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Active vibration control of a composite sandwich plate

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ABSTRACT

Active vibration control of a free-edge rectangular sandwich plate is proposed and tested. The experimental setup consists of a honeycomb panel having a carbon-fiber reinforced polymer (CFRP) outer skins and a polymer-paper core, subjected to an orthogonal disturbance, due to an electrodynamics exciter and controlled by Macro Fibre Composite (MFC) actuators and sensors. MFC parches consist of rectangular piezoceramic rods sandwiched between layers of adhesive, electrodes and polyamide film. The MFC actuators and sensors are controlled by a programmable digital dSPACE® controller board. The control algorithm proposed in this paper is based on the Positive Position Feedback (PPF) technique and is successfully applied with different combinations of inputs/outputs (Single Input Single Output, MultiSISO, Multi Input Multi Output) in order to control the first four normal modes. The control appears to be robust and efficient in reducing vibration in linear (small amplitude) and nonlinear (large amplitude) vibrations regimes, although the structure under investigation exhibits a relativity high modal density, i.e., four resonances in a range of about 100 Hz. The control strategy allows to effectively control each resonance both individually or simultaneously.

1. Introduction

Experimental studies in nonlinear dynamics play a key role and have a high relevance both to validate numerical and theoretical models and to highlight and discover complex behaviours of mechanical systems and structures. Nowadays, the increasing use of new materials (composite carbon fibre, etc.) makes more important to refine the testing procedures due to the intrinsic nonlinear properties and their innovative applications. In such work an accurate study of a thin walled structure was carried out to better understand the nonlinear dynamics of shells and plates; once the behaviour has been defined experimentally a further step has been made to solve the problems due to nonlinear vibrations, applying different kinds of active vibration control. Indeed, today's industry makes an extended use of thin walled composite structures, especially in the field of aerospace. Composite laminated structures are characterised by an extremely high specific stiffness, together with a good flexibility in achieving complex shapes. As a typical example, the bodywork and elements of the frame of racing cars and bikes can be mentioned. Typically loose fixings and fastenings further contribute to the development of large amplitude vibrations, which can result in noise disturbance for passengers, functional problems or even sources of danger.

In general, the simplest way to reduce vibrations is to design the system with additional damping, by using special materials or adding physical damping devices. This approach is called passive vibration control (or redesign) and is a very well developed subject area for linear vibration problems, see Soong and Dargush [1] and Lam et al. [2]. Passive techniques, such as the classical tuned mass damper (see Den Hartog [3] for a description relating to linear vibration) have been extended to nonlinear systems with good results [4]. Passive solutions are often preferred in practice as they can be built into the system and there is no control element, which eliminates any issues with stability or robustness. However, for a growing class of structures for which reduced weight and flexibility are important features, passive redesign is not an effective design solution. Active control systems are becoming desirable, since they do not require the increase of mass due to dampeners, stiffeners and absorbers; moreover they can adapt to disturbances deterministically unknown, varying with time or even to large amplitude phenomena. Examples of practical active vibration control systems include active engine supports in vehicles and active vibration isolation systems for propeller aircraft. Depending on the problem, increasing stiffness and damping or isolating the structure are the most common solutions to obtain

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a vibration reduction. Stiffening the structure consists of shifting the resonance frequency beyond or over the frequency band of excitation. Increasing damping consists of reducing the resonance peaks by dissipating the vibration energy. Isolation consists of preventing the propagation of disturbances to sensitive parts of the systems.Several techniques are well known to increase the damping of a structure (passive vibration control) with fluid dampers, eddy currents, elastomers or hysteretic elements, or by transferring kinetic energy to dynamic vibration absorbers [3].

Another common method is the use of transducers as energy converters that transform vibration energy into electrical energy that is dissipated in electrical networks, or stored (energy harvesting). Formerly, semi-active devices,also called semi-passive, (passive devices with controllable properties) have been used. The magneto-rheological fluid damper and piezoelectric transducers with switched electrical networks are examples.

Such techniques have very high rate of effectiveness, but when high performance is needed active control is the best choice. Active vibration control involves a set of sensors, a set of actuators and a control algorithm; the design of the system implies many issues such as the sensors and actuators configuration and how to assure stability and robustness. The power requirement is crucial to define the size of the actuators and the cost of the setup.

Compared to other control problems, structural control has a number of specific features:

1. the systems under investigation generally have a large number of degrees of freedom (DOFs) and a large number of modes;
2. in general attention must be paid to the fact that the high-frequency modes outside the frequency band of interest influence the position of the open-loop zeros of the system;
3. many structures involved in structural control are lightly damped ($\zeta \sim 0.001$ to 0.05).

