Dynamic Response Predictions of Tuned and Mistuned Bladed Discs Using Finite Element Method

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Abstract

The power production of any country plays a vital role in its economy. One such promising power production methods is by means of gas turbines. The effects of random blade mistuning on the dynamics of advanced industrial turbine rotor, using a finite element formulation for tuned and mistuned bladed discs are studied. The technique uses modal data obtained from finite element model to create computationally in expensive models of mistuned bladed disc in a systematic manner. The free vibration responses of the rotor are considered and obtained results are compared with other solutions. More over a simple method is presented for computing natural frequencies of non-integer harmonics, using conventional cyclic symmetry finite element analysis. This procedure enables quantification of frequency veering data relevant to the assessment of mistuning sensitivity (veering curvature). It provides a tool for quantifying structural inter blade coupling in finite element rotor model of a complexity and size. The natural frequencies and displacements of the individual blades are obtained for various nodal diameters. All the frequencies are plotted against different nodal diameters. Also displacements are plotted against blade numbers and significant trends from the results are obtained.

1. Introduction

Modern gas turbines, catastrophic failure occurs, when bladed-disc assemblies fail due to large amplitude of vibrations. To know such failures it is essential to accurately predict the natural frequencies and mode shapes of the assembly in order to avoid resonance in the operation speed range of the rotor. In dynamic analysis of a turbo machinery rotor, one assumes traditionally that the blades are identical. The assumption of cyclic symmetry enables analysts to reduce the computational time considerably by modeling a single sector rather than modeling the entire blade assembly. The primary focus in this project work is to demonstrate mistuned response statistics accurately and efficiently using FEM modeling. In order to be useful for industrial application the formulation should be based on parent finite element models of arbitrary complexity and size [1,2]. The procedure involves a component mode analysis of the rotor using a single blade and the cyclic disc as components. This method represents the first effort to generate reduced order models of turbo machinery rotor directly from finite element models in a systematic fashion. Since the introduction of this technique, several finite element based reduced order modeling technique for mistuned bladed discs are followed using model expansions in terms [3] of component models nominal (tuned) system models with projection of mistuning date [4], or a combination of both approaches in the form of a secondary model analysis reduction techniques (SMART) [5,6]. Some of the more recent techniques exhibit significant improvement over the chosen approach in accuracy and computational efficiency. It is therefore relevant to employ this method to demonstrate the suitability of FEM Model to computer mistuned bladed disc dynamics

2. Material selection and Analytical Formulation

The material selection for gas turbine rotor based on the requirements of specific strength, stiffness and thermal property. The available material of Forged alloy steel of 850A 368-510 (HY 19467) is meets the requirements and minimizes the thermal expansion. The following factors are considered while selecting the material. At high operating temperatures, the material posses good strength high yield strength high creep strength, hot erosive and corrosive resistant, further more it cab castable forgeable, weldable, machinable and no embrittlement during operations.

The outer radius of the rotor is determined by knowing the hoop stress or radial stress of the rotor from equation 1.

$$K_1 \frac{\rho}{g} \omega^2 R^2 = MaximumHoopStress \quad \text{-- (1)}$$

The inner radius for the rotor, assumed the rotor is hollow type, 50mm. The thickness at Rim side by eq. 2

$$\frac{3.27 \times 10^7}{2\Pi 500t} = t_R \quad \text{-- (2)}$$

The thickness of the hub side by eq. 3

$$T_H = \frac{t_R}{e^{\frac{\rho \omega^2 R^2}{2fg}}} \quad \text{-- (3)}$$

Modal analysis is a powerful tool to determine the free and forced vibration of multi degree system. For a two degree freedom system, there are two natural frequencies and each natural frequency has its own unique mode shape. The system will vibrate with harmonic motion only when the amplitudes of the masses vibrate in accordance to one of the natural modes. In a multi degree of freedom system also, the system vibrates with a harmonic motion at a natural frequency when and only when the amplitudes of vibration of the masses in the system satisfy anyone of its mode shapes. However, if the system is disturbed in a general manner not satisfying the mode shape of the system, the harmonic motion or the principal character of the system vibrating at a natural frequency is destroyed and the motion becomes periodic with several harmonic contents in it.

3. Results and Discussions

The modal analysis is carried out to study the dynamic response of tuned and mistuned bladed discs. The focus will be on the effects of structural inter-blade coupling, which governs the transmission of vibration energy between blades through the disc. The industrial rotor illustrated in Fig. 3.1 is the second stage of a Fr-6 turbine rotor used in an advanced gas turbine application. There are 92 blades in the rotor. The tuned final element model is represented by the single sector model in Fig.3.2, using ANSYS 10.0 cyclic symmetry routines. Fig. 3.3 shows the natural frequency and mode shapes for different nodal diameters. It is convenient to describe the mode shapes of a tuned rotor in terms of nodal diameter (nodal lines across the diameter of the disc).

