

A Study of the Contact Interface for Compressor Blisks with Ring Dampers Using Nonlinear Modal Analysis

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Abstract. The integrally bladed disks, also known as blisks, have been widely used in industrial turbomachinery because of their benefits in aerodynamic performance and mass reduction. Friction damping is considered as the major damping sources in turbomachinery. However, in blisks, the friction damping is negligible due to the lack of the contact interfaces. The friction ring dampers are one of the emerging external damping sources for blisks. In this paper, a full-scale blisk with a friction ring damper is studied, where a 3D contact element is used to compute the contact frictions. The blisk and ring damper is investigated using their damped nonlinear normal modes. The modal damping can be directly calculated and used to quantify the friction damping generated by the ring damper. The contact behaviour within the contact interface is further analysed. The nodes with initial gap show less damping ability. The separations within the contact interface are expected to be avoided to achieve a better damping performance.

1. Introduction

The bladed disk is an important structural component in industrial turbomachinery. The bladed disks are almost designed to their structural limit due to the higher efficiency required. In turbomachinery, the bladed disks are usually working under extreme environment, i.e. high thermal stress, centrifugal stress and vibration loading. The large vibrational stress on the bladed disks leads to the high cycle fatigue which is considered as the major reason for most failures of aircraft engines. In industrial turbomachinery, the material damping and aerodynamic damping are relatively low. Therefore, the external damping source is required to reduce the vibrational amplitude under resonance. The friction damping is a common and effective damping technique used in industrial turbomachinery. The friction dampers have been well-established in literature [5, 12, 18].

However, the use of blisks has become popular in the industry. The traditional bladed disk consists of a single disk and with several individual blades. In blisks, the structure is manufactured by a single-piece component. The blisks usually have a lower mass and better aerodynamic performance compared to the traditional bladed disks. However, due to lack of the contact interfaces, the internal damping of the blisks is extremely low and extra contact surfaces are expected to increase the friction damping. One of the emerging techniques is the use of the friction ring damper. A friction ring is held within the groove of the blisk by the centrifugal



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force. The friction ring damper is known as an efficient damping technique in other rotating structures, i.e. labyrinth seals [11] and gears [24, 22].

In industrial turbomachinery, the dynamic analysis of the blisks with friction dampers has been attempted by many researchers. Laxalde et al. proposed a lumped parameter model and the influences of the mass of the ring damper have been studied using forced response analysis [6]. Sun et al. applied the nonlinear modal analysis on the same lumped parameter model [17]. Full-scale structure for the blisk with ring damper has been numerically analysed and studied in literature by many researchers [9, 1, 10, 19]. The experimental works were completed by Laxalde et al. and results are compared between the experiments and numerical simulations [7]. The influence of the geometry on V-shaped ring damper has been studied by Tang and Epureanu [21]. Most researches for the ring damper focused on the damping ability of the ring damper. The contact interfaces between the blisk and ring damper have seldom been studied in literature. In this paper, a full scale blisk with friction ring damper is considered as the academic test case to study the contact behaviours within the contact interfaces between the blisk and ring damper. In addition, based on the results from this work, some guidelines of the design of ring damper are proposed.

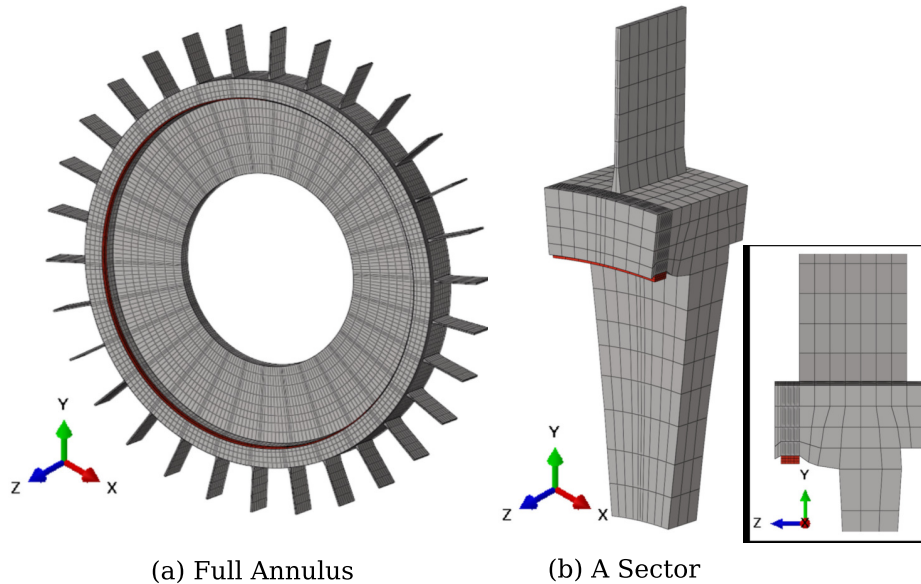


Figure 1. Finite Element Model: (a) Full Annulus of the Blisk and Ring Damper (Red); (b) A Sector of the Blisk and Ring Damper (Red).

