The Journal of CPRI,
Vol. 8, No. 3, September 2012 pp. 199–206

Critical Speed Analysis of a Single Stage Impulse Type High-Speed Steam Turbine Rotor Disc

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Resonance is a common thread that runs through almost every branch of engineering. Yet, this phenomenon often goes unnoticed, silently resulting in inconveniences, such as causing a bridge to collapse or a helicopter to fly apart, to name a few. It is, therefore, of utmost importance to avert resonance, for which determining the frequency of the system becomes indispensible. In complex rotating structures, as one considered in this paper, theoretical determination of frequency is as difficult and laborious as a task can be. Finite element analysis has proven to be an effective tool to handle such a task. Resonant vibrations incited by the running speed harmonic excitations, steam impinging frequency, engine order excitations are fundamental causes for failure of turbine components. The mainstream discipline that is encompassed by this paper is the modal analysis. Modal analysis is performed to estimate the critical speeds and study the mode shapes of the bladed rotor disc under prestressed condition.

Keywords: *Resonance, Steam turbines, Blade vibration, Disc vibration, Finite Element Analysis (FEA).*

1.0 INTRODUCTION

Steam turbines are among the most suitable and viable prime movers for power generation. Steam turbines are mainly used to drive machines, pumps, compressors and electrical generators of various capacities. In an impulse type steam turbine, the blades are set on the rim of the revolving wheel mounted on a central shaft. Steam passing through a fixed nozzle passes over the curved blades and absorbs the kinetic energy of the expanded steam thereby turning the wheel and shaft (turbine rotor) on which they are mounted.

With phenomenal increase in power consumption and consequently enormous demand for power generation, plants with smaller capacities are going to be most commonly used sources in the coming years. In spite of several advantages, these turbine rotors [1] seem to fail mainly due to resonant stresses developed at the critical speeds of the rotor. Also, blade failures are caused by resonant conditions triggered by the coincidence of the blade, natural frequencies with the nozzle excitations. In this context, modal analysis on turbine rotors during its operation has acquired significance with the main objective of preventing failure of turbines due to resonant stresses.

Modal analysis of turbine rotor assembly may help in a reliable design. Further, this may reduce high factor of safety employed in the industry and

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contribute to reduction in weight and enhanced performance.

In this context, the present paper presents an attempt that has been made to carr yout systematically a detailed modal analysis on a bladed turbine rotor for various speeds in order to study the critical speeds and mode shapes of the bladed rotor disc.

2.0 VIBRATIONS

2.1 Blade Vibration

The motion of a body about its equilibrium position, which repeats itself after a certain interval of time, is known as vibration. Vibration is a physical phenomenon occurring in bodies due to their over-elastic properties or due to various external sources, which cause them to continuously oscillate about their mean position. Blade failures due to fatigue are predominantly due to vibrations, caused by resonant conditions triggered by the coincidence of the blade natural frequencies with the nozzle excitations. The causes of blade vibrations are loose fittings, misalignment, bent shaft, eccentric journals, insufficient vibration isolation and defective bearing, and the factors affecting blade vibrations are blade taper, asymmetry of blade cross-section, centrifugal forces and blade setting angle.

2.2 Disc Vibration

Various periodically varying forces are conducive to exciting some of the natural frequencies of the disc causing resonant vibrations in the disc. Causes of disc vibrations are force transmission from blade to the disc, non-uniformity of steam flow around the disc, inertia forces of unbalanced residual masses, running the turbine close to critical speeds and steam pressure pulsation.

3.0 FINITE ELEMENT TECHNIQUES

Finite Element Analysis (FEA) is a numerical technique that can be used to approximate the structural dynamic characteristics of vibrating mechanical systems. This technique can be used for structural dynamic studies of existing equipment or to evaluate the dynamic characteristics of machines and structures prior to fabrication. It is important to realize that the finite element method [2] is an approximate numerical technique. The accuracy of a solution obtained by FEA depends on several factors, the most important of which are:

- degree of refinement of the finite element mesh.
- appropriateness of the finite element types used to model a machine/structure.
- boundary conditions used at the limits of the finite element model.

Since a finite element model provides a more detailed mathematical description of a mechanical system than an experimental model, it is well -suited for structural dynamic modification studies.

An experimental model is usually sufficient to serve as a valuable root-cause analysis tool. However, the finite element model [3] is required to propose and incorporate design modifications directed at removing resonance excitation problems.

The finite element method provides the vibration analyst with a numerical technique to evaluate

- natural frequencies and mode shapes to ensure that equipment does not operate at or near any resonant frequencies.
- stiffness of supporting structures/ foundations.
- areas of structural weakness or probable resonances which can be addressed during the design stage.

The finite element method [4] can also be used to estimate stress levels in rotating equipment during operation, which includes both centrifugal stresses and stresses due to the application of dynamic loading, such as unbalanced forces.

