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ACTIVE DAMPING OF BLADED DISK ASSEMBLIES

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Although simple and effective, the performance of passive control techniques is limited in terms of authority and robustness. This paper studies the potential of using an active control system for mitigating the vibrations of a bladed rail assembly. First, we discuss the optimal placement of the piezoelectric patches. Although locating the patches at the blade root maximizes the strain energy, it results in important perturbations of the aerodynamic flow. Therefore, a pair of sensor and actuator is placed inside the bladed rail in order to have an as large as possible electromechanical coupling and a collocated system, i.e., alternating pole and zero. Second, an active control strategy using an integrator is designed and assessed in terms of attained closed-loop damping. The performance of the active damping is eventually compared with that of classical piezoelectric shunt damping.

Keywords: Bladed rail, High modal density, lightly damped structure

1. Introduction

Although lightweight design of turbomachinery structures reduces energy consumption, the associated low internal damping may result in high cycle fatigue problems caused by large resonant stresses. This has motivated researchers to develop vibration mitigation strategies using damping enhancement approaches such as passive, semi-active and/or active control systems.

When it comes to rotating structures, friction damping devices [1] are considered as an effective passive control system. With these techniques, energy is dissipated when relative motions occur between the structure and the damping device [2]. Although friction is a nonlinear phenomenon and is very difficult to assess, many numerical simulations and experimental tests investigated the performance of this control technique [3] and [4].

In turbomachinery applications, passive piezoelectric shunt damping has also been the subject of many research activities. This control technique was first introduced by Hagood and von Flotow [5]. The application of piezoelectric shunt on a bladed structure was first studied by Schwarzendahl et al. [6], where maximum amplitude reduction of the first mode was highlighted to optimize the shunt circuit. Mokrani et al. [7] tuned the shunt elements using the average frequency of the first bending modes family. Tang et al. [8] introduced an active-passive hybrid piezoelectric shunt network to damp the vibration of rotationally periodic structures. They showed that the hybrid configuration can dissipate more energy than purely passive control systems while requiring less control effort than purely active ones. In order to reduce the complexity required to implement a passive system targeting multiple modes, Kauffman et al. [9] proposed a semi-active approach using low-power frequency-switching. The use of negative capacitance to enhance the control performance of the passive shunt has been studied on a test-rig of a bladed disk model with eight blades [10]. Moreover, active control systems are employed to reduce fan rotor-stator interaction noise [11] and the vibration of subscale composite fan blades [12]. In most of the studies using piezoelectric patches, those elements are mounted directly on the blades. A major issue of such design is that this can disturb the aerodynamic flow.

This paper considers the vibration mitigation of the bladed rail shown in Figure 1a, which can be viewed as a simplified bladed assembly. Indeed, the bladed rail is a periodic structure which exhibits multiple modes in a narrow bandwidth of frequencies known as families of modes. The aim of this work is to assess the control performance of an active control system to damp the resonances of the first family in comparison to that of passive piezoelectric shunt damping. Similar control strategies will be effective to be implemented to any rotationally periodic structures which have similar dynamic behaviour.

Several active control laws, i.e. direct-velocity-feedback (DVF) [13], integral-force-feedback (IFF) [14], digital shunt absorber [15], positive-position-feedback (PPF) [16] etc., have been proposed to damp the vibration of flexible structures. Among them, PPF is proposed as an effective control technique for those