The dynamic analysis of the blisk with ring damper is achieved by nonlinear modal analysis. The nonlinear modal analysis has been well established in past several decades. The nonlinear modes were defined by Rosenberg [14] in a conservative nonlinear system. Then, Shaw and Pierre extended the nonlinear modes to non-conservative nonlinear systems using the concept of invariant manifold [16]. The nonlinear modes for a non-conservative nonlinear system were named as damped Nonlinear Normal Mode (dNNM). After that, the extension of concept of complex nonlinear mode was described by Laxalde et al. [8], which can be applied to compute the dNNMs for systems with non-conservative nonlinearities, i.e. system with friction joints. After that, a numerical method, namely Extended Periodic Motion Concept (EPMC), was proposed by Krack [4]. In this method, the dNNMs are numerically calculated based on a periodic concept. In this work, the nonlinear modes of the system are calculated using the EPMC.

The present work aims to study the contact behaviours within the contact interface between the blisk with ring damper using nonlinear modal analysis. The numerical method EPMC is used to calculate the dNNMs, which can be considered as the resonant solutions for forced response [20]. The paper is organised as follows: the Finite Element (FE) model used is firstly given; then, the modelling methodology is described; after that the numerical methods used to compute the dNNMs are briefly described; the contact behaviours within the contact interface are investigated; finally, some guidelines for the design of the ring dampers are proposed.

2. Finite Element Model

A full-scale FE model is used in this work to represent the blisk and ring damper, as shown in Fig.1(a). This FE model is considered as the test case to study the contact behaviours of the ring damper. The model is built in a commercial FE solver ABAQUS. The whole structure is made by a homogenous and isentropic titanium material. The blisk and ring damper are assumed to be perfectly cyclic symmetry and modelled by a sector with cyclic symmetric boundary conditions as shown in Fig.1(b). There are 30 sectors in total. The ring damper is held to the underneath of the blisk.

3. Modelling Methodology

The FE model for the blisk and ring damper has been described in the previous section. The objective of this section is to describe the methodology used in the present work. Firstly, a linear modal analysis of the blisk is completed and a specific mode is selected for the subsequent studies. Then, the rotor dynamic analysis is applied to this selected mode and a resonance case under engine order excitation is obtained. After that, the resonance case is analysed by a nonlinear static analysis. The contact pressure/gap distribution within the contact interface is computed from the nonlinear static analysis and is considered as preloading conditions for the nonlinear modal analysis. The linear modal analysis, rotor dynamic analysis and nonlinear static analysis are completed in commercial FE solver ABAQUS.

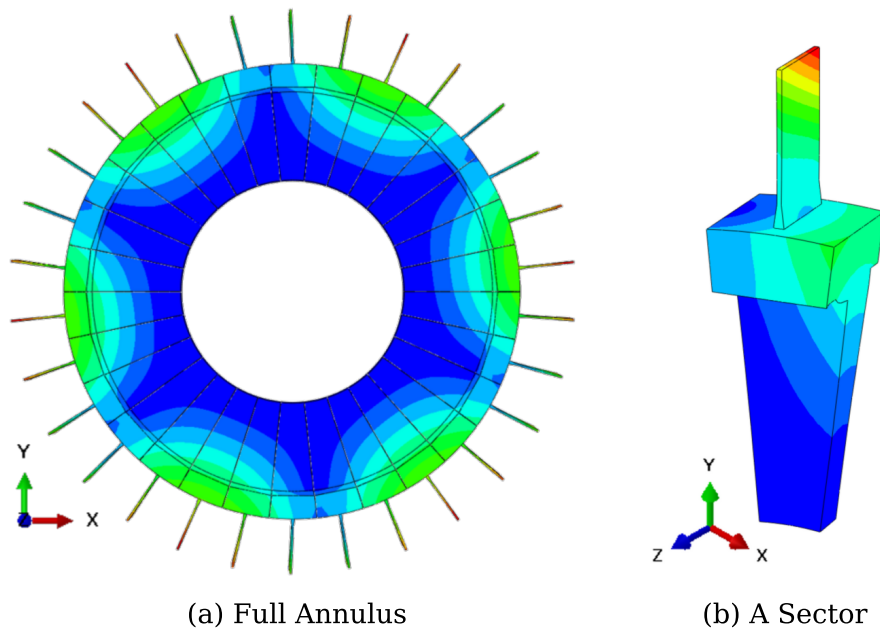


Figure 2. Modeshape of the Mode Studied: (a) Full Annulus; (b) A Sector.