The latter point means that the stability margin of the uncontrolled modes is small, sometimes very small, and that they are subject to spillover, which means that the control system always tends to destabilise the flexible modes just outside the control bandwidth: the only margin against spillover instability is provided by damping of the residual modes. The combination of a large number of modes with a small stability margin calls for specific control strategies emphasising robustness, with respect to the residual dynamics (high-frequency modes) and also with respect to the changes in the system parameters. Control systems with collocated (dual) actuator/sensor pairs exhibit special properties which are especially attractive in this respect. The recent results regarding the forced nonlinear vibrations of composite sandwich plates, the newest and up-to-date technologies developed in the field of active control systems and the new smart materials structures allow to perform an innovative research. In Ref. [5,6] it is described the vibration characterisation of a carbon–epoxy/honeycomb panel with free edges, in linear and nonlinear field. In such articles a complete modal analysis, obtained by advanced techniques, can be found together with a nonlinear study of damping. The damping values of composite structures undergoing large amplitude vibrations are not easily predictable, therefore they could result a problem for the effectiveness and stability of a control technique.

Free boundary condition is used in this work because is an interesting application for/to active control algorithms, since it reduces the influence of temperature, assembly and non-ideal boundary conditions, but at the same time includes rigid modes of vibration.

Piezoelectric technology is today dominant in the realisation of sensors and actuators for the control of mechanical vibrations, especially in the case of thin-walled shells and plates; it features lightweight elements and good frequency characteristics, the drawback is a limited actuation force. The advances in real-time embedded controllers, versatile and with extremely high performances make affordable the application of a modal Positive Position Feedback by means of piezoelectric actuators affordable. This has in fact proven effective and promising in the case of continuous aerospace structures, with respect to traditional negative position feedback and feedforward (be it adaptive or not), see Kumar [7], Carra et al. [8], Zilletti et al. [9].

The technique of Positive Position Feedback was extensively studied by Kwak [10] and Friswell [11]. The PPF is particularly effective if focused on chosen frequencies and modes, although, in Ref. [12] it is shown how the operation of one actuator can be extended to the control of more than one mode, and has good characteristics against spillover if it is correctly applied. Moreover, the PPF is easy to design if damping ratios are well known and it can be extended to the nonlinear field. The effect of PPF active control is to add a "*virtual damping*" to the system at the desired targeted frequency.

It is therefore goal of this study to obtain both a broadband vibration reduction, an example is the MPPF proposed by Nima Mahmoodi et al. [13], and the suppression of single or multi vibration modes, see Omidi and Mahmoodi [14]. The application to carbon-epoxy/honeycomb panels with free edges seems interesting from an industrial point of view. This is especially true if the applied sensors and actuators are inexpensive, lightweight and easily bondable. The optimal positioning of the piezoelectric patches can moreover be determined by a common finite element analysis; strains of the composite structure can actually suggest where to put the transducers used for the modal control. Fanson and Caughy [15] proposed the PPF control method based on the modal displacement signal, where the controller is very effective in suppressing specific vibration modes, thus, maximising damping in target frequency band without destabilising other modes.

2. Positive Position Feedback

In the single DOF case, to the one degree of freedom vibrating system a compensator is added, sharing a second order equation – in modal coordinates. The two equations of system and compensator are:

$$\ddot{\xi} + 2\varsigma\omega\dot{\xi} + \omega^2\xi = g\omega^2\eta \quad (1a)$$

$$\ddot{\eta} + 2\varsigma_f\omega_f\dot{\eta} + \omega_f^2\eta = \omega_f^2\xi \quad (1b)$$

$\xi,\ \zeta,\ \omega$ are modal displacement, modal damping ratio and natural frequency of the structure respectively $\eta,\ \zeta_f,\ \omega_f$ are the degree of freedom (e.g. force), damping ratio and natural frequency of the compensator, while g is the gain of the compensator. The global system is made of two coupled second order systems and it is absolutely stable for values of gains $g < 1$ [10,16]. The compensator implies active damping if η is -90° out of phase with respect to ξ, at ω. For lower frequencies it adds active flexibility and for higher frequencies active stiffness. Therefore the compensator is usually tuned at $\omega_f = \omega$, in order to have the desired mitigation of vibration amplitude.

$$\mathrm{H(s)} = \frac{\omega_\mathrm{f}^2}{\mathrm{s}^2 + 2\zeta_\mathrm{f}\omega_\mathrm{f}\mathrm{s} + \omega_\mathrm{f}^2} \quad (2)$$

At frequencies above ω_f the slope of the transfer function amplitude is negative and very steep (-40 dB/octave), so it reduces the problem of spillover for vibrations at higher frequencies. On the contrary, at frequencies below ω_f the bode diagram of H(s) has an amplitude of 0 dB, which can cause spillover, that implies to the undesired effect of control on modes outside the frequency band