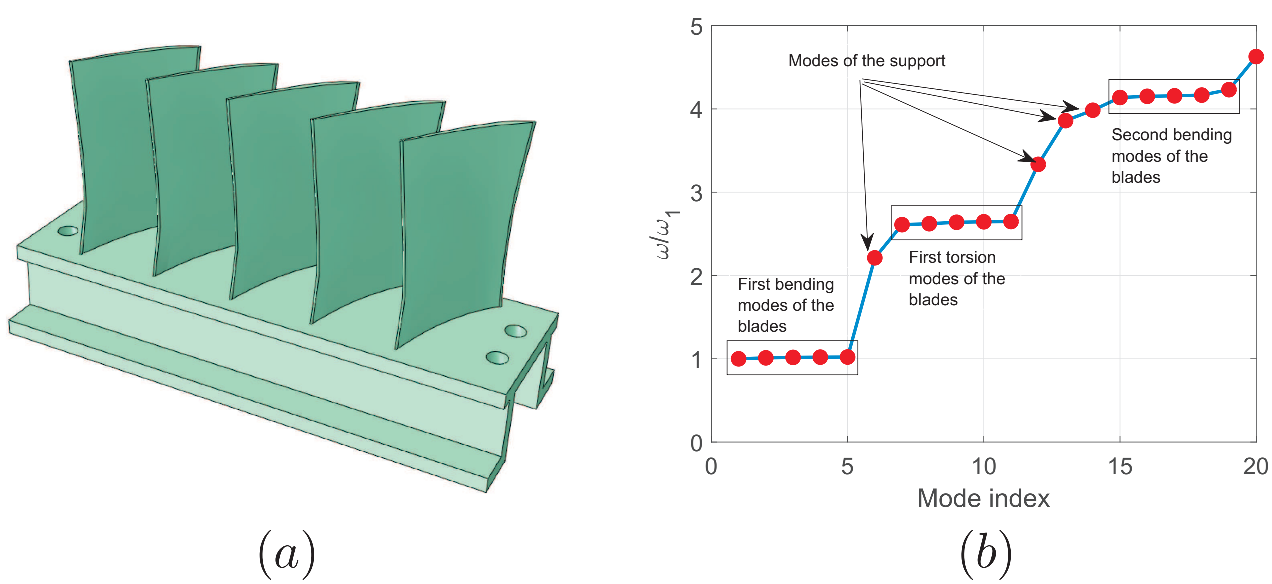

Figure 1: (a) Model of the bladed disk assembly. (b) Normalized resonance frequencies of the bladed assemblies (ω_1 is the first bending mode frequency of the first family)

primary systems having no high-frequency roll-off. It has been optimized analytically using maximum damping [17], H_2 [18], H_∞ [19] on a simple structure with finite number of degrees of freedom (DOF). While very effective, it can be implemented only on the plant where the zeroes alternate with poles. For such plants, an integral feedback control is of interest because, first, it is broadband which is able to damp multi resonances and it is also less sensitive to resonance uncertainty which is almost the case here thanks to the mistuning of the blades. Both control techniques, i.e. active and passive, are optimally designed and compared in terms of the closed-loop damping.
The remainder of the paper is organized as follows. The next section discusses the numerical model of the system under consideration. The optimal location of the piezoelectric patches is investigated in this section too. The design of the control systems is studied in Section 3. The conclusions are drawn in Section 4.

2. Modeling of the bladed disk assembly

The monobloc bladed rail (shown in Fig. 1a) is considered in this study. It is made of Aluminum and the geometry of the blades is similar to that of the conventional compressor bladed drum (BluM) [20]. The normalized resonance frequencies of this structure are illustrated for the first twenty modes in Fig. 1b. One sees that resonances appear in families, where the motion of the blades dominates, as well as isolated ones, which corresponds to the modes of the support. The resonances in the families are very close to one another and the separation between them depends on the rigidity of the support. The more the support is rigid, the more those resonances are close to each other. We are interested in the first family of modes. The mode shapes of this family are shown in Fig. 2. As it can be seen, the support has almost no motion while maximum displacements occur at the tip of blades.
In order to damp the vibration, piezoelectric patches are required to be mounted on the part of the structure which maximizes the strain energy and subsequently maximizes the observability and controllability of all modes. Of course, for the first family of modes, the maximum strain energy occurs at the root of each blades. However, aerodynamic flow becomes a critical issue in this case. We propose to mount those active elements on the internal part of the support. The strain energy distribution of the first five modes corresponding to the first modes family is shown in Fig. 3 for the internal part of the support. This plot