After applying the linear modal analysis, the family of the 1st disk mode with nodal diameter 3 is selected as the test case as shown in Fig.2. The modeshape of the selected mode is highly dominated by the motion of the disk. When the engine rotating speed is 2081 rpm and 27th engine order excitation is applied, then selected mode is excited. This resonance case is considered for the subsequent study. The contact pressure/gap distributions are obtained by the nonlinear static analysis and shown as a map on contact interface in Fig.2. The contact pressure/gap distributions show the initial static contact status for the contact nodes pairs within the contact interface. When the contact pairs show a positive gap, the nodes are separated with the initial gap (i.e. the nodes near the cyclic boundary). The contact pressure is concentrated on the middle of the contact interface. These nodes with positive contact pressure are generally in-contact in a static equilibrium status.

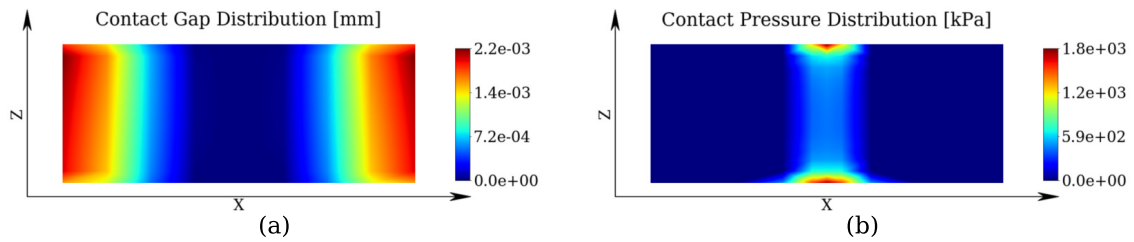


Figure 3. Contact Preloading within the Contact Interface for the Selected Resonance Case: (a) Contact Gap Distribution; (b) Contact Pressure Distribution.

4. Nonlinear Modal Analysis

The description of the FE model and modelling methodology have been given in previous sections. In the present section, the numerical formulations of the nonlinear modal analysis are demonstrated. Firstly, the mass and stiffness matrices of the full-scale FE model is reduced by Craig-Bampton method [3] and cyclic symmetric reduction [13]. The computational expenses can be reduced to an acceptable level. Then, the reduced mass and stiffness matrices are used in nonlinear modal analysis. The contact friction forced is calculated based on a well-established 3D contact element [23], which is a node-to-node contact model. In this contact model, there are normally three contact conditions, which are sticking, separation and siding.

The equation of motion for an autonomous (undamped and unforced) nonlinear system has been given in Eq.(1), where $\underline{\mathbf{M}}$ and $\underline{\mathbf{K}}$ are the reduced mass and stiffness matrices; $\underline{\mathbf{Q}}(t)$ is the displacement of the system; $\underline{\mathbf{F}}_{nl}(\dot{\underline{\mathbf{Q}}}, \underline{\mathbf{Q}})$ is the contact friction calculated from the 3D contact element in [23]. The total energy of the system is dissipated due to the rubbing motions within the contact interface. Therefore, the system is non-conservative and solutions are not periodic. According to the EPMC, an artificial damping term, $\underline{\mathbf{C}} = -\zeta\omega_0\underline{\mathbf{M}}$, is introduced into the system to balance the energy dissipated by the contact frictions, where ω_0 is the resonant frequency and ζ is the modal damping ratio. In addition, based on the concept of nonlinear modes, the dNNMs are energy dependent. In other words, the dNNMs and other modal properties (i.e. resonant frequency and modal damping ratio) vary with the energy level of the system. Therefore, a modal amplitude α is introduced and the solutions of the system can be represented as: $\underline{\mathbf{Q}} = \alpha \cdot \underline{\mathbf{Q}}_0$. Then, the equation of motion is rewritten and shown in Eq.(2). There are two extra constraints required to solve the equation below which are a mass normalisation and phase normalisation, which can be found in [4].