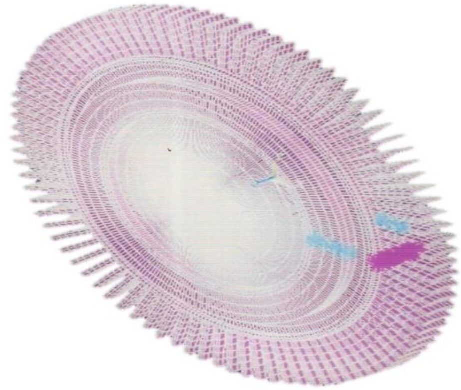

Fig. 3.1: Industrial rotor elemental plot for tuned model

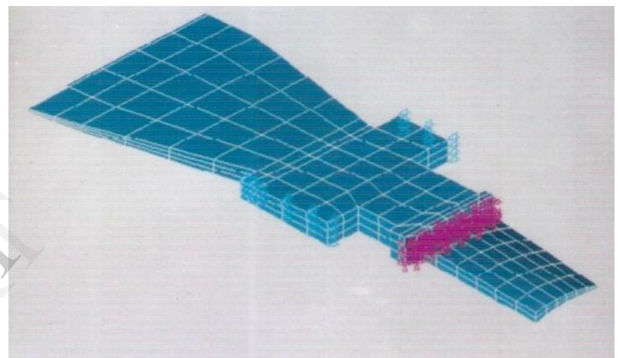

Fig. 3.2: Single sector tune model

Forced and Free response of the rotor for the mistuned model shown in Fig. 3.3 to 3.5, these mistuning parameters are based on experimental natural frequency measurements on a rotor. The mode shapes for a mistuned model at a frequency 159.25 Hz and 0.00379 Hz respectively. The nodal diameter description implies that the mode shapes of the rotor are spatially periodic, which is true for tuned rotors.

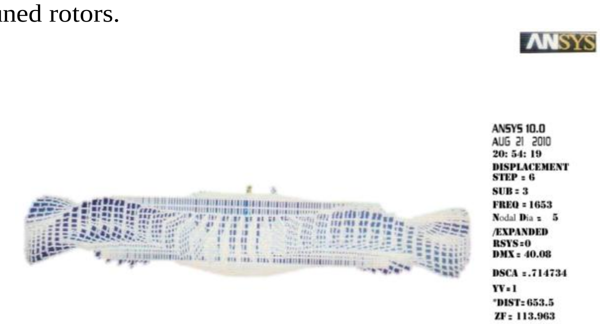

Fig.3.3: Tune model natural frequency and mode shape plot of 5th nodal diameter

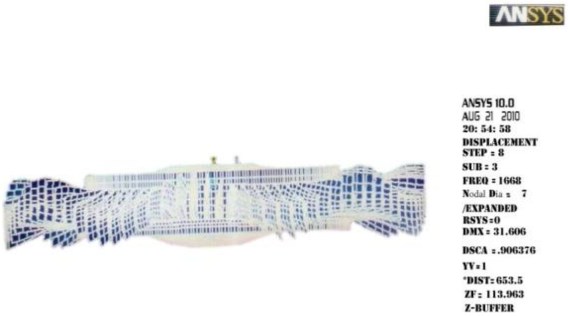

Fig. 3.4: Tuned model natural frequency and mode shape plot of 7th nodal diameter

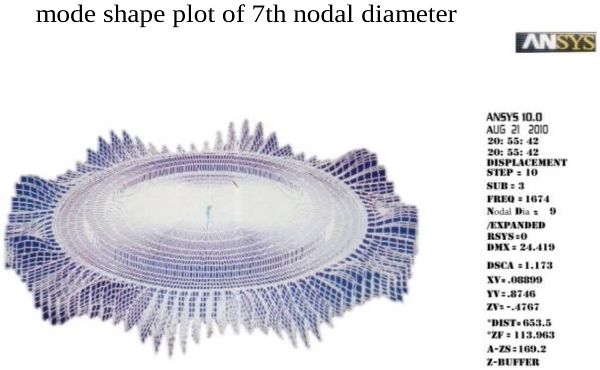

Fig. 3.5: Tuned model natural frequency and mode shape plot of 9th nodal diameter

Based on the frequencies generated to the corresponding nodal diameter the tune disc of the model for the Flexural mode, axial mode and torsional mode are plotted in the below figures.