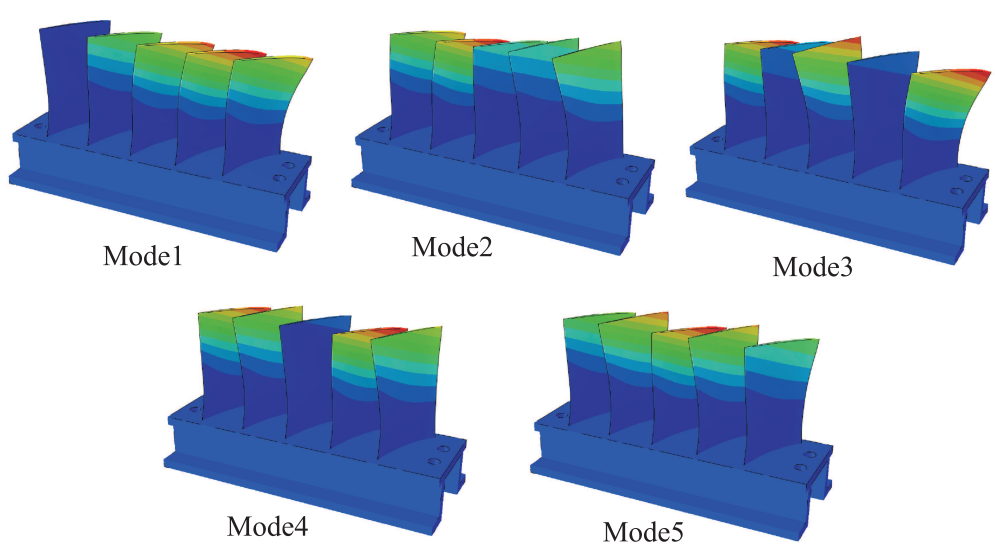

Figure 2: Bending mode shapes of the first family

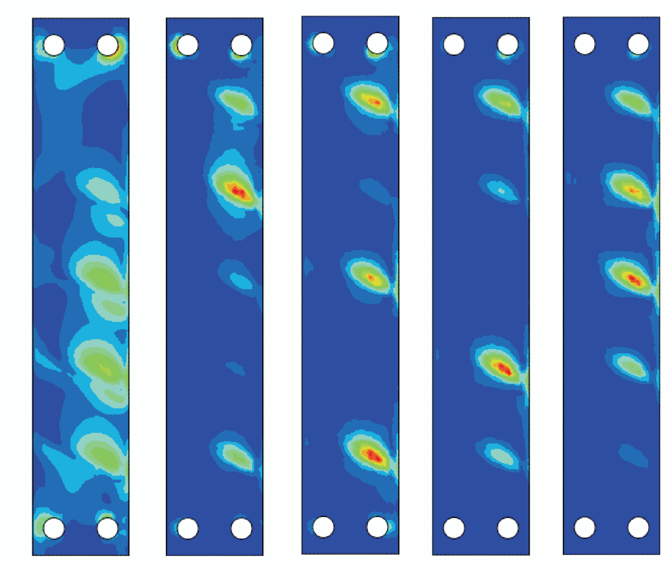

Figure 3: Strain distribution of the internal part of the support for the first family of modes

is interesting because it shows that some local strain distribution is imposed by the motion of each blade in this region. Consequently, five piezoelectric patches (PIC255 $0.015 \times 0.01 \times 0.0005 m^3$) are mounted there as it is shown in Fig. 4a for the application of the piezoelectric shunt damping. Also, smaller piezoelectric patches (PIC255 $0.015 \times 0.005 \times 0.0005 m^3$) are used in pairs of sensor and actuator (Fig. 4b) for the implementation of active control system.

