Vibration prediction of rotating composite fan blades comprising viscoelastic damping treatments

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Abstract

To reduce the risk of fatigue problems and flutter phenomena, the structural damping of turbomachinery components is increased by integrating constrained viscoelastic patches into rotating composite blades. However, the temperature, the frequency (of the vibration mode to be damped) and the rotation speed are all parameters which may affect significantly the efficiency of the viscoelastic damping treatment.

In the presented work, a numerical tool is developed to predict the structural damping of a dynamic rotating composite fan blade while varying the temperature and the rotation speed. This study is based on a finite element model of a fan disk sector equipped with constrained viscoelastic patches. This work will allow quantifying the influence of temperature and rotation speed on the efficiency of the passive damping treatment and contribute to the design of higher performing blades.

Key words: Rotating blade, Finite element method, Viscoelastic damping

1. Introduction

Advanced aircraft engines have to meet the increasing environmental regulatory requirements, leading to lightweight structures, which are more suscep-

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tible to excessive vibrations. To reduce the risk of fatigue problems and flutter phenomena, the structural damping of turbomachinery components has to be increased. This study concerns the vibration reduction in the low frequency range of a rotating composite fan blade of a turbojet engine by embedding constrained viscoelastic patches into the host structure. This passive damping treatment relies on the characteristics of viscoelastic materials which allow energy dissipation. It has been successfully applied to many industrial applications over the last decades.

One of the challenges in the integration of constrained viscoelastic patches into rotating composite blades is the frequency- and temperature-dependence of the mechanical properties of the viscoelastic material. Moreover, the amount of structural damping provided by the patches depends on the shear strain levels in the damping layer. Since the centrifugal force acting on the blade generates rotation-prestress, causing stiffening effects, the structure deforms less and so does the viscoelastic layer. Therefore, the temperature, the frequency (of the vibration mode to be damped) and the rotation speed are all parameters which may affect significantly the efficiency of the damping treatment. The design of higher performing blades requires a numerical tool to predict their dynamic behaviour and estimate the influence of temperature and rotation speed on modal damping factors.

In the prediction of the dynamic behaviour of rotating composite blades, many papers can be found in the the literature [1, 2]. However, a limited number of studies concerns the dynamic behaviour of rotating structures with constrained viscoelastic layers. Most of them are limited to rotating plates [3, 4] or beams [5, 6], which do not account for the complex geometry and structure of current composite turbine blades [7]. In addition, the mechanical properties of the damping layer are often considered as constant [3, 4] even though the modulus of viscoelastic materials is known to be strongly frequency-dependent. In the aforementioned literature, parametric studied are performed to investigate the influence of the properties and dimensions of the constrained viscoelastic layer, and of the rotation speed on the modal damping of the rotating structure. The

influence of temperature seems to have never been studied, while fan blades are subjected to significant variations of temperature in operating conditions.

In the presented work, a numerical tool is developed to predict the structural damping of a dynamic rotating composite fan blade while varying both the temperature and the rotation speed. This study is based on a 3D finite element model of a fan disk sector equipped with constrained viscoelastic patches. The first goal of this study is to investigate the efficiency of the passive treatment in operation. The fan blade under study is on the outside of the engine, so it will undergo large temperature variations depending on the region were the aircraft is operating. Additionnally, during a flight cycle, the fan blade will rotate at different angular speeds. The frequency- and the temperature-dependency of the viscoelastic material's properties is taken into account to evaluated the efficiency of the damping treatment at each operating point. The second goal of this work is to study the influence of the material selection on the efficiency of the damping treatment and to investigate the potential advantage of combining different viscoelastic materials.

The next section describes the finite element model of the blade disk sector, and the dynamic modelling of the rotating structure with embedded viscoelatic patches when subjected to harmonic excitation. Results are presented and discussed in the third section, and conclusions are given in the last section.

2. Numerical model

2.1. Description of the finite element model

The structure under study is an assembly of a disk sector and a 3D woven carbon epoxy composite fan blade from the LEAP engine. Because of the complex geometry and the 3D-woven architecture, the structure is meshed by 3D finite elements (Figure 1). Each finite element of the blade has its own local coordinate system and is associated to a homogenised material behaviour, to account for the anisotropy of the woven composite [8].

