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OPTIMIZATION OF BLADED DISK PASSIVE DAMPING ELEMENT

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Abstract: *This paper is aimed at reducing vibrations of bladed disk in particular operating state. The damping element used in this contribution is realized by a strap with isosceles trapezoidal cross-section area which is placed in hoop dovetail groove in the blade shrouding. The friction effect between the damping element and the groove side walls is included. Sliding between the contact surfaces leads to the dissipation of energy which causes decreasing of undesirable vibrations. The frictional force is influenced by centrifugal force acting on the damping element. The centrifugal force depends on disk angular speed, mass of the damping element and also on the size of contact area. It is shown that the axial vibrations of bladed disk in particular operating state is influenced by dimensions of the damping element cross-section Dimensional optimization is used to find the optimal dimensions. The process of finding optimal dimensions is described in this paper. The finite element model of bladed disk involving passive damping element is created and the critical operating state is selected. The selection of design variables is discussed. The objective function is defined and optimization is performed.*

Keywords: ***Bladed disk vibrations, vibrations damping, transient analysis, optimization.***

1. Introduction

The steam turbines are high loaded equipments. There are many possible resonant states which may result in high cycle fatigue failure. Sometimes the operating state can be close to the resonance state, which causes undesirable vibrations of bladed disk. One way of decreasing the excessive vibrations is employing a frictional effect of the suitable damping element. The damping element can be realized by a strap having isosceles trapezoidal cross-section area and can be placed in hoop dovetail groove in the blade shrouding. This type of the damping element is suitable especially for damping of the vibrations in the axial direction along with the mode shape with the nodal diameters. The damping effect is reached by sliding between the contact surfaces. The modal properties of the bladed disk are influenced by the sliding distance. Since the friction force depends on centrifugal force acting on the damping element and on angle of the side walls of groove and strap, the sliding distance can be influenced by dimensions of damping element. Optimal dimensions for considered operating state can be found.

The strategy of finding optimal dimensions can be following. Afterward the finite element model is created the modal analysis is performed since the knowledge of the modal properties (natural frequencies and mode shapes) is important for determination of the critical resonance states. The critical speed and corresponding excitation frequency are determined based on the selected resonance state. Since resonance frequency is influenced by the damping, it is difficult to determine corresponding critical excitation frequency exactly. Thus the transient analysis is performed, and excitation frequency is swept in this simulation. The assumed critical excitation frequency is between initial and end value of excitation frequency. This condition ensures that disk pass through the resonance state.

The objective function quantifies the vibration in axial direction. The definition is based on displacement of nodes on the disk circumference.

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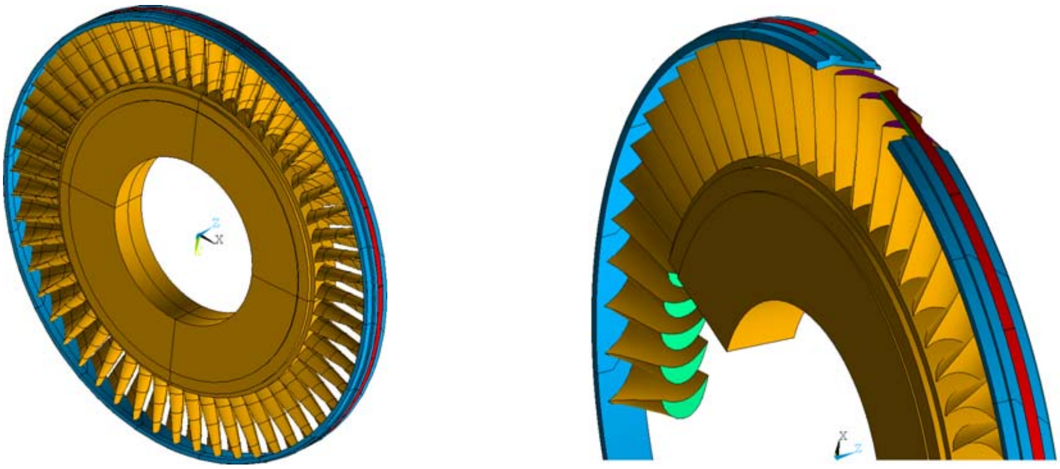
2. Model of Bladed Disk

The ANSYS v11.0 is used to create the finite element model. The geometry is shown in the Fig.1a). The bladed disk has 54 blades, which are coupled in 18 packets by segmental shrouding. The damping element (highlighted by red color in Fig. 1) is placed in the dovetail groove, created in the shrouding. The mesh is generated by 8-nodes structural elements SOLID 45. The contact respecting Coulomb's Law of friction is modeled between the side walls of the groove and the damping element. The surface-to-surface contact elements are used for the contact definition. Element type CONTA173 is used on the side walls of the damping element, and element type TARGE170 is used on the side walls of the groove. Static coefficient of friction is set to 0.7, kinetic coefficient is set to 0.6.