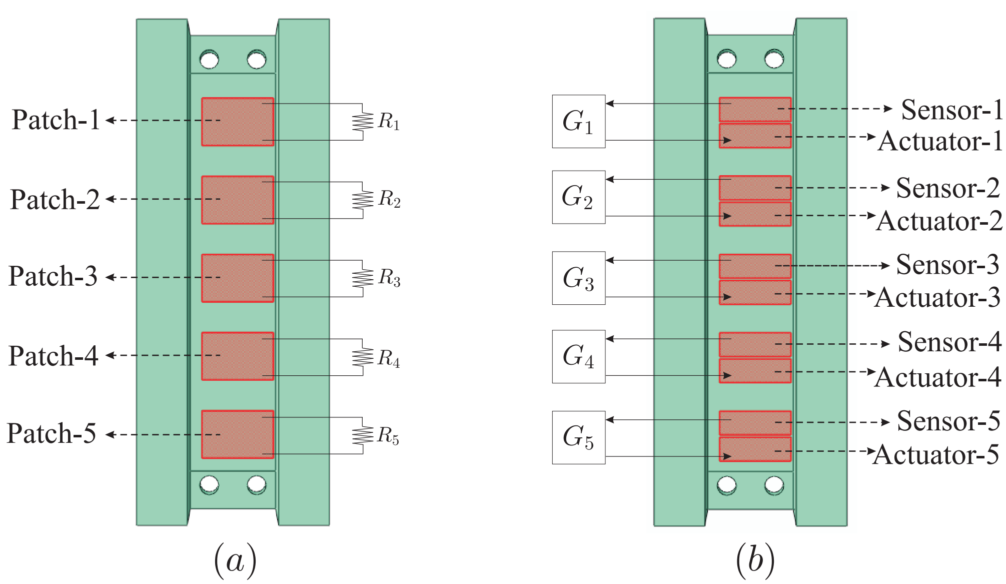

Figure 4: Configuration of the piezoelectric patches for the implementation of (a) the passive resistive shunt and (b) the active control system

3. Control design

The aim of this section is to design and compare a passive and an active control device which are less sensitive to resonance uncertainties. Although resonant controllers like piezoelectric RL shunt damping outperform broadband vibration absorbers in terms of vibration attenuation, tuning their parameters is highly dependent on the precise estimation of the primary system resonances. Therefore, resistive shunt damping is proposed in the present study for the passive control system due to the robustness to fluctuations of resonances. The piezoelectric patches shown in Fig. 4a are then modeled as charge actuators and voltage sensors. Fig. 5a illustrates the frequency-response-function (FRF) from the charge actuator of each piezoelectric patch to the voltage sensor of the same patch. Note that the frequency is normalized with respect to the frequency of third mode. Basically, the frequency of zeroes corresponds to the resonances of the structure when the electrodes are short-circuited and the frequency of poles is related to the resonance of the structure when the electrodes are open-circuited. It has been shown by Preumont [21] that assuming the modes are well separated in frequency, the separation between the frequency of the pole and the zero determines the maximum attainable damping ratio in the closed loop response for the i-th mode. The maximum modal damping (ξ_i^{max}) is then given by:

$$\xi_i^{max} = \frac{\omega_i^p - \omega_i^z}{2\omega_i^z} \tag{1}$$

where ω_i^z and ω_i^p are the frequencies of zero and pole of the i-th mode, respectively. Fig. 6a depicts the maximum achievable closed-loop damping of each mode by using different piezoelectric patches. In order to optimize the damping of each mode, one patch is used to maximize the damping of one mode. Therefore, first, fourth, fifth, second and third patch are used to maximize the damping of mode 1, mode 2, mode 3, mode 4 and mode 5, respectively.

On the other hand, low electromechanical coupling between the transducers and the structure significantly diminishes the control authority of the passive technique. Therefore, let us use a simple broadband active control law, i.e. an integrator, to damp the resonances of the first family based on the configuration shown

