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Vibration reduction of a woven composite fan blade by piezoelectric shunted devices

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Abstract. This study concerns the vibration reduction in the low frequency range of a composite fan blade of a turbojet engine with piezoelectric devices. The interest is to increase lifespan and avoid flutter phenomena by reducing the vibration amplitude. The solution considered in the work consists in using piezoelectric elements connected to a passive electric circuit usually called shunt. The use of woven composite materials for fan blades enables to plan on embedding piezoelectric materials, for instance in the form of patches inserted between the composite and the coating material. The work presented during this conference will illustrate the feasibility of a piezoelectric shunted device integrated in an industrial application that doesn't require electrical supply. For such a structure, it is shown that a purely passive resonant shunt can significantly reduce the level of vibration of the second bending mode and that a good correlation between experiments and simulations validates the best fitting finite element model.

1. Introduction

This study concerns the vibration reduction in the low frequency range of a composite fan blade of a turbojet engine with piezoelectric devices. The interest is to increase lifespan and avoid flutter phenomena by reducing the vibration amplitude.

Piezoelectric damping devices have been first studied by Hagood and Von Flotow in 1991 [1]. The two electrodes of the piezoelectric element are shunted with an electrical impedance. The aim of the electrical components is to dissipate, by the Joule effect, the electrical energy converted by the piezoelectric material, from the mechanical energy provided by the structure's vibrations.

Several circuits can be connected to the piezoelectric device. A circuit involving passive components is of great interest for its simplicity, its stability and the fact that it doesn't require electrical supply. The resistive shunt is the simplest one but has a limited efficiency. The resonant shunt leads to a higher vibration reduction when it is tuned to the mode to damp, due to the electrical resonance generated by the use of the inductor coupled with the piezoelectric capacitance. Other shunt techniques, such as switches, require electrical energy, thus are not totally passive systems [2].

Many recent studies using piezoelectric shunt damping techniques applied on turbomachinery fan blade and bladed disk, aim to reduce structural vibrations. For instance, Sénéchal [3] works on a titanium alloy fan blade where a mosaic of 5 mm thick piezoceramic patches are bonded on the surface. Delpero and Bachmann [4] propose the integration of piezoelectric transducers in laminated composite fan blade subscale prototypes. Bachmann works on



embedding piezoceramic devices while Delpero studies the integration of a piezocomposite patch between layers of the laminated composite. Choi [5] tests various piezocomposite patches with different material properties to identify the best performing piezoelectric transducer. Then, the selected piezocomposite is applied to a laminated composite fan blade and experimental vibration reduction tests are performed at several rotation speeds. Zhou [6] focuses on piezoelectric shunt damping applied into mistuned bladed disks with several piezoceramic patches. The transducers are fixed onto the disk surface between adjacent blades. Consequently, the blade vibration can be significantly reduced due to the blade-disk coupling. Mokrani [7] has lately considered the resonant shunt damping of rotationally periodic structures using an array of piezoelectric patches, with an application to a bladed disk representative of those used in turbomachinery. He mentions the possibility of using a purely passive resonant shunt since the required inductance for the resonant shunt tuned on the target mode are low. However the author doesn't detail the value of the resistance needed in the shunt, which can be lower than the resistance of purely passive inductors.

The aim of this paper is to develop an efficient piezoelectric shunted device that doesn't require electrical supply to reduce vibrations of a turbojet fan blade. The piezoelectric system is integrated below the protective coating of the woven fan blade, after the resin transfer molding (RTM) process. Even if the resonant shunt is more sensitive to detuning than the resistive shunt, the resistive shunt requires much more piezoelectric materials embedded in a structure than a resonant shunt to obtain equivalent performance. Thus this study focuses on the resonant shunt only. The advantage of the investigated approach is that the piezoelectric damping system is both efficient, integrated and purely passive. According to the authors knowledge, this is the first attempt of piezoelectric damping meeting those requirements for a complex structure.

