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Using Complex Nonlinear Normal Mode to Design a Frictional Damper for Bladed Disk

Y. Sun, J. Yuan, L. Pesaresi, E. Denimal and L. Salles

Dynamics Research Group
Imperial College London
London, United Kingdom
ys5113@ic.ac.uk

Abstract A numerical methodology is described to design a frictional damper for blade structure within aircraft engine. A finite element beam-platform model is used for preliminary design stage. The frictional damper is designed based on two parameters, contact angle and vertical position of the platform. Nonlinear modal analysis is used to investigate the nonlinear dynamic behaviour and damping performance.

High cycle fatigue (HCF) is one of the common failure of turbines bladed disks within aircraft engines and normally caused by large vibrational stress. Dry friction dampers are widely used in turbomachinery industrial, since vibrational energy can be released through rubbing motion between the contact surfaces. The placement of dry friction damper is also important. In literature, there are several types of dry friction dampers within blade-disks [3]: root joints in Fig.1a [6], tip shrouds in Fig.1b [4] and underplatform dampers in Fig.1c [5]. Those interaction force, friction, between contact surfaces are strongly nonlinear due to stick-slide and separation, leading to complex dynamic behaviour. Therefore, after taking the friction into consideration, the techniques for nonlinear dynamic analysis are required to solve this complex problem.

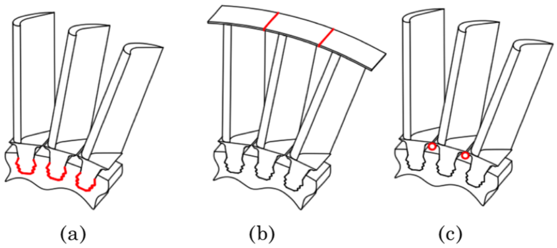


Figure 1: Different frictional dampers [1]

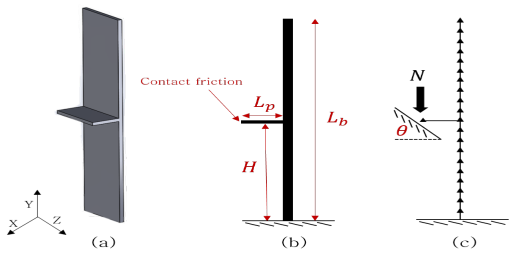
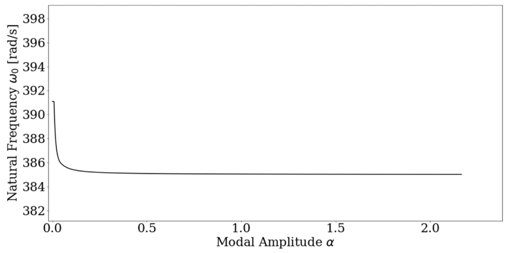


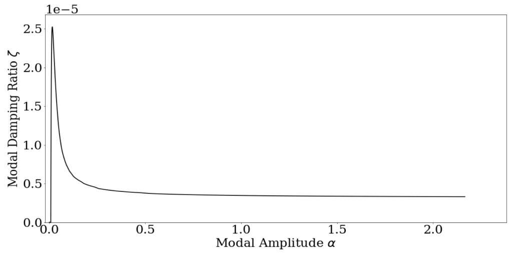
Figure 2: Beam-Platform Model

The objective is to determine the influence of the position of the damper on nonlinear dynamic behaviour of the system. Hence, a 2-D case is investigated. The blade is modelled with finite element beam-platform model displayed in Fig.2, where the platform can be located between the ground and tip of the beam. The position of platform is characterized by its vertical position H. The tip of the platform is in contact with frictional damper, the contact is characterized by the contact angle denoted θ. Therefore, this frictional damper is designed based on two parameters, contact angle $\theta \in [0°, 90°]$ and vertical position of the platform $H/L_b \in (0, 1]$. By varying the design parameters, all three types dampers given in Fig.1 can be modelled. A 2-D contact model is used to simulate the contact forces. Ideally, there are three contact status: separation, sticking and sliding.