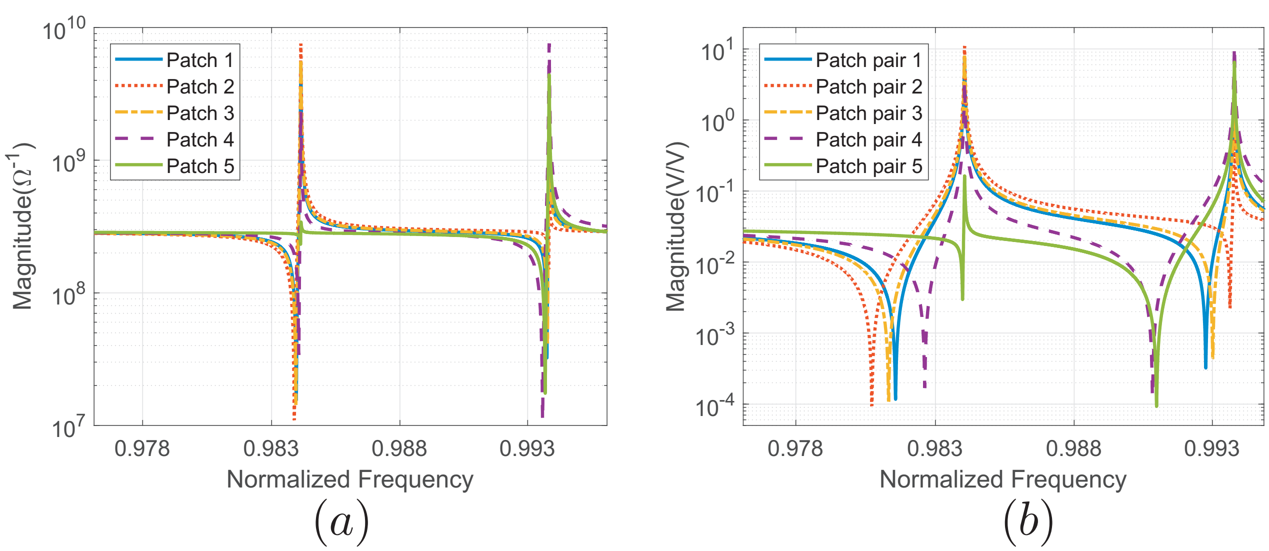

Figure 5: (a) The magnitude of the transfer function from charge to the voltage of each piezoelectric patches. (b) The magnitude of the transfer function from voltage actuator to the voltage sensor of each pair of patches. (The frequency is normalized with respect to the frequency of third resonance.)

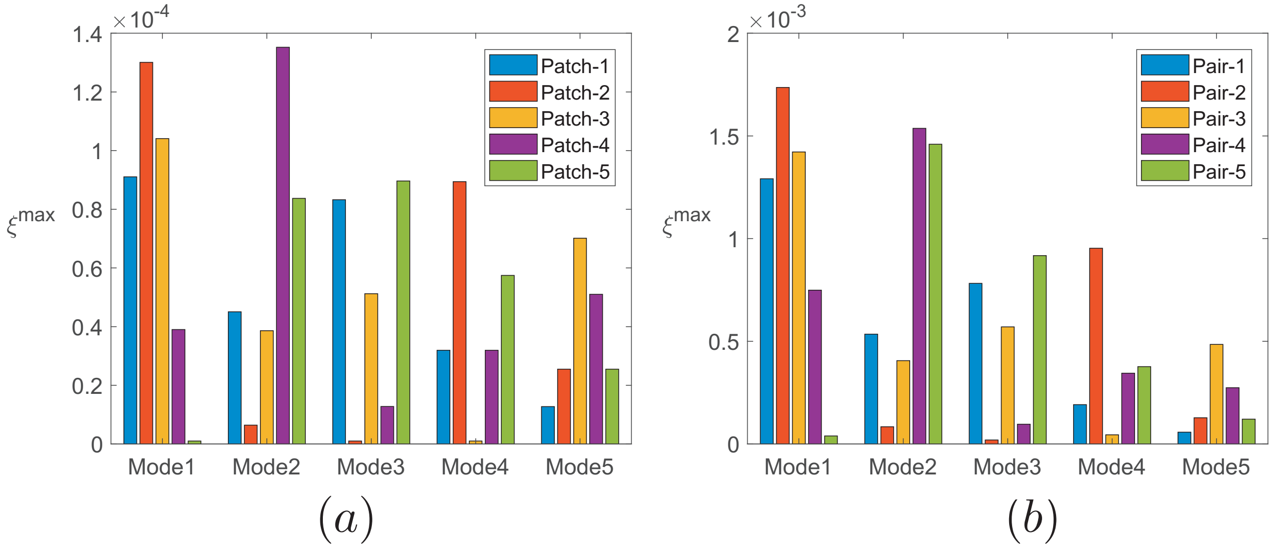

Figure 6: The maximum attainable closed-loop damping of each mode by the implementation of (a) the passive resistive shunt, or (b) the active control system