2. Experimental setup

Three piezoelectric coupling effects can be involved in damping applications. In this study, the 3-1 coupling effect of the piezoelectric elements is used to reduce vibrations. In this case, the expansion or contraction of the transducers is perpendicular to the electric field as shown figure 1, with E_3 the component of the electric field in the transverse direction and ε the longitudinal strain.

Throughout the article, frequencies are shown in an adimensional unit system to preserve confidentiality of the composite structure.

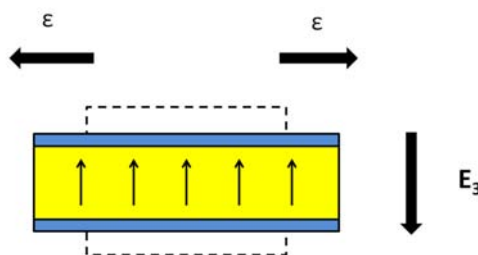


Figure 1: 3-1 piezoelectric coupling effect

To improve the damping level, a key issue is to maximize the modal electromechanical coupling factor [8], which characterizes the energy exchanges between the mechanical structure and the piezoelectric patches for a given mode.

The structure studied is a woven carbon-epoxy composite fan blade. Contrary to the applications with titanium alloy [3][6][7], for which the electric potential of the structure is

the same as the potential of one of the two electrodes of the piezoelectric element, the composite material is highly resistive and can be considered as an electrical insulator. Thus, the insulating property involves the use of piezoelectric transducers with returned electrodes in order to easily reach the electrodes of the piezoelectric elements. Otherwise, if piezoelectric devices without returned electrodes are involved, electric wires must be placed between the structure and the patch during the bonding step. The issue is that the bonding is thus degraded, reducing the electromechanical coupling.

The glue properties and its thickness influence the electromechanical coupling. In order to maximize the electromechanical coupling, the glue thickness must be minimized and its shear modulus must be high. The viscoelastic property must also be taken into account for the selection of the glue. Considering those criteria, an epoxy glue is chosen because of its high mechanical properties and because its behaviour can be considered elastic in the low frequency range.

The coupling also depends on the piezoelectric material used. Thus, the piezoceramic requires an high d_{31} piezoelectric coefficient value since the 3-1 coupling effect is involved to convert mechanical energy into electrical energy.

Because of the complex shape of the blade and its stiffness and mass properties, one patch is not enough to damp the structural vibrations in the low frequency band. Thus, a mosaic of patches is considered in the damping technique and the piezoelectric devices are connected in parallel in order to minimize the inductance value of the shunt. To obtain a sufficient reduction level on the second bending mode while reducing the piezoelectric device dimensions, 9 PZT piezoceramic patches ($50 \times 30 \times 0.2$ mm) are applied to the fan blade in figure 2 and figure 3. Then, in order to increase the electrical energy, the patch mosaic must be located in area where the strains are maximum for the target mode, which is the second bending mode in this study. Thus, a modal analysis of the blade without piezoelectric patches is performed to determine this area. This area changes with the rotation speed but in this study the patches location is chosen at zero spin speed.



Figure 2: Patches mounted on the intrados

The experimental setup of the clamped fan blade is presented in figure 4. The excitation device is a non-contact system involving a coil and a magnet stuck on the blade. The magnet's mass (about 4 g) is negligible compared to the structure's mass and has no influence on the structure's dynamic. The velocity is measured with a laser Doppler vibrometer. The frequency response functions are measured at the tip on the extrados and the magnet is stuck on the other blade side (see figure 5). The reference signal is the current in the coil, which can be considered proportional to the force applied on the blade in the frequency band studied [8]. The signal used is a white noise signal because it enables to excite all the modes of the clamped blade.