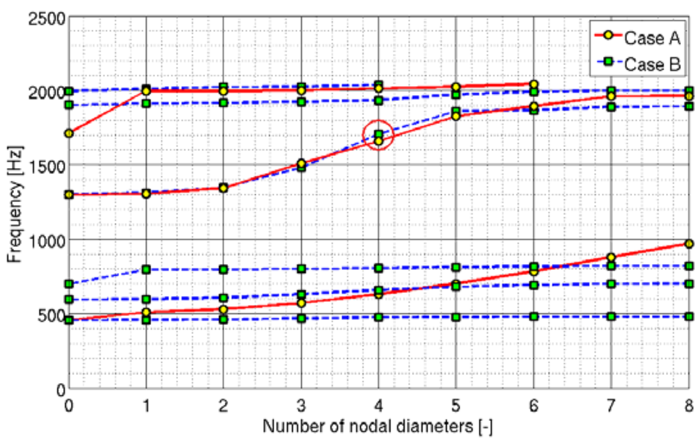


a) *b)*

Fig. 1: Model of Bladed Disk.

3. Critical Operating State

Critical operating state is selected based on the results of the modal analysis. Since including of the contact-type nonlinearities is not allowed in the modal analysis, two linear models are considered for performing modal analysis. In the first case the contact algorithm for contact elements between damping element and groove. Thus the strap and shrouding are stick together. The modal analysis results are labeled as "Case A" in the Fig. 2. In the second case the damping element is removed and contacts are deleted. Results are labeled as "Case B" in the Fig.2. Resonance frequencies of model including friction are considered between frequencies of Case A and Case B, or in close surroundings eventually. The operating state (resonance frequency and number of nodal diameters) that dimensions of damping element are optimized for is selected based on previous work (Lošák, Malenovský 2007). This operating state is marked by red color in Fig. 2.



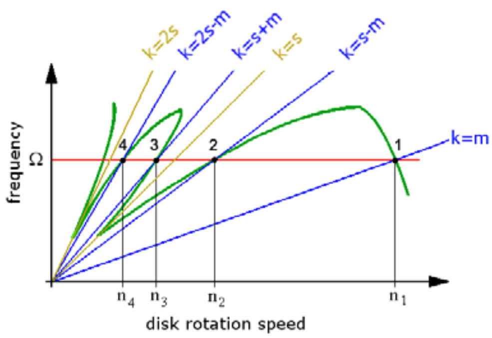


Fig. 2: Natural Frequency vs. Number of nodal Diameters Plot. *Fig. 3: Campbell Diagram.*

4. Boundary and Initial Conditions

The bladed disk is rigidly fixed on the inside diameter in the disk bore. The bladed disk is excited due to nonuniform pressure distribution behind stator grid. This pressure fluctuation is represented by axial and tangential harmonic force, acting on the end of each blade. The force on each following blade is shifted against the previous of the phase angle, which is calculated as:

$$\beta = \frac{z}{r} 2\pi \tag{1}$$

where: β is phase angle, z is number of the stator blades, and r is number of the rotor blades.

The frequency of the excitation force depends on selected operating state. The operating state assumed in this paper is marked by the red circle in Fig. 2. Determination of the corresponding critical rotation speeds is shown in the Fig. 3. The critical rotation speed is defined as cross point of the disk natural frequency with k-multiple of the natural frequency (Campbell, 1924). Calculation of *k* is also shown in the Fig. 3, *m* is number of the nodal diameters and *s* is number of the cyclic sectors, n_1 through n_4 are critical disk rotation speeds. As stated in Section 3 the excitation frequency is swept to ensure that the disk passes the resonance state. Thus it is necessary to calculate the excitation frequencies for both limiting states and define angular acceleration of excitation frequency. The angular velocity and corresponding frequency of excitation force can be determined as:

$$\alpha = \frac{\omega_e - \omega_i}{t} \tag{2}$$

$$F(t) = F_0 \sin\left(\left(\omega_i t + \frac{1}{2}\alpha t^2\right) + \beta(l_r - 1)\right) \tag{3}$$

where α is angular acceleration of excitation force, ω_i, and ω_e is excitation angular frequency corresponding to initial resp. end rotation speed, t is time of simulation. $F(t)$ is time-depended force (axial or tangential) F_0 is force amplitude (axial or tangential), β is phase angle (see Equation 1), l_r is sequential number of rotor blade (1 through 54).