The friction contact at the interface between the disk and the blade [9] is not

taken into account in this work. Instead, rigid elements are added into the finite element model to ensure the permanent contact between the disk and the blade. Two viscoelastic patches have been designed to mainly damp the first mode of vibration, which is the most critical for flutter. They are positioned on top of the extrado wall of the blade [7] (see Figure 1). Each viscoelastic patch is composed of a 0.3 mm thick viscoelastic layer and a 0.6 mm thick composite constraining layer. Two viscoelastic materials are used : material A for the viscoelastic patch 1 and material B for the viscoelastic patch 2. Both materials are thermo-rheologically simple, which means they respect the time-temperature superposition principle. This principle supposes an equivalence between time (or frequency) and temperature effects : viscoelastic properties measured at one temperature can be shifted along the time (or frequency) axis to obtain the viscoelastic properties at another temperature. For both materials, the time-temperature superposition principle was applied to DMA measurements to obtain the storage modulus and the loss factor on the frequency range of investigation at any temperature comprised between -40^oC and 70^oC. The temperature-dependent viscoelastic properties are plotted at two different frequencies (corresponding to the first and the fifth vibration mode of the structure) in Figure 2. A complex, frequency- and temperature-dependent shear modulus $G^*(f,T)$, and a constant Poisson ratio are considered in this work for each viscoelastic material. To properly account for the shear behaviour in the damping layer, 20-node hexaedra are used to mesh the whole structure, leading to a finite element model of 838 000 degrees of freedom.

Finally, cyclic symmetry boundary conditions are applied to the fan disk sector [10] to account for the cyclo-symmetry geometry of the fan blade assembly. This study is focused on the zero-order harmonics so that the cyclic symmetry boundary conditions resume to symmetric boundary conditions, applied on the two sides of the disk connected to adjacent disk sectors. The structure is harmonically-excited by a point load on the blade, indicated on Figure 1.(b). This load point has been chosen to excite all the vibration modes on the frequency range of interest.

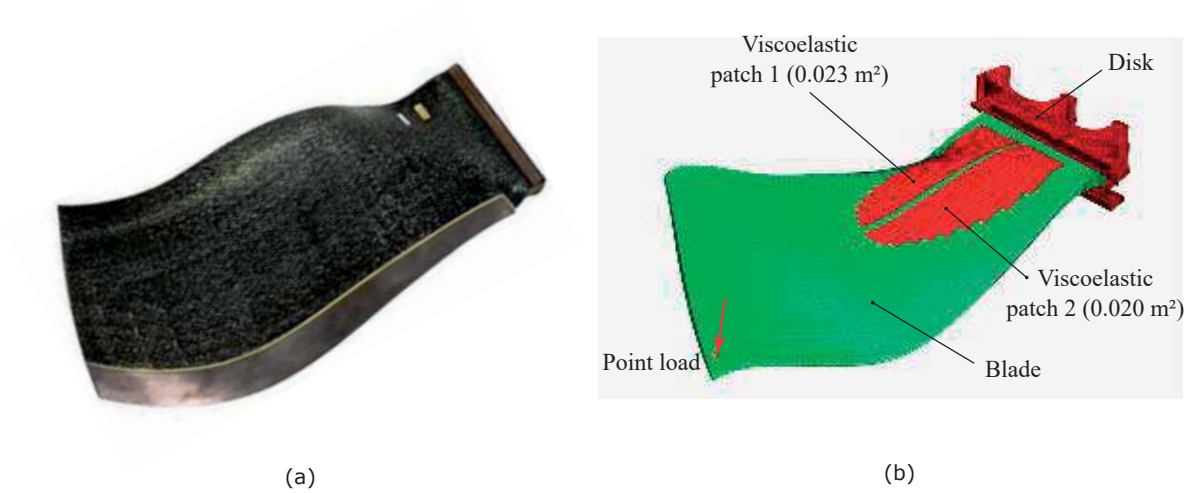

Figure 1: (a) 3D woven carbon epoxy composite fan blade and (b) finite element model of the fan disk sector with two viscoelastic patches.