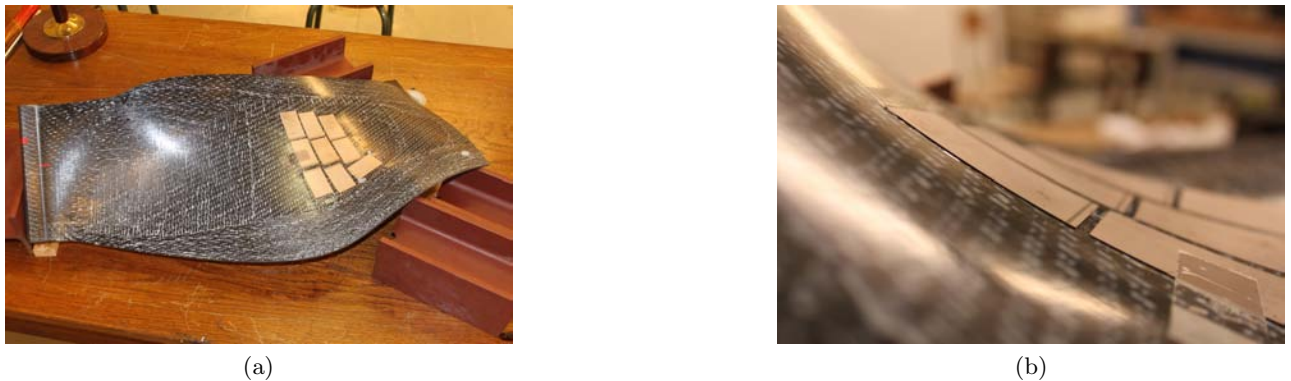


Figure 3: Blade with patches (a) General view (b) Zoom on the surface

The inductor is specially designed for the application and enables to have a purely passive resonant shunt thus avoiding the use of synthetic inductors contrary to the other studies concerning similar applications. To define the optimal shunt parameters, the method proposed by Thomas et al. [8] is considered, which gives

$$L = \frac{1}{C\omega_{OC}^2} \quad R = \sqrt{\frac{3}{2}} \frac{k_c}{C\omega_{OC}} \quad k_c = \sqrt{\frac{\omega_{OC}^2 - \omega_{SC}^2}{\omega_{SC}^2}} \quad (1)$$

where C is the blocked capacitance, ω_{OC} and ω_{SC} are the open and short circuit angular frequencies of the mode to control and k_c is the coupling factor. The inductor involves a ferrite core that enables to reach high inductance value [9].

To tune the resonant shunt on the second bending mode, the inductance required is 0.6 H (see figure 6) and the resistance value is 40 Ω . The resistance needed to tune the resonant shunt on the second mode is low and corresponds to the value of intern resistance of the inductor. So there is no need of an additional resistor, the resistance being included into the inductor.