$$\underline{\mathbf{M}} \cdot \ddot{\underline{\mathbf{Q}}}(t) + \underline{\mathbf{K}} \cdot \underline{\mathbf{Q}}(t) + \underline{\mathbf{F}}_{nl}(\dot{\underline{\mathbf{Q}}}, \underline{\mathbf{Q}}) = 0 \quad (1)$$

$$\underline{\underline{\mathbf{M}}} \cdot \alpha \cdot \ddot{\underline{\underline{Q}}}_0(t) + \underline{\underline{\mathbf{C}}} \cdot \alpha \cdot \dot{\underline{\underline{Q}}}_0(t) + \underline{\underline{\mathbf{K}}} \cdot \alpha \cdot \underline{\underline{Q}}_0(t) + \underline{\underline{F}}_{nl}(\alpha, \dot{\underline{\underline{Q}}}_0, \underline{\underline{Q}}_0) = 0 \quad (2)$$

The numerical method used to solve the Eq.(2) is the harmonic balance method (HBM) with alternating frequency/time (AFT) method [2]. The HBM with AFT has been widely used to solve nonlinear dynamic problems in frequency domain. In nonlinear modal analysis, the dNNMs are expected to be computed for a range of modal amplitude. Therefore, the well-known continuation method is used to track the evolution of the system against a specific chasing parameter [15].

5. Results

The mode in 1st disk family with nodal diameter 3 is excited by engine order 27 excitation under engine rotational speed 2081 rpm. This resonance case is considered for the nonlinear modal analysis. The dNNMs and other modal properties are computed for a range of modal amplitude. To achieve a physical description of the modal amplitude, the displacement of the tip of the blade is used to represent the vibration amplitude.

The evolution of the resonant frequency ω_0 is demonstrated in Fig.4(a). When the structure is vibrating at low amplitude (the displacement is smaller than 1.5×10^{-2} mm), the ring damper is stuck to the underneath of the blisk. The resonant frequency ω_0 is a constant and equals to the sticking natural frequency of the structure [18]. When the vibration amplitude is greater than 1.5×10^{-2} mm, the contact pairs within the contact interface start to slide leading a softening effect (decrease in resonant frequency ω_0). The evolution of the modal damping ratio ζ is demonstrated in Fig.4(b). At the low vibration amplitude, the contact pairs are either sticking or separating (depending on the initial contact status). Further increasing the vibration amplitude, some contact pairs start to slide and the energy is dissipated by the rubbing motion between the contact nodes in contact pairs. The maximum modal damping ζ is around 0.1%.

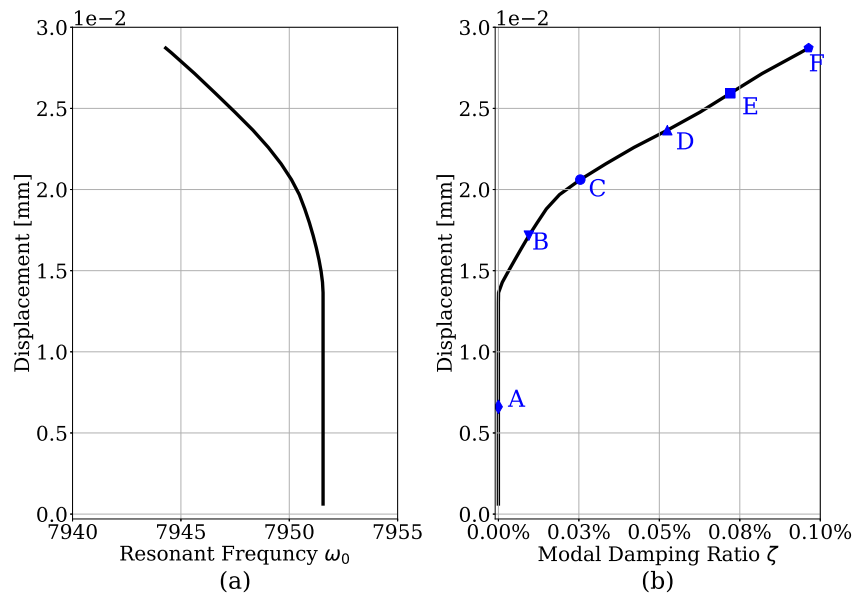


Figure 4. Results from Nonlinear Modal Analysis: (a) Resonant Frequency Plot; (b) Modal Damping Ratio Plot.

