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Optimal Vibration Reduction of the Flexible Shaft-Disk-Blades System Using a Set of Nonlinear Energy Sinks

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ABSTRACT: In this paper, the application of nonlinear energy sinks for indirect vibration reduction of the blades in a flexible shaft-disk-blades system of a real steam turbine is conducted. 37 packets of seven-connected blades are mounted on the disk. The cyclic symmetric finite element analysis is employed to perform frequency analysis of this system. For the 11th mode, which is a combination of the second bending mode of shaft and the third bending mode of disk-blades, a two degrees of freedom reduced order model is identified. Nonlinear energy sinks with a small mass, an essential nonlinear stiffness and a linear damping are installed on the reduced order model in the anti-node position of the disk. The Runge-Kutta method is used to solve the nonlinear equations of motion numerically. Optimum stiffness and damping of the absorbers are determined to minimize the vibration amplitude of the blades. The results show that the occurrence of strongly modulated response around the resonance leads to the desired vibration reduction of the blades. If the absorbers have large nonlinear stiffness or low damping, a saddle-node bifurcation and a wide island is appeared in the negative detuning frequencies, and the blade could experience large amplitude periodic oscillation.

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1. Introduction

Recently, modern turbines are designed close to their critical operating points with lower stability margin. Therefore, accurate dynamic analysis and using advanced vibration control systems are mandatory for them. Dynamic analysis of the shaft-disk-blades system of the turbine could be conducted using only a sector with fewer Degree Of Freedoms (DOFs) instead of the entire model [1,2]. The real resonances of bladed disk system are determined through Singh's Advanced Frequency Evaluation (SAFE) diagram. The major excitation on the blades is usually vane passing flow. The 2nd -6th Nodal Diameters (NDs) are more important in steam turbines and the occurrence of resonance in these NDs should be considered [3]. In-plane motion of a row of blades, at the first ND, is coupled by the bending and transversal modes of the rotor and No coupling exists with the torsional modes [4]. Three types of coupled motion could be seen in the system including inter-blade, shaft-disk-blade, and disk-blade [5]. The vibration amplitude of blades could be damped by adding Nonlinear Energy Sink (NES) into blades [6], on the disk of the Jeffcott rotor [7], or on bearings [8]. Because of the lack of own natural frequency of the NES, unlike the linear absorbers, two peaks is not appeared around resonance point of the system, and hence, NES has more effective performance.

Application of the NES in a real flexible shaft-disk-blades system with a lot of blades has not studied yet and is conducted in this research. The case study includes 259 blades in 37 packet of 7-connected blades by a tip shroud. A periodic model of the structure is analyzed using finite element software and various localized and coupled modes of the system are identified. Some NESs are added to the disk to suppress the blades vibration indirectly. Each NES includes a small mass, an essentially nonlinear stiffener and a linear damping. In the vicinity of the 1:1 resonance of the 11th mode of the system, nonlinear dynamic system equations are solved numerically using the Runge-Kutta method and the NES parameters are optimized for the best vibration suppression. System behavior sensitivity to the NES parameters is surveyed too.

2. Methodology

The disk-blade system is the eleventh stage of a twelve-stage 30 MW steam turbine. It is made of steel and its weight is 2001 kg. The periodic model of this system includes a set of 7-connected blades and a slice of the disk and shaft. This model has been meshed in a finite element software and the centrifugal force caused by the angular speed of 3000 rpm is applied on elements. Furthermore, the structural and modal analysis are conducted. The interaction between modes is investigated using the SAFE diagram. A strong structural coupling exists between blades, disk, and shaft motions in the 11th mode, and therefore, a high level of energy capturing

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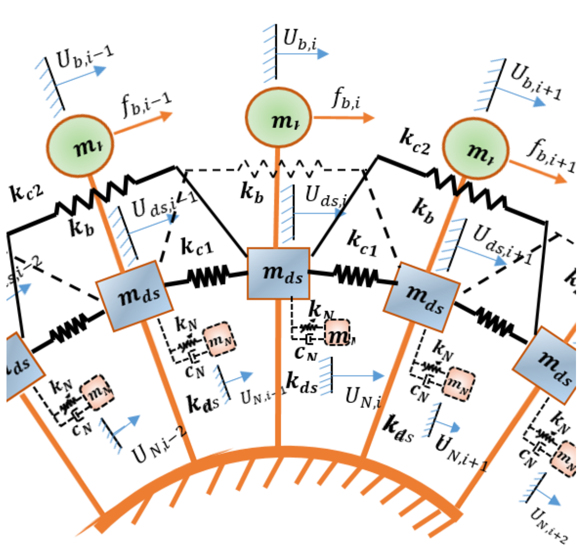

Fig. 1. Schematic of the two-DOF reduced order model; NES has been shown by dotted lines.

and suppressing by the NESs, mounted on the disk, could be occurred. Thus, this mode is selected for the NES tuning. Furthermore, a 2DOF reduced order model (Fig. 1) is identified for modelling of the 11th and 12th modes.