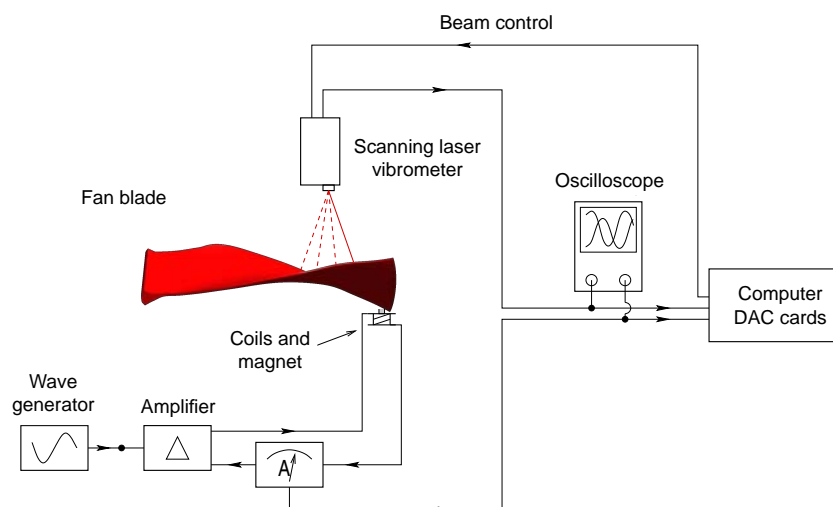


Figure 4: Experimental setup scheme

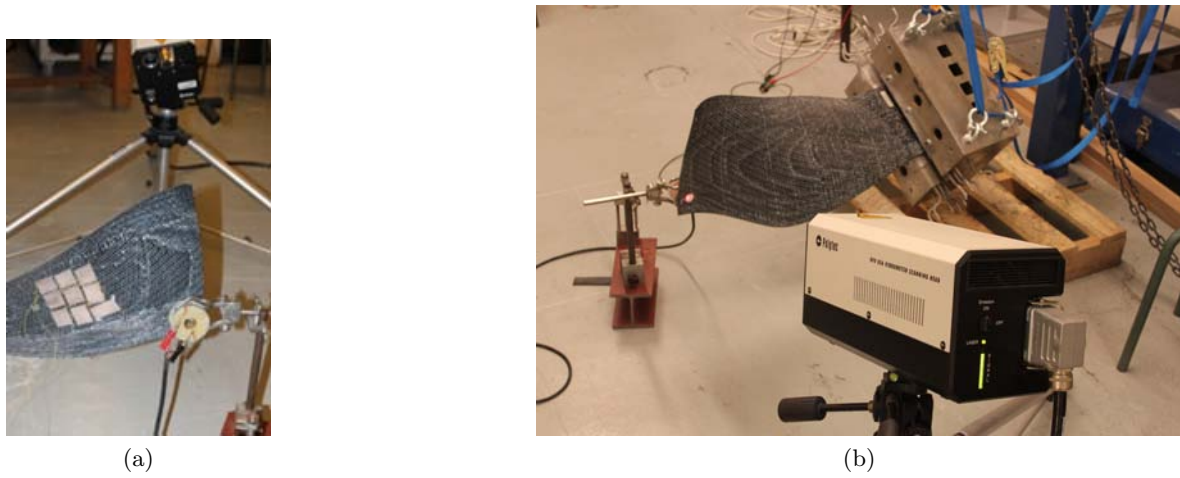


Figure 5: Experimental setup (a) intrados (b) extrados



Figure 6: Inductor of the resonant shunt for the second bending mode

3. Finite element formulation of an elastic structure with piezoelectric elements

In this section, the general formulation of the equations that govern the mechanical and electrical state of an elastic structure equipped with piezoelectric patches is used to derive a finite element model as presented in [10].

After discretization of the variational formulation by finite element method, we find the following matrix equation:

$$\left(-\omega^2 \begin{bmatrix} \mathbf{M}_u & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} + \begin{bmatrix} \mathbf{K}_u & \mathbf{K}_{u\psi} \\ -\mathbf{K}_{u\psi}^T & \mathbf{K}_\psi \end{bmatrix} \right) \begin{Bmatrix} \mathbf{U} \\ \mathbf{\Psi} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F} \\ \mathbf{Q} \end{Bmatrix}, \quad (2)$$

where ω is the angular frequency, $\{\mathbf{U}\}$ is the column vector of nodal values of u_i , of length N_m (number of mechanical dof), $[\mathbf{M}_u]$ and $[\mathbf{K}_u]$ are the mass and stiffness matrices, of size $N_m \times N_m$, and $\{\mathbf{F}\}$ is the column vector of mechanical forces, of length N_m . Moreover, $\{\mathbf{Q}\}$ and $\{\mathbf{\Psi}\}$ are the column vectors of electric charges density and potential, of length N_{el} (number of electrical dof that corresponds to the number of nodes contained in the piezoelectric elements), $[\mathbf{K}_{u\psi}]$ is the electromechanical coupling matrix, of size $N_m \times N_{el}$, and $[\mathbf{K}_\psi]$ is the dielectric matrix, of size $N_{el} \times N_{el}$.