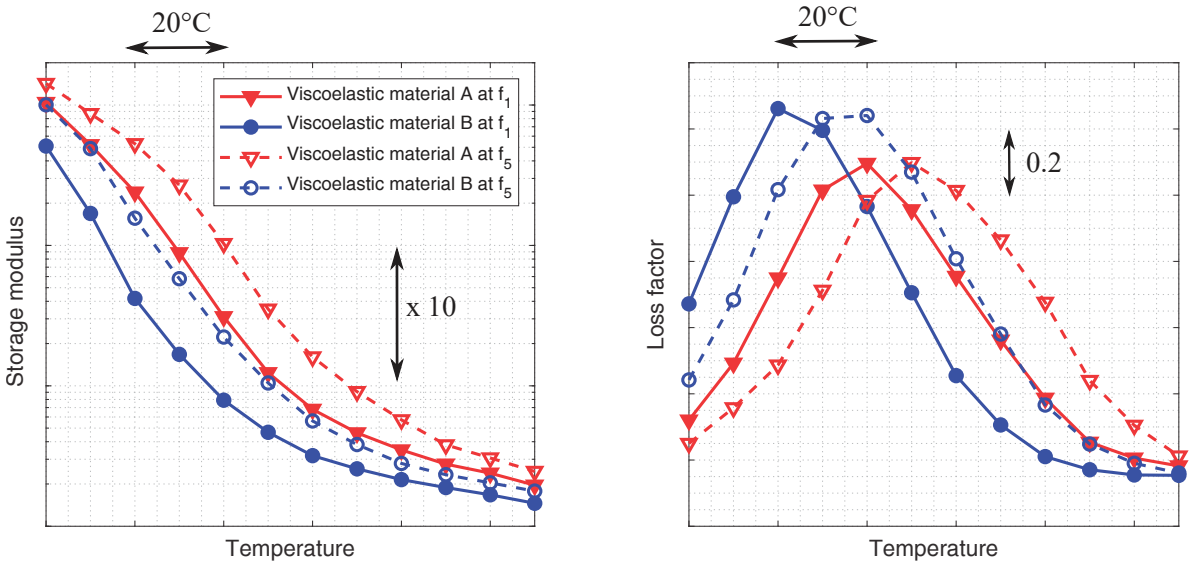

Figure 2: Temperature-dependent properties of viscoelastic materials A and B at two frequencies.

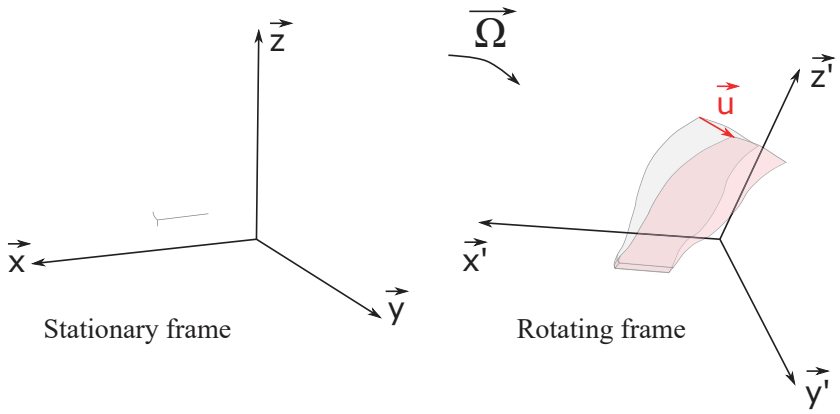

Figure 3: Stationary and rotating reference frame.

2.2. Dynamic modelling

Let us consider that the flexible fan blade is rotating at an arbitrary rotation $\mathbf{\Omega} = \begin{bmatrix} \Omega_x & \Omega_y & \Omega_z \end{bmatrix}^T$ with respect to the stationary frame (see Figure 3). The chosen reference configuration, in which the equations of motion are derived, is the rotating frame. In this reference configuration, the displacement between the undeformed and the deformed configuration is denoted $\mathbf{u}(\mathbf{x}, t)$, where $\mathbf{x}$ is the coordinate vector of the undeformed structure in the reference configuration.

Using the finite element method, the displacement field can be approximated by:

$$\mathbf{u}(\mathbf{x}, t) = \mathbf{N}(\mathbf{x})\mathbf{U}(t) \tag{1}$$

where $\mathbf{N}(\mathbf{x})$ is the matrix of shape functions and $\mathbf{U}(t)$ is the nodal displacement vector. This leads to the following approximated strain field:

$$\boldsymbol{\varepsilon}(\mathbf{x}, t) = \frac{1}{2}\left(\nabla\mathbf{u}(\mathbf{x}, t) + (\nabla\mathbf{u}(\mathbf{x}, t))^T\right) = \mathbf{B}(\mathbf{x})\mathbf{U}(t) \tag{2}$$

where ∇ is the gradient operator and $\mathbf{B}(\mathbf{x})$ is the discretized gradient matrix. After discretization, the equations of motion of an elastic structure in rotation are:

$$\mathbf{M}\ddot{\mathbf{U}} + \mathbf{G}\dot{\mathbf{U}} + (\mathbf{K} - \mathbf{K}_c)\,\mathbf{U} = \mathbf{F}_\Omega + \mathbf{F}_{ex} \tag{3}$$

where:

- $\mathbf{M} = \int_V \rho\mathbf{N}^T\mathbf{N}dV$ is the mass matrix

- $\mathbf{G} = 2\int_V \rho \mathbf{N}^T \mathbf{\Omega} \mathbf{N} dV$ is the gyroscopic matrix
- $\mathbf{K} = \int_V \mathbf{B}^T \mathbf{C} \mathbf{B} dV$ is the stiffness matrix
- $\mathbf{K}_c = -\int_V \rho \mathbf{N}^T \mathbf{\Omega}^2 \mathbf{N} dV$ is the supplementary stiffness matrix
- $\mathbf{F}_\Omega = \int_V \rho \mathbf{N}^T \mathbf{\Omega}^2 \mathbf{x} dV$ is the nodal centrifugal force vector
- $\mathbf{F}_{ex}$ is the nodal external force vector.

with ρ the density, $\mathbf{C}$ the elasticity matrix and $\mathbf{\Omega}$ the rotation speed matrix:

$$\mathbf{\Omega} = \begin{bmatrix} 0 & -\Omega_z & \Omega_y \\ \Omega_z & 0 & -\Omega_x \\ -\Omega_y & \Omega_x & 0 \end{bmatrix} \tag{4}$$

Note that the gyroscopic matrix $\mathbf{G}$, the supplementary stiffness matrix $\mathbf{K}_c$ and the nodal centrifugal force vector $\mathbf{F}_\Omega$ all depend on the rotation speed. Considering the slenderness ratio of the structure and the range of rotation speeds investigated in this work, the effect of the gyroscopic matrix on the dynamic behavior of the rotating blade can be neglected [2, 11].

However, a geometric stiffness matrix depending of the nodal displacement vector, $\mathbf{K}_g(\mathbf{U})$, should be added to account for the centrifugal stiffening effects [12]:

$$\mathbf{M}\ddot{\mathbf{U}} + (\mathbf{K} - \mathbf{K}_c + \mathbf{K}_g(\mathbf{U}))\,\mathbf{U} = \mathbf{F}_\Omega + \mathbf{F}_{ex} \tag{5}$$

The solution of Equation (5) is performed in two steps [13].

The first step consists in solving the following non-linear static problem to determine the non-linear equilibrium position of the structure submitted to centrifugal forces $\mathbf{U}^0$:

$$\left(\mathbf{K} - \mathbf{K}_c + \mathbf{K}_g\left(\mathbf{U}^0\right)\right)\mathbf{U}^0 = \mathbf{F}_\Omega \tag{6}$$

The second step consists in linearising the dynamic problem around the nonlinear static equilibrium position, by assuming the dynamic displacement $\mathbf{U}$ is a

small linear perturbation $\mathbf{U}^*$ around the non-linear static equilibrium position $\mathbf{U}^0$:

$$\mathbf{U} = \mathbf{U}^0 + \mathbf{U}^* \tag{7}$$

Then Equation (5) becomes :

$$\mathbf{M}\ddot{\mathbf{U}}^* + \left(\mathbf{K} - \mathbf{K}_c + \mathbf{K}_g\left(\mathbf{U}^0\right)\right)\mathbf{U}^* = \mathbf{F}_{ex} \tag{8}$$

Under harmonic excitation, this dynamic system can be solved in the frequency domain :

$$\left(\mathbf{K} - \mathbf{K}_c + \mathbf{K}_g(\mathbf{U}^0) - \omega^2\mathbf{M}\right)\mathbf{U}^* = \mathbf{F}_{ex} \tag{9}$$

By incorporating viscoelastic materials into the structure, the stiffness matrix becomes complex, frequency and temperature dependent :

$$\mathbf{K}^*(\omega, T) = \mathbf{K}_e + G_1^*(\omega, T)\mathbf{K}_v^1 + G_2^*(\omega, T)\mathbf{K}_v^2 \tag{10}$$

where $\mathbf{K}_e$ is the stiffness matrix associated to the elastic part of the structure, $\mathbf{K}_v^1$ (resp. $\mathbf{K}_v^2$) is the stiffness matrices associated to the viscoelastic material of the first patch (resp. second patch), and $G_1^*(\omega, T)$ (resp. $G_2^*(\omega, T)$) is the complex frequency and temperature dependent shear modulus of the viscoelastic material of the first patch (resp. second patch).

To sum up, the dynamic response of the blade-disk sector with viscoelastic patches to a harmonic load at a given rotation speed is computed in two steps :

- **Step 1** Computation of the non-linear static equilibrium position of the structure subjected to centrifugal forces $\mathbf{U}^0$:

$$\left(\mathbf{K}^*(0, T) - \mathbf{K}_c + \mathbf{K}_g(\mathbf{U}^0)\right)\mathbf{U}^0 = \mathbf{F}_\Omega \tag{11}$$

- **Step 2** Computation of the dynamic linear perturbation $\mathbf{U}^*$ in the frequency domain:

$$\left(\mathbf{K}^*(\omega, T) - \mathbf{K}_c + \mathbf{K}_g(\mathbf{U}^0) - \omega^2\mathbf{M}\right)\mathbf{U}^* = \mathbf{F}_{ex} \tag{12}$$

Equation (11) can be solved by a Newton-Raphson method [13, 14], or an approximation of $\mathbf{U}^0$ can be computed under small strain assumption [11]. The

latter approach is herein adopted. Considering the size of the finite element model, it is more efficient and has already been validated in a previous study by comparison with the Newton-Raphson approach. The model was built and the calculations made with an in-house software in Python, which has been validated by comparison with several commercial softwares.