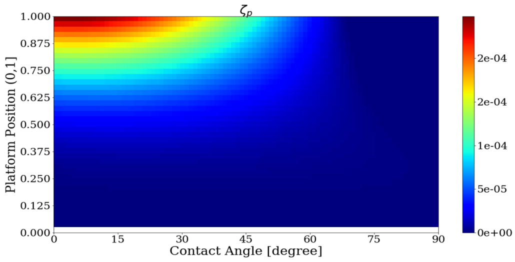
Complex nonlinear normal modes are computed through the nonlinear modal analysis based on the method proposed by Krack [2]. Harmonic Balanced Method with continuation technique is the numerical approach used to solve the autonomous equation of motion in Eq.1, where $\mathcal{Q}$ is mass normalized displacements; $\mathcal{F}_c$ is contact forces. ζ is a negative artificial damping to



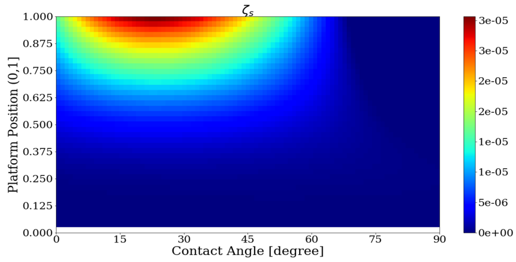
(a) Frequency-Amplitude Plot



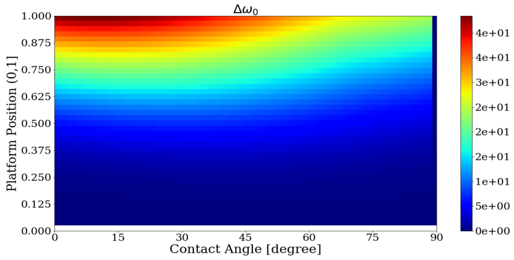
(b) Damping Ratio-Amplitude Plot



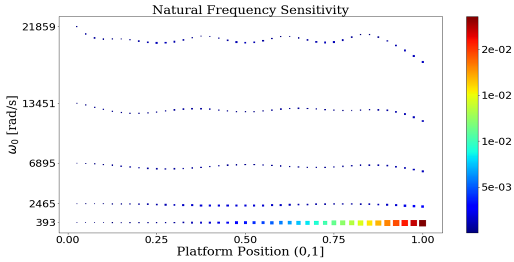
(c) Maximum Damping Ratio ζ_p



(d) Damping Ratio at Large Amplitude ζ_s



(e) Natural Frequency Shift $\Delta\omega_0$



(f) Natural Frequency Sensitivity

Figure 3: Overview of the results

compensate the energy lost due to friction [2]. System vibrates at its natural frequency ω_0 and α is modal amplitude. Detailed description of numerical approach can be found in [7].

$$\mathbf{M}\alpha\ddot{\mathcal{Q}}(t) - (\zeta \times \mathbf{K})\alpha\dot{\mathcal{Q}}(t) + \mathbf{K}\alpha\mathcal{Q}(t) + \mathcal{F}_c(\alpha\mathcal{Q}, t) = 0 \tag{1}$$

The 1^{st} bending mode is chosen to assess the damping performance. After the nonlinear modal analysis, natural frequency ω_0 and modal damping ratio ζ are calculated within range of modal amplitude α. As shown in Fig.3a and 3b, the natural frequency and modal damping ratio are plotted against the modal amplitude. When the modal amplitude is low, the contact status is sticking and the whole system is purely linear. The natural frequency of the system is constant and there is no energy lost due to the friction. Then, the modal amplitude is increased to certain value, the platform starts to slide. In this case, two contact points are partial-sliding-partial-sticking. In this case, both natural frequency and modal damping ratio reach to a steady value. The softening effect is caused by change of the stiffness at the contact points. To assess the damping performance, three objectives are chosen: modal damping ratio at peak ζ_p and stable region ($\alpha = 2$) ζ_s as well as the shift of natural frequency $\Delta\omega_0$ in Fig.3c, 3d, 3e.

The damping performance is evaluated for whole design space and it appears that it is highly sensitive to both design parameters. The optimized damping performance is achieved while the contact angle θ is around 25°-30°. Shift of natural frequency can be explained by natural frequency sensitivity to the contact stiffness as shown in Fig.3f. Generally, underpaltform damper with desired contact angle is able to provide effective damping and acceptable shift of natural frequency. Uncertainty Quantification with Latin Hypercube Simulation will be investigated in near future to taking wearing effect and manufacturing tolerance of frictional damper into consideration.

References

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