We consider now the case where the piezoelectric elements have the shape of a slightly curved

shell such as a piezoceramic patch, with its upper and lower surfaces covered with a very thin layer of conducting material to obtain electrodes. The equipotentiality condition on each patch electrode surface is introduced by assigning a single dof for voltage on all nodes of each electrode surface. Note that for the p -th piezoelectric patch, $\{\Psi^{(p)}\}$ can be defined such as:

$$\{\Psi^{(p)}\} = [\psi_+^{(p)} \quad \Psi_{int}^{(p)} \quad \psi_-^{(p)}]^T \quad (3)$$

where $\psi_+^{(p)}$ and $\psi_-^{(p)}$ denote the upper and lower electric potential respectively and $\{\Psi_{int}^{(p)}\}$ denotes the column vector of the electric potential in all nodes of the p -th patch except those of the upper and lower surfaces. Then, to reduce the problem size, condensation method is applied to those internal electric potential.

The discretized formulation equation (2) is particularly adapted when the piezoelectric patches are shunted, that is to say, connected to an electrical network. In this case, neither the potential difference $V^{(p)}$ between upper and lower electrodes nor the electric charge $Q^{(p)}$ contained in the electrodes surfaces are prescribed by the electrical network [8] but the latter imposes only a relation between them (see equation 4).

$$V^{(p)} + L\ddot{Q}^{(p)} + R\dot{Q}^{(p)} = 0 \quad (4)$$

where L and R are the inductance and resistance values of the resonant shunt.

A finite element model of the woven fan blade with piezoelectric patches on the surface is presented figure 7. The dovetail nodes are set to zero to take into account of the experimental clamping. The glue isn't modeled because its mechanical properties and its thickness are assumed to be negligible.

The finite elements tested are linear and quadratic hexahedral elements for both classical and piezoelectric elements. The piezoelectric elements nodes have four degrees of freedom: the displacements in the three space directions and the electric potential. Several meshes are achieved, with linear or quadratic hexahedral elements and with a different number of elements through thickness. The comparison of the different models with the experimental test aims to determine the most suitable model comparing accuracy and calculation cost.

For the frequency response functions, a modal projection approach is used considering experimentally identified modal damping. The modal basis is truncated to the first fifteen modes of the blade and the harmonic forced vibration response is achieved for three electric shunts: open circuit, short circuit and resonant shunt tuned on the second bending mode.



Figure 7: Experimental setup and finite element model

Table 1: Non-dimensional frequencies and coupling factor for the various models

Modes	Mode type	Elastic		Short-circuit		Open-circuit		Coupling coeff.	
		H08	H20	H08	H20	H08	H20	H08	H20
1	1B	0.36980	0.37060	0.36980	0.37060	0.36982	0.37061	0.85%	0.83%
2	2B	1.00000	1.00000	1.00000	1.00000	1.00015	1.00015	1.74%	1.71%
3	1T	1.30428	1.30681	1.30428	1.30681	1.30428	1.30682	0.32%	0.30%
4	3B	2.54418	2.53807	2.54418	2.53807	2.54422	2.53810	0.56%	0.51%

4. Numerical and experiment results

The fan blade tested in this project is a prototype without the protective coating. The vibration reduction results from numerical simulation and experiment are presented figure 8, for the model with quadratic elements and one element through the patches thickness and are compared with the experiment results. A vibration reduction of 16 dB for the second bending mode is observed for both models and for the experiment. Those attenuation results are very high taking into account the dimensions and mass of the damping device.

The slight difference between the frequency response functions of models and the experiment validate the hypotheses made on the numerical models through the convergence study. For the linear elements models with one and three piezoelectric elements through the patches thickness the required resistance is 30 Ω and the inductance needed is 1.04 H, thus the resonant shunt parameters are the same and do not vary with the number of elements through thickness. The conclusion is identical with the quadratic elements, the models with one and three elements required the same resistance and inductance values which is 30 Ω for the resistance and 1.07 H for the inductance. However, as expected, the inductance values vary for the two types of elements tested since the eigenvalues are distinct. Thus, the inductance value is 1.04 H when linear elements are used and 1.07 H with quadratic elements. The inductance required for the quadratic model is higher than for the linear model since the eigenfrequency of the second mode is lower with a quadratic model than with a linear elements model.