To reduce the computational time, Equation (12) is solved by using a multi-modal projection method, which is adapted to structures with frequency-dependent damping [15].

3. Results and discussion

This section discusses the influence of operating conditions on the efficiency of viscoelastic damping treatment. The typical ranges of temperature and angular speed considered by engine manufacturers is $[-40^o\text{C}, +70^o\text{C}]$ and $[0, 2850]$ RPM respectively. Following the dynamic modelling described in the previous section, frequency responses of the rotating blade disk sector with viscoelastic patches are computed at different operating points, under the assumption of stationary heating regime.

3.1. Non-rotating case

In this section, the influence of temperature on the dynamic response of the non-rotating structure with viscoelastic patches is investigated. Figure 4 shows that a variation of temperature does not have a significant influence on the resonance frequencies of the structure. However, it has a strong influence on the amplitude of the dynamic response. Results also evidence a broad temperature range on which the damping treatment is efficient : from -10^oC to 20^oC.

The efficiency of viscoelastic patches depends on a number of parameters, such as the damping ratio of the viscoelastic material at a given temperature and frequency, the thickness ratio or the stiffness ratio between the core layer and

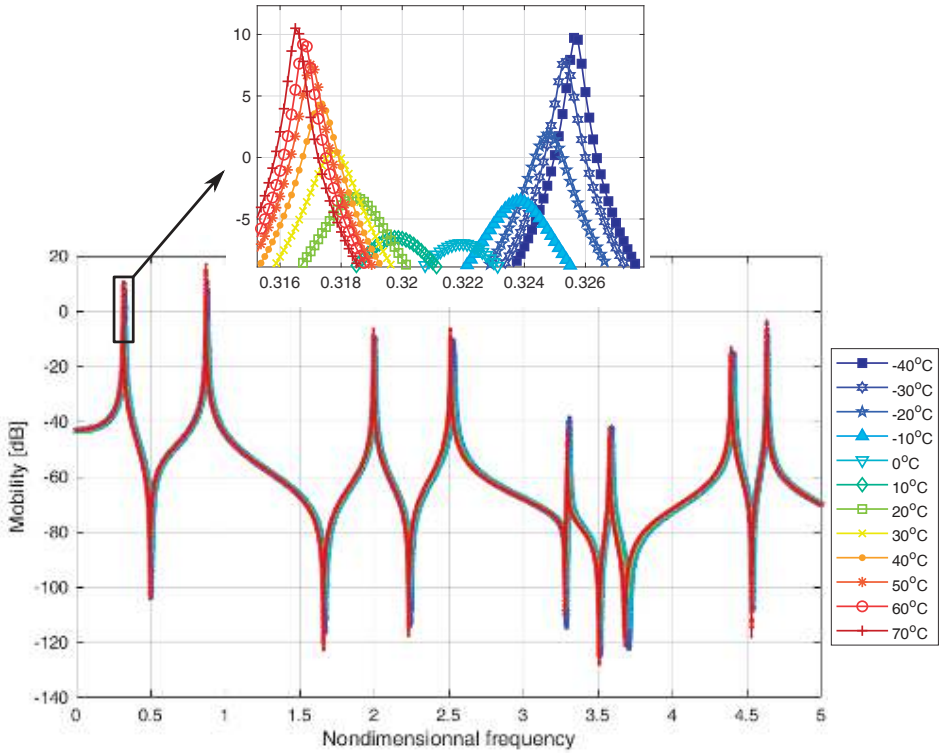

Figure 4: Influence of temperature on the dynamic response of the non-rotating structure with viscoelastic patches.

the elastic faces [16, 17]. In this work, the thickness ratio of both patches are the same, and the constraining layer of both patches are made of the same composite material. However, the viscoelastic material used as core layer in both patches is different. The interest in combining two different viscoelastic materials for the patches will be discussed in Section 3.3.

3.2. Rotating case

In this section, the influence of the rotation speed on the efficiency of the damping treatment is investigated. To that purpose, the modal parameters of the first six modes are extracted from the frequency responses using the LSCF method [18].

