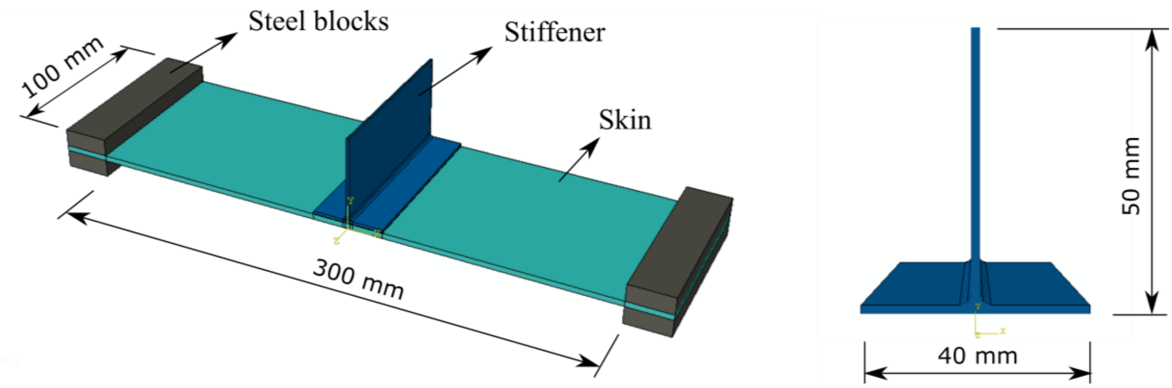

Figure 1 – Model used in the simulation (left) and detail of the stiffener (right).

The stiffener is an extruded inverted T-shape stiffener, 1.5 mm thick, made of the aluminum alloy 2024-T3. The skin is made of Glare 5-3/2-0.3, which consists of three 2024-T3 aluminum alloy layers 0.3 mm thick bonded together with glass fiber reinforced plastic (GFRP) with the layup [0º/90º/90º/0º]. The final layup of the skin is [Al/[0º/90º/90º/0º]/Al/[0º/90º/90º/0º]/Al], with a total thickness of 3.134 mm.

The adhesive layer is modeled with 0.35 mm thick, and perfect adhesion is assumed between the skin and the stiffener. To obtain a realistic result, the bonding is assumed to be made of an adhesive whose fatigue properties have already been characterized in the literature. Therefore, the mechanical and fatigue properties of the adhesive used in the joint came from Beber et al [3], who studied the fatigue behavior of a toughened one-component epoxy-based adhesive. The structure is held in place by four steel blocks at which a fixed boundary condition is applied (i.e. zero displacements and rotations), and perfect adhesion is assumed between the steel blocks and the skin. The skin has a width of 100 mm and a span of 300 mm, as shown in Fig. 1, which also shows the stiffener's dimensions. The whole structure is modeled in Abaqus™ with 83076 second-order solid elements (C3D20).

Table 1 shows the mechanical properties of the FML skin, aluminum stiffener, and adhesive used in the simulation.

Table 1 – Mechanical properties of the materials used in this work.

GFRP		**Al 2024-T3**		**Adhesive**	
Property	**Value**	**Property**	**Value**	**Property**	**Value**
E_{11}	48.9 GPa	E	72.4 GPa	E	1.6 GPa
E_{22}, E_{33}	5.5 GPa	υ	0.33	υ	0.4
G_{13}, G_{23}, G_{12}	5.5 GPa	ρ	2.78 kg/m^3	ρ	1.20 kg/m^3
$\upsilon_{12}, \upsilon_{23}, \upsilon_{13}$	0.33	-	-	-	-
ρ	2.48 kg/m^3	-	-	-	-

The S-N curve of the adhesive is obtained from the S-N curve of the bulk adhesive under room temperature shown in [3].

To obtain the Frequency Response Function (FRF) of the structure, a unity impact load is applied at the top corner of the stiffener, in a frequency range from 50 to 180 Hz, and the stress response is registered for a point located at the bottom corner of the stiffener in the middle of the adhesive layer. Both points are shown in Fig. 2.

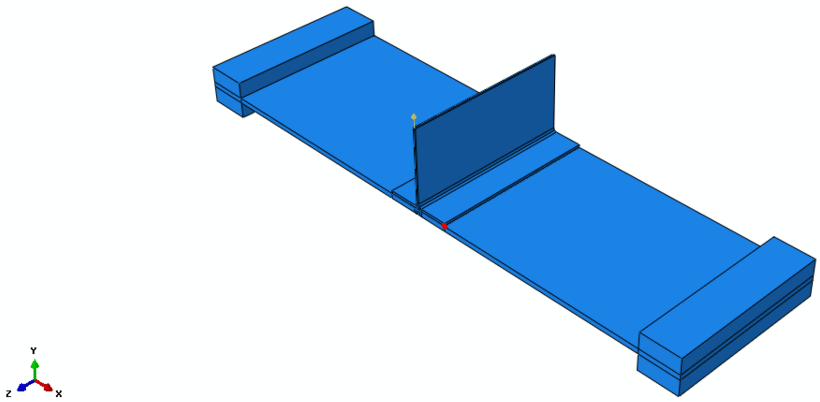

Figure 2 – Impact force (yellow arrow) and point from which the stress response is recorded (red dot).

The FRF can then be used to obtain the PSD of the stress response, PSD_S, to a given force input, PSD_F, as shown:

$$PSD_S = |FRF|^2 PSD_F \tag{10}$$

4. RESULTS

The numerical analysis revealed that the first natural frequency of the structure corresponds to a bending mode that happens at 133.7 Hz. The PSD of the stress response was obtained for the case of a constant force PSD of 2.5 N^2/Hz applied at the top of the stiffener, as shown in Fig. 2, in the frequency range from 50 to 180 Hz, calculated according to Eq. (10). To investigate the impact of damping on the system, three levels of modal damping were applied in the first mode of the simulation: 1% damping, 1.5% damping, and 2% damping. Then, applying Eq. (2), the PDF of the stress amplitudes were obtained, which are shown in Fig. 3.

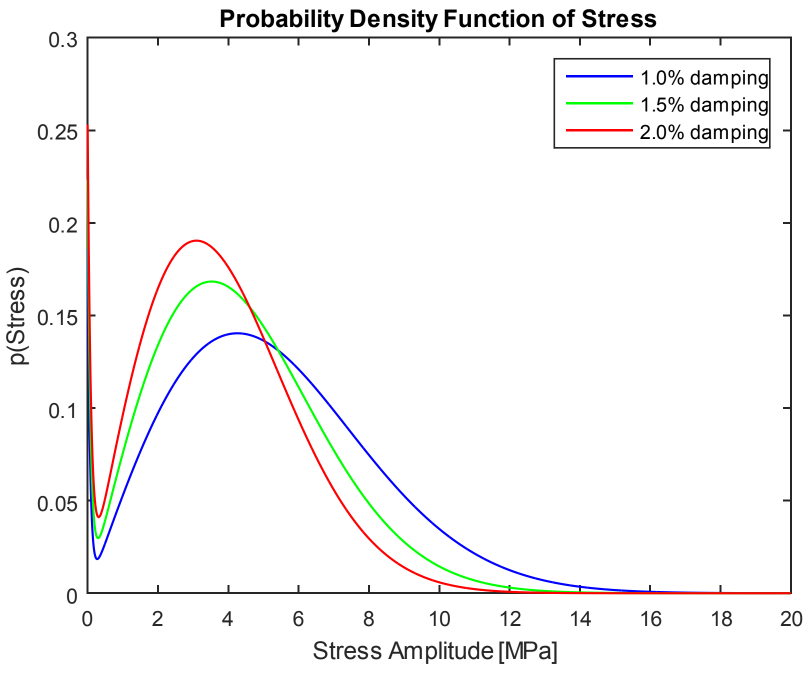

Figure 3 – PDF of the stress response for three levels of modal damping.

It is possible to observe that, as damping increases, the PDF curve is shifted to the left, showing a higher probability of occurrence of stress levels of lower amplitude, and also decreasing the maximum stress amplitude expected. To quantify the impact of damping on the fatigue life of the structure, Eq, (8) was used. However, the fatigue data available for the adhesive was obtained from experiments with a stress ratio of 0.1, while the random stress originated by the force PSD can be approximate to a constant stress ratio of -1. Thus, the Smith-Watson-Topper equation was used to convert the PDF obtained to an equivalent PDF with 0.1 stress ratio. The results for fatigue life are shown in Tab. 2.

Table 2 – Fatigue life for different damping levels.

Damping Effect			
Damping	1%	1.5%	2%
Fatigue Life	130.3 min	1931.2 min	12168.2 min

As it can be seen, damping has a huge impact on the fatigue life of the structure. As damping increases from 1% to 1.5%, the fatigue life increases by one order of magnitude. Similarly, from 1.5% to 2% damping, the fatigue life increase by more than six times.

5. CONCLUSIONS

Crack nucleation is investigated using the S-N approach and a probability density function of the stress calculated via Dirlik's method. The method is applied to predict the fatigue life of a bonded-joint under spectrum loads assuming cohesive failure of the adhesive.

Numerical results show the strong influence of damping in the structure's response and its effect on the fatigue life, in which an increase in the modal damping of the structure also leads to an increase in its life. This happens due to the dissipation effect caused by damping, which decreases the stress level in the adhesive layer.

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