The meshes constituted of linear hexahedral elements and with only one element through the patch thickness assumes that the electric potential is linear through thickness. Consequently, the induced potential is neglected in this case but this assumption is correct here since the patches are thin. However, the model with quadratic elements and one element through the patches thickness better fits with the experiment that is why this model is more suitable than the others.

The determination of the effective electromechanical coupling factor k_c shows that it converges with the increase of the elements number through the patches thickness. The values of the modal coupling coefficient, about 1.7% for all the configurations, shows that a damping device can be efficient even if the coupling coefficient is low.

In order to illustrate our results, Table 1 shows the non-dimensional frequencies and coupling factors for the four first modes considering respectively one linear or one quadratic element in the patches thickness. It is noticed that the coupling coefficient converges and it is verified that the localisation chosen for the patches is well suited to damp the second bending mode.

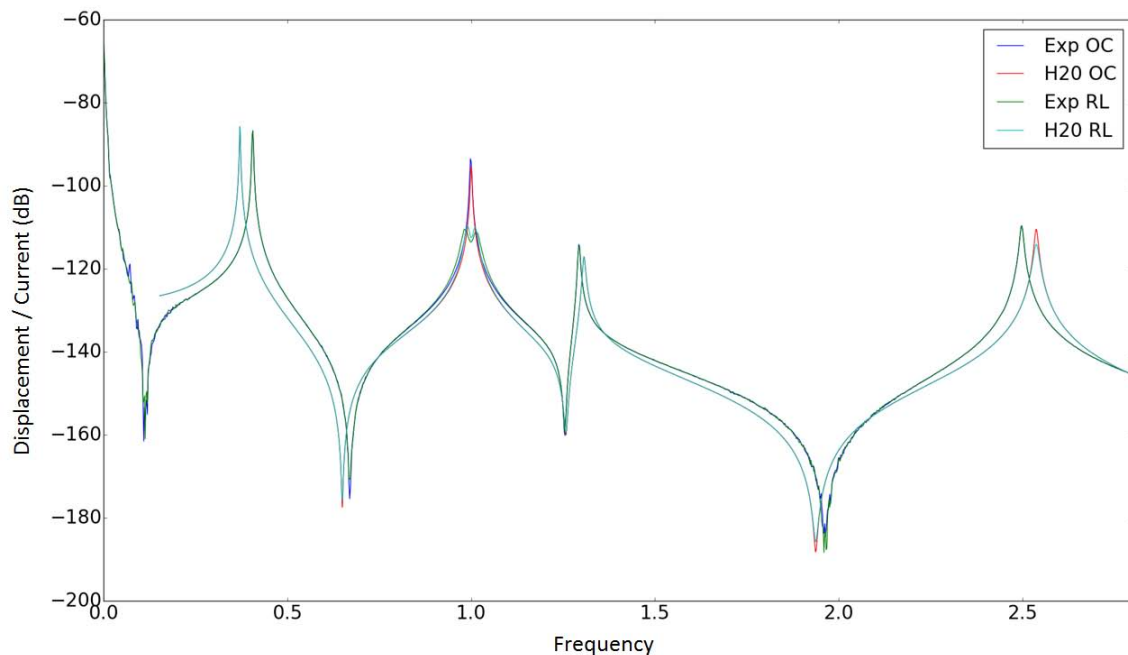


Figure 8: Comparison between the experiment and the model with quadratic elements and one element through the patches thickness

5. Conclusion

In this study, experimental vibration reduction tests are performed with a lightweight piezoelectric shunted device to damp the second bending mode of a turbojet fan blade. Neglecting the influence of the rotating speed, the system significantly reduces the vibration amplitudes (about 16 dB) with a purely passive resonant shunt. The fan blade tested in this project is a prototype without the protective coating, which will be added in the final configuration in order to protect the device and to smooth the surface for the aerodynamic performance. Different numerical models have also been tested in this work and comparisons with experimental results enable to select the best fitting finite element model (with quadratic shape functions). The chosen model will be used in the future to simulate several configurations of piezoelectric patches (dimensions and locations) in order to maximize the vibration reduction of various target modes.

6. Acknowledgement

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