The modal parameters (resonance frequencies and modal damping) of the first vibration modes of the non-rotating and the rotating structure (2852 RPM) are plotted as a function of temperature in Figure 5 and 6. Figure 5 evidences the well-known stress-stiffening effect : eigenfrequencies increase with the rotation

speed.

Figure 6 shows that the damping ratios are higher for the first mode of vibration, and more sensitive to a change of temperature or rotation speed. This is due to the fact that the viscoelastic patches have been designed to mainly damp the first vibration mode. For higher vibration modes, the viscoelastic layer undergoes less shear deformation, resulting in smaller damping ratios and less sensitivity to the parameters' variations under consideration. In particular, the damping ratios of the sixth mode are very small and are not affected by the rotation speed as the the sixth mode is a local vibration mode (see Figure 7). Some studies have shown that modal damping ratios generally decrease with increasing rotation speed [4]. This tendency is observed in Figure 6, regardless of the temperature. The decrease in modal damping ratio with increasing rotation speed is due to the stiffening effect, since the damping layer in the viscoelastic patches undergo less deformation. It should be noted that self-heating phenomena in the viscoelastic layers are not taken into account in this study. The amplitude of vibrations for this type of fan blade in operating conditions is considered small enough to neglect the temperature rise from the dissipation-induced self-heating.

3.3. Combining damping treatments

In the two previous sections, the viscoelastic material used as core layer in both patches is different. In this section, the interest in combining various damping materials for the patches is discussed. By comparing the master curves of the viscoelastic materials A and B in Figure 2, one can notice that they have distinct glass-rubber transition regions, characterised by a peak of loss factor and a drop in the storage modulus. The viscoelastic patches may then behave differently in temperature. This is investigated by studying three configurations:

1. **Configuration 1A-2B**: material A for the viscoelastic patch 1 and material B for the viscoelastic patch 2 (see Figure 1)
2. **Configurationn 1A-2A**: material A for both patches

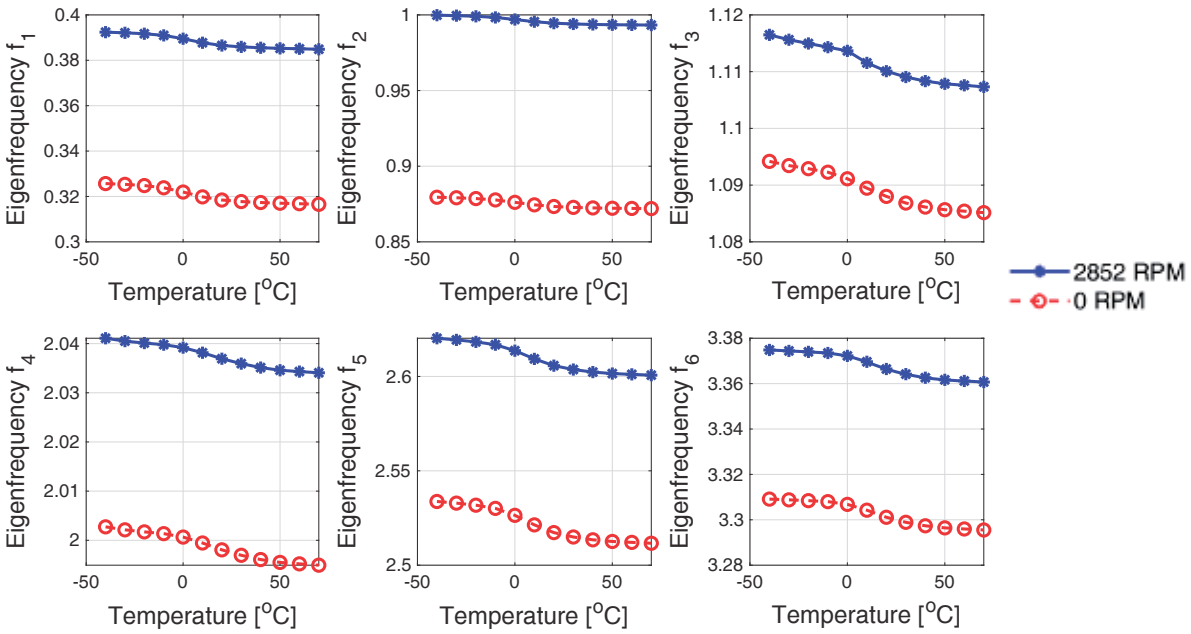

Figure 5: Influence of the rotation speed on the first resonance frequencies of the structure (frequencies are nondimensionnal).