To further understand the contact phenomena within the contact interface, six cases with different vibration amplitudes are selected and marked in the Fig.4(b). The distributions of the energy dissipation within the contact interface are computed and shown in Fig.5. For each

contact node pair, the energy dissipated by one vibration period is the area enclosed by the hysteresis loop. For Case A, the modal damping ratio is zero and no energy dissipated shown on the map. By considering rest of the cases, the modal damping ratio increases with the vibration displacement (see Fig.4(b)). It is obvious to notice that the nodes in the middle of the contact interface show a higher damping ability. Most of the vibrational energy is dissipated by the nodes in the middle of the contact interface. The nodes near cyclic boundaries have negligible energy dissipated. The nodes in the middle are in-contact at the static equilibrium status whereas the nodes near cyclic boundaries show initial gaps. When the blisk starts to vibrate, these in-contact nodes transit from the sticking condition to the sliding condition. The energy of the system is dissipated due to the rubbing motion of the sliding nodes within the contact interface. However, the contact conditions for these separated nodes are not completely sliding. During one vibration period, the nodes show both separation and sliding. Due to existence of the separation part, the energy dissipated by these nodes are relatively lower than the sliding nodes. The similar results can be found in [18].

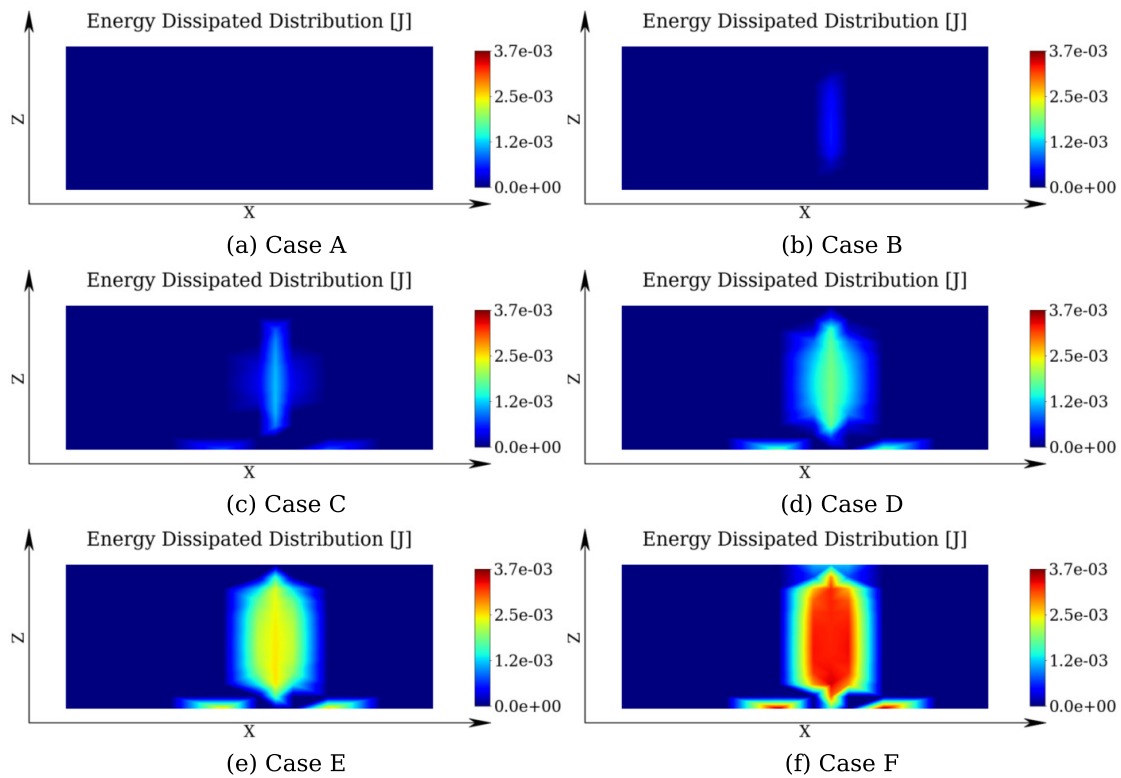


Figure 5. Energy Dissipation Distributions for Six Cases Shown in Fig.4(b).

6. Conclusions

A full-scale finite element model is used in this paper to represent the blisk and ring damper. This blisk with ring damper is studied using nonlinear modal analysis. In nonlinear modal analysis, the evolutions of the resonant frequency and modal damping ratio are directly computed to understand the dynamic behaviour and damping performance of the ring damper. The contact behaviours and the maps of the energy dissipated within the contact interface are investigated. The results show that the nodes with initial gap has less damping ability. To improve the damping performance of the ring damper, it is important to avoid the separation within the

contact interface. The geometric design of the ring damper can be considered as the future work to optimise the damping performance by manipulating the initial contact status.

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