4.0 METHODOLOGY

4.1 Modal Analysis

Modal analysis basically determines the natural frequencies and the corresponding mode shapes of a structure or a machine component while it is being designed. Modal Analysis is essential in anticipating the vibration levels or determining what actions to be taken if the vibration is to be controlled. The primary objective of modal analysis is to assist the designer to design [5,6] the structure and determine its weight, stiffness and frequency of operation in such a manner so as to minimize any vibrations by precluding resonant vibration.

4.2 Prestressed Modal Analysis

In prestressed modal analysis, the modal response of a structure is analyzed when a mechanical stress is applied to the structure. This can be performed to calculate the frequencies and mode shapes of a prestresssed structure, such as spinning turbine disc.

4.3 Model Considered for Analysis

This is an impact type steam turbine. The disc has 72 shrouded blades arrayed around its periphery as shown in Figure 1. The turbine casing has 10 nozzles, from which steam impinges on the blade surfaces tangentially, and torque is developed.

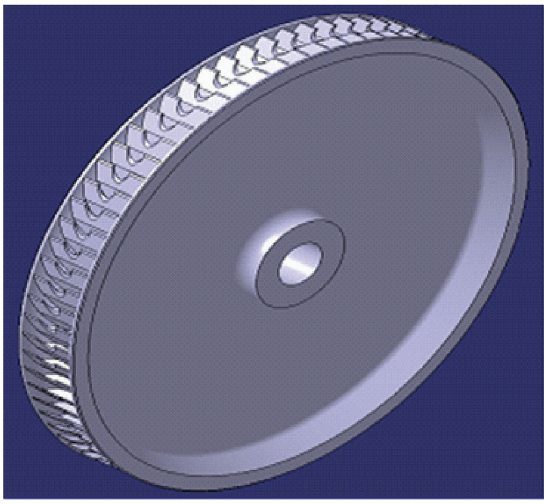

FIG. 1 BLADED DISC OF THE STEAM TURBINE ROTOR

4.4 Geometric Modeling

A 100 cyclic symmetry model is used for the analysis instead of the entire model, in order to save time, memory requirements and ANSYS computational resources. This model enshrouds two blades, Figure 2. This model comprises all the nodes, elements, material properties, real constants, boundary conditions and other features that are used to represent the physical system.

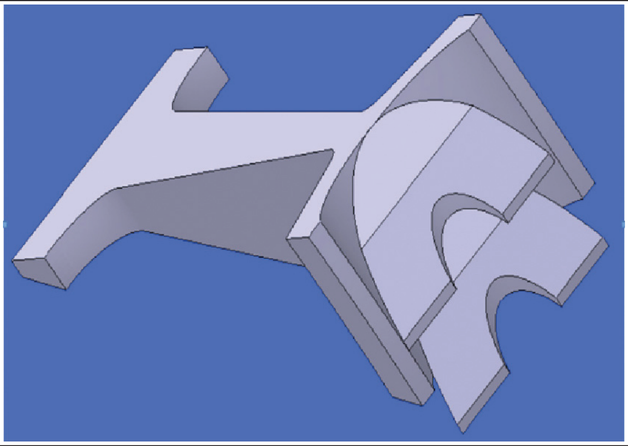

FIG. 2 10º SECTOR OF BLADED DISC OF THE STEAM TURBINE ROTOR

4.5 Finite Element Modeling

The sector shown in Figure 2 was meshed using tetrahedron brick elements, and due to the complexity of geometry, the component was free meshed as shown in Figure 3. Then, the cyclic

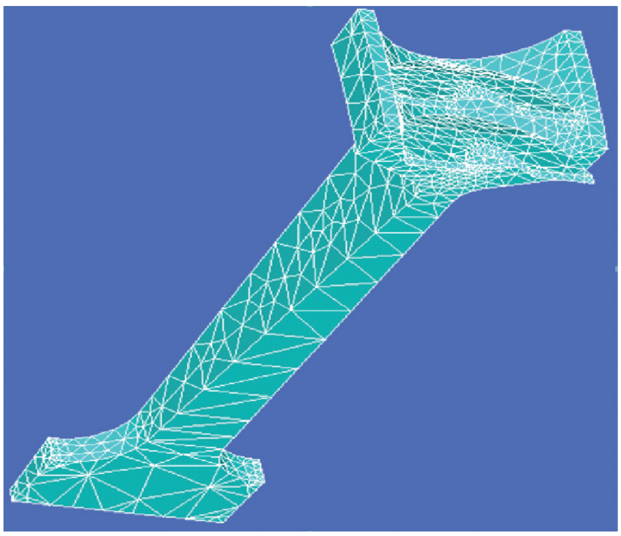

FIG. 3 MESHED 10º SECTOR OF BLADED DISC

symmetry model option was activated to expand the 100 sector 36 times about the z – axis to obtain a meshed 3600 model.

A cyclic symmetry analysis requires that a single sector be modeled, called the basic sector. A proper basic sector represents one part of a pattern, so that if repeated N times in cylindrical coordinate space, yields the complete model.

The detailed procedure of carrying out prestressed modal analysis in ANSYS has been shown in Figure 4.