in Fig. 4b. Fig. 5b shows the magnitude of the transfer function from each actuator to the collocated sensor for the first two modes. One sees that the system contains alternating zero-pole configuration with no high frequency roll-off. For each mode, the frequency of the pole is the same for all pairs because it corresponds to a resonance frequency of the structure, while the frequency of the zero changes. According to Fig. 6b which demonstrates the maximum achievable closed-loop damping using different pair of patches, the order of the closed loop damping ratio is ten times higher than that of resistive shunt. As it was already explained in the previous section, the maximum displacement of the first family of modes occurs at the tip of the blades. Then, a representative compliance curve is defined as a performance

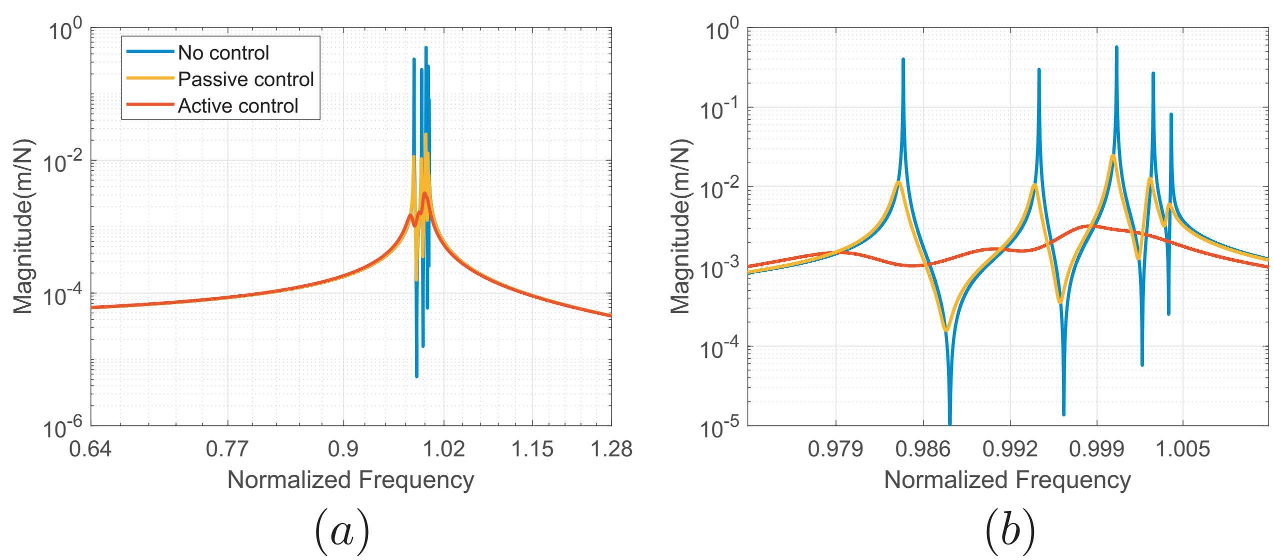

Figure 7: (a) FRFs of the system with no control (blue curve), the active control (red curve) and the passive absorber (yellow curve), (b) zoom close to the resonances. (The frequency is normalized with respect to the frequency of third resonance.)

index from an excitation force to a displacement sensor which are located at the tip of one blade. Fig. 7 presents the FRFs of the performance index obtained with and without control devices. This figure shows that the active control system provides an attenuation of the response amplitude which is 18dB greater than for the passive resistive shunt

4. Conclusions

Two vibration damping devices based on active and passive control systems have been designed and implemented numerically on the bladed rail assembly to damp the resonances of the first modes family. Due to the lack of possibility of locating piezoelectric patches on the blades to maximize the strain energy, the transducers have been placed on the support. It has been shown that the control performance of the passive resistive shunt in terms of closed-loop damping ratio is low because of the low electromechanical coupling between the transducer and the structure. Great improvements have been brought to the total damping of the system by using simple active control system. Future work will concentrate on designing a new active control law which improves the damping capability of the active control technique.

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