Around the 1:1 resonance of the system, vane passing flow exerts as an excitation force on blades with the frequency of Vane Passing Frequency (VPF). The deformation of the shaft or disk could only change the magnitude of the force. In each sector, a NES is mounted on the anti-node position of the desired mode shape on the disk (Fig. 2). Sector equations of motion are as follow:

$$\begin{aligned} & m_b\ddot{u}_{k,b} + c_b\dot{u}_b + k_b\left(u_{k,b} - u_{k,ds}\right) = f_b \cos\omega t \\ & m_{ds}\ddot{u}_{k,ds} + (k_b + k_{ds} + 2k_{c1}\left(1-\cos\left(\sigma_k\right)\right) + \\ & \quad 2k_{c2}\left(1-\cos\left(2\sigma_k\right)\right)u_{k,ds} - k_b u_{k,b} + \\ & \quad \epsilon k_N\left(u_{k,ds} - u_N\right)^3 + \epsilon c_N\left(\dot{u}_{k,ds} - \dot{u}_N\right) = 0 \\ & m_N\ddot{u}_N + \epsilon k_N\left(u_N - u_{k,ds}\right)^3 + \epsilon c_N\left(\dot{u}_N - \dot{u}_{k,ds}\right) = 0 \end{aligned} \tag{1}$$

Aerodynamic damping was included in the system. System vibration is studied in the vicinity of the 1:1 resonance at the first ND of the 11th mode, therefore, the relative excitation frequency is $\omega/\omega_{2,11}=1+\epsilon\sigma$, where σ is the detuning parameter and ϵ is a small number.

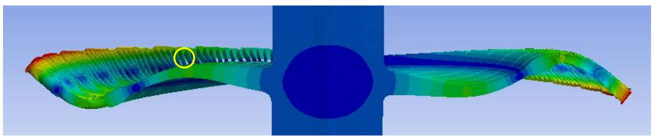

Fig. 2. Shaft-disk-blades system cross section view at the first ND of the 11th mode; circle shows the position of NES.

3. NES Optimization

Nonlinear equations of motion of the system has been solved numerically using the Runge-Kutta method. Fig. 3 displays the 3Dimension (3D) plot of the blade maximum frequency response in terms of NESs' stiffness and damping.

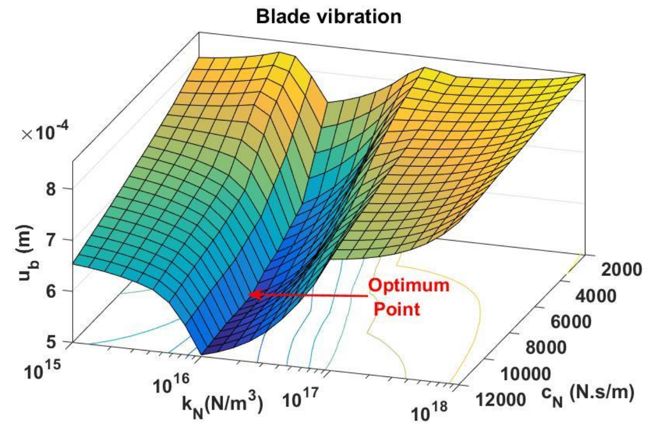

Fig. 3. Maximum of the blade frequency response in terms of the NES stiffness and damping.

The optimized parameters of NESs are as follow:

$$\begin{aligned} & k_N = 1.0\times10^{16}\,\frac{\mathrm{N}}{\mathrm{m}^3} \\ & c_N = 8000\,\frac{\mathrm{N.s}}{\mathrm{m}} \end{aligned} \tag{2}$$

4. Discussion and Results

Shaft-disk-blades frequency response with the optimized NES has been shown in Fig. 4. The Strongly Modulated Response (SMR) has occurred in a frequency bandwidth of 4 Hz in the range of $-0.02<\sigma<0.15$. Nonlinear system responses larger than the linear system is occurred in the ranges of $0.12<\sigma<0.2$ and $0.33<\sigma<0.47$, of course, too smaller than the resonance amplitude. In other frequency ranges, the blade vibration amplitude is equal or smaller than the system without NES. Decreasing damping and increasing stiffness create an island near resonance and a saddle-node bifurcation occurs in frequency response. Therefore, blades response may be attracted by the large amplitude stable node in some initial conditions.

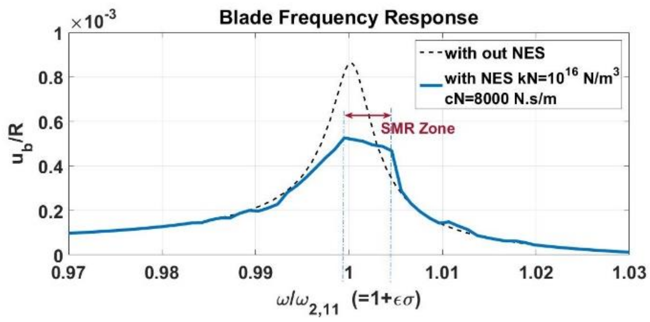

Fig. 4. Blade frequency response with NES (solid line) and without NES (dashed line).

5. Conclusions

Shaft is coupled with disk-blade motion only at zero and first NDs. The optimized NES could reduce the blade vibration amplitude up to 38% and its performance is more sensitive to the nonlinear stiffness than linear damping.

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