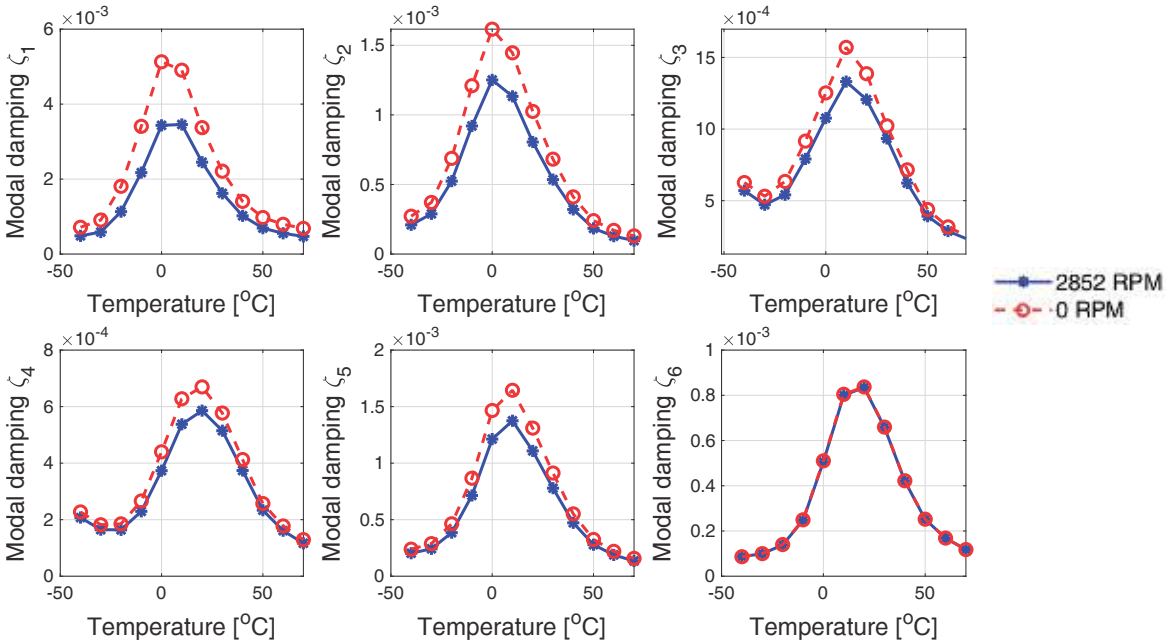

Figure 6: Influence of the rotation speed on the first modal damping ratios of the structure.

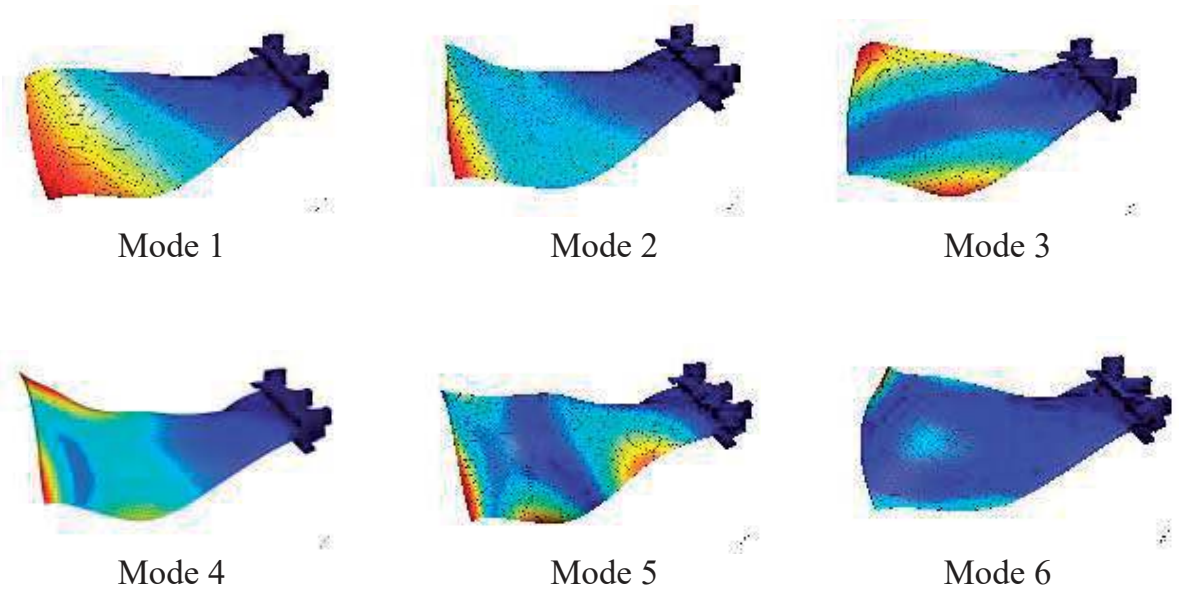

Figure 7: Unity-scaled mode shape of the first six vibration modes.