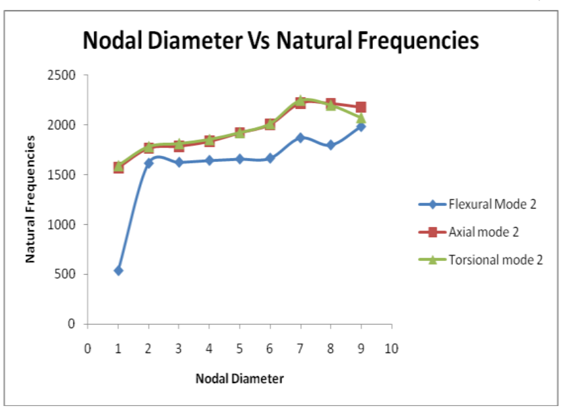

Fig. 3.6 Natural frequencies for different nodal diameters for a tune disc mode1 (ROM)

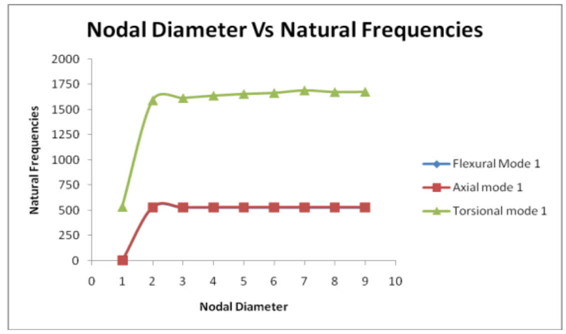

Fig. 3.7 Natural frequencies for different nodal diameters for a tune disc mode2 (ROM)

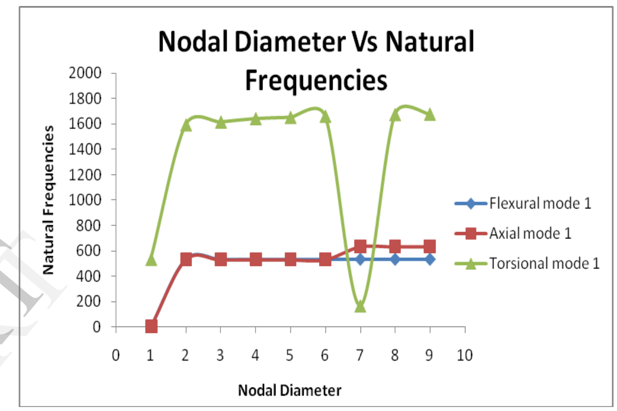

Fig. 3.8 Natural frequencies for different nodal diameters for a tune disc mode 1 (FEM)

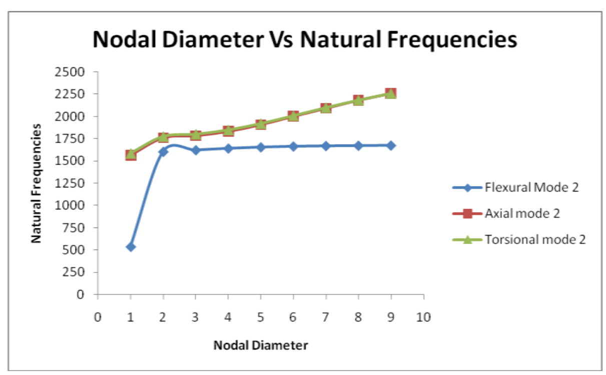

Fig. 3.9 Natural frequencies for different nodal diameters for a tune disc mode 2 (FEM)

4. Conclusions

The model analysis is carried out for dynamic response of tuned and mistuned bladed discs. The studies are, with the appropriate levels of mistuning and it's inter blade coupling presents, The mistuned responses can exceed tuned response levels by about 200 percent. The existence of localization phenomenon is observed in industrial turbo machinery rotor, which causes significant amplitude increase in the forced response of the rotor. The analysis of Eigen frequency veering are used to determine the inter blade coupling strength for a bladed disc. It is found that the analysis makes possible to predict the maximum blade stresses due to mistuning, which is crucial for assessing the safety of a rotor design. As the number of nodal diameters increases the disc stiffens rapidly, and the disc dominated modes are presenting the significance of disc stiffness by increasing the number of nodal diameters. The maximum displacements are occurring for blade numbers 3 & 49 at a natural frequency of 159.25 Hz, due to the resonance which causes damage of the rotor.

5. References

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