5. Optimization

Since the damping element cross-section dimensions is optimized, the design variables are characteristics dimensions of the cross section area - height of damping element, cross section middle width and side wall slope angle. The damping element cross section along with visualization of design variables limits are shown in Fig. 4.

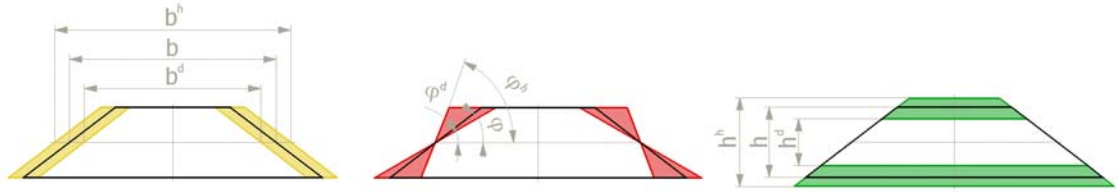


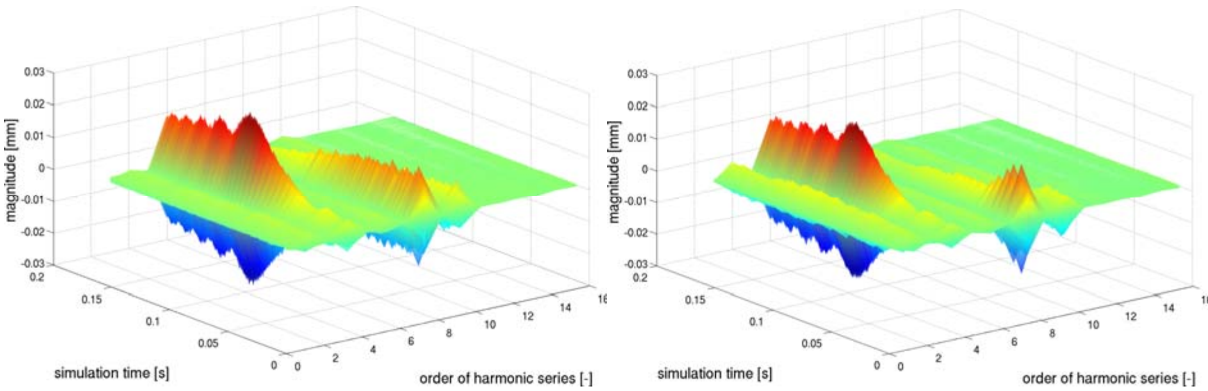
Fig. 4: Optimization Variables and Dimension Limits Visualization.

The objective function has to quantify the vibrations of the bladed disk. Since some nodes on the disk circumference can lie on the nodal diameters, the displacement in all nodes of the disk circumference is evaluated. This approach ensures that objective function quantify the axial vibration all around the disk. The objective function is shown in Equation 4. It is a sum of displacement absolute value of every node integrated over time.

$$\psi(\mathbf{x}) = \sum_{i=1}^{n} \left(\int_{T_1}^{T_2} |q_i(\mathbf{x}, t)| dt \right) \tag{4}$$

where: $\psi(x)$ is objective function, which depends on design variables x, T_1 and T_2 are initial and end point of evaluated time interval. Variable q_i is axial displacement of particular node, n is number of nodes on the disk circumference.

The Subproblem Approximation Method is used to find the minimum of the objective function. The number of nodal diameters vs. simulation time and magnitude is shown in Fig. 5a for starting values of optimization variables and in Fig. 5b for final values. Figures show that optimization of the passive damping element dimensions leads to decreasing of vibration. It can be seen that level of vibration is lower for mode shape with four nodal diameters as well as for mode shape with eight nodal diameters as. It seems that vibrations associated with eight nodal diameters mode shape are damped even more intense than those associated with four nodal diameters mode shape.



a) *b)*

Fig. 5: Representation of nodal diameters and amplitudes during simulation for a) not optimized and b) optimized dimension of damping element.

6. Conclusions

The strategy of optimization of the passive damping element cross section dimensions is shown in this paper. The dimensions are optimized for selected operating state with particular number of nodal diameters. The objective function composition is discussed and optimization is performed. The optimal dimensions are found. It is shown, that optimization leads to decreasing vibration generally, not only those associated with selected operating states and corresponding number of nodal diameters. It turns out that it is possible to use this process to optimize passive damping element geometry to decrease undesirable oscillations of bladed disk

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