3. **Configuration 1B-2B**: material B for both patches

Figure 8 compares the modal damping factors of the first five modes extracted from FRF computations at 2852 RPM for the three configurations. Results indicate that material A is most efficient to damp vibration at lower temperatures (around 0^oC for the first vibration mode) while material B is most efficient at higher temperatures (around 15^oC for the first vibration mode). This can be related to the master curves from Figure 2, which indicate that material A has a lower glass transition temperature than material B. More importantly, combining two different damping treatments, as in configuration 1A-2B, allows a tuning of the temperature range of efficiency of the damping treatment. However, it does not seem to broaden the temperature range of efficiency.

4. Conclusion

In this work, a finite element model of a rotating composite fan disk sector with constrained viscoelastic patches is developed to predict the influence of temperature and rotation speed on the efficiency of the damping treatment. The frequency and the temperature dependency of viscoelastic materials are

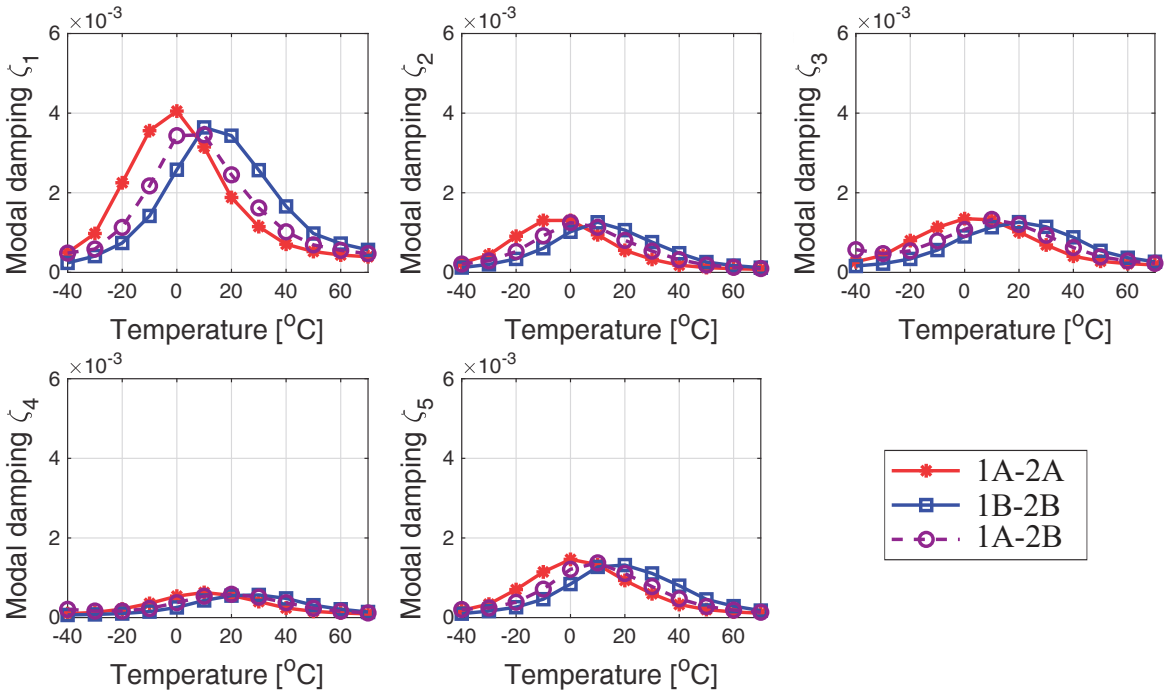

Figure 8: Modal damping ratios of the first five modes of the structure for three configurations. 1A-2A: material A used for both viscoelastic patches, 1B-2B: material A used for both viscoelastic patches, 1A-2B: material A for the viscoelastic patch 1 and material B for the viscoelastic patch 2.

taken into account and the multi-model projection method is used to compute the dynamic response of the rotating structure.

Results from frequency response analyses indicate that both temperature and rotation speed affect the modal damping ratios of the structure. The strong dependency of the mechanical properties of viscoelastic materials naturally implies that the damping treatment is efficient only on a reduced temperature range. This constitutes the main drawback of constrained viscoelastic patches. However, combining damping treatment made of different viscoelastic materials allows to tune temperature range of efficiency of the damping treatment.

It is also shown that modal damping ratios decrease when increasing the rotation speed, due to the stress stiffening effect. Moreover, the way modal damping ratios vary with temperature is not affected by the rotation speed. Therefore, for a given operating temperature range, appropriate viscoelastic materials can be chosen in the early design process without having to consider centrifugal effects.

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