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Study of surface bonding imperfection effects on equivalent identified dynamic Young's and shear moduli using a modal based joint identification method

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ABSTRACT

In this paper the effects of defects on the equivalent identified dynamic Young's and shear moduli of flexible structural adhesive joints have been investigated. Adhesive joints were subjected to initial surface bonding imperfection, in order to simulate the defects. Several debond patterns have been applied artificially on single-lap joint specimens and joint identification process has been performed for both bending and shear modes to see how surface bonding imperfection can affect equivalent identified dynamic Young's and shear moduli of the adhesive. Using a direct modal based method to identify mechanical characteristics of joints has been shown that degradation of the equivalent moduli of debonded joint is correlated to both frequency and mode shapes. The results reveal that debonding is easily detectable in bending modes whereas degradation in shear modes is also correlated to debonding orientation and mode shapes. Three identical specimens have been tested for each case to prove the consistency of the results.

1. Introduction

Bonding, as a method to join two different materials with different mechanical properties together, has many advantages over other traditional methods. High strength to weight ratio, distributed stress field over the bonding surface, working as a sealant simultaneously, etc. [1,2] have attracted considerable interest from manufacturers and researchers in this type of connection medium. This interest has resulted in large volume of research work in the relevant area [3].

On the other hand, adhesives have some issues that limit their application in joining major parts of advanced structures like aircrafts or space structures. One of these limitations is the absence of a low-cost and reliable Non-Destructive Evaluation (NDE) method to monitor the adhesive joint's structural health state and remaining service life of it intelligently [4] needs to be further developed. Although, Local Damage Detection (LDD) methods like Acoustic emission [5], X-ray [6], etc. can be used for fault detection purposes, it is limited to large scale structures with low sensitivity outcome in addition to large amount of time and cost.

Vibration analysis is a tool that can expose dynamic properties of structures by means of their time responses, modal parameters or frequency response functions. Any kind of damage and/or material degradation can be modeled as changes in mechanical properties. In many experimental test subjects it has been observed that damage can affect measured structural

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dynamic parameters and therefore vibration analysis, especially equipped with neural network and genetic algorithm, has the potential to build an intelligent NDE package [7]. Most of the recent experimental studies on vibration based damage detection have dealt with composite plates, sandwich panels, beams, etc. [8–10] that could be viewed as the main mechanical parts of a structure. It is expected that vibration analysis of a structure concerned with damage and/or imperfection in its main parts can reveal the existence of damage and therefore the pattern of changes in dynamical properties can be derived through experiments with different amounts and locations of damages.

Because of a joint's role in connecting structures with different impedances at connection points, the joining medium is usually subjected to high level of strains and hence joint's dynamic mechanical properties can significantly affect dynamical response of the structure [11–13]. This effect can be used in reverse order to identify the bonding medium (here adhesive) mechanical properties.

In order to understand clearly the effects of damage and/or imperfection on joint apparent mechanical properties, a joint identification method will be used. In this research several artificially built initial debond on the adhesion surface of single-lap joints have been applied and a direct modal based joint identification method has been developed to investigate the pattern of changes in the equivalent identified dynamic Young's and shear moduli of a nominated adhesive.

2. Modal based joint identification

Nobari et al. [14] has provided a Direct Modal Based (DMB) identification method to identify mechanical characteristics of joints. In this method modal parameters (natural frequencies and mode shapes) measured from experiments are considered as the true data and any discrepancy between Finite Element (FE) model which is called analytical model in this paper, and true model are justified with miss-modeled mechanical properties of the joint.

Similar to well-known Inverse Eigen Sensitivity (IES) identification method [15], DMB identification method takes advantage of the differences between analytical and experimental modal parameters. However, DMB has certain advantages over IES, the most important one being the fact that mode pairing is not necessary when implementing DMB. Mode pairing task which is an integral part of IES method can be very difficult in the case of practical industrial structures.

In order to omit errors caused by other parts of the structure, a model updating process should be applied on every adherent structure, separately. A brief description of the method is presented here:

First the following definitions are made:

A	Analytical model derived from FE
X	True model derived from experimental test
$[\ddots_a]$	Analytical parameters
$[\ddots_x]$	Experimental parameters
$[\ddots_x^a]$	Updated analytical parameter which has minimum discrepancy with true model
Δ	Miss-modeled structure that should be added to the analytical model to update it

Assuming that Δ is fully compatible with A, equation of motion for coordinates of A can be written as follow:

$$[M]\{\ddot{x}_a\}+[K]\{x_a\}=\{f\} \tag{1}$$

where, $\{f\}$ is the vector of interconnection forces between coordinates of A and Δ. Now if we transform coordinates from physical to principal coordinates and separate n_a analytical modes into kept coordinates related to modes that are in the frequency range of interest as $\{p_{ak}\}$ and eliminated coordinates related to modes out of the frequency range of interest as $\{p_{ae}\}$, the following equations are derived:

$$\{x_a\}_{n_a\times 1}=[\varphi_a]_{n_a\times n_a}\{p_a\}_{n_a\times 1}=[\varphi_{ak}|\varphi_{ae}]\begin{Bmatrix}\{p_{ak}\}\\ \{p_{ae}\}\end{Bmatrix} \tag{2}$$

$$\{\ddot{p}_{ak}\}+[\omega_{ak}^2]\{p_{ak}\}=[\varphi_{ak}]^T\{f\} \tag{3}$$

$$\{\ddot{p}_{ae}\}+[\omega_{ae}^2]\{p_{ae}\}=[\varphi_{ae}]^T\{f\} \tag{4}$$

where $[\varphi_a]$, $[\varphi_{ak}]$ and $[\varphi_{ae}]$ are the mode shape matrices of the analytical modes, kept modes and the eliminated modes, respectively. Now we assume that the minimum frequency of the eliminated modes is much higher than the maximum frequency of the interest range, $\omega_{\max\ of\ investigation} \ll \omega_{ae}$, so the inertia of the kept modes are negligible relative to the stiffness of the eliminated modes and Eq. (4) can be rewritten like below:

$$[\omega_{ae}^2]\{p_{ae}\}\cong[\varphi_{ae}]^T\{f\} \tag{5}$$

Because of compatibility and equilibrium between A and Δ, the equation of motion of Δ is:

$$[Z_\Delta]\{x_a\}=-\{f\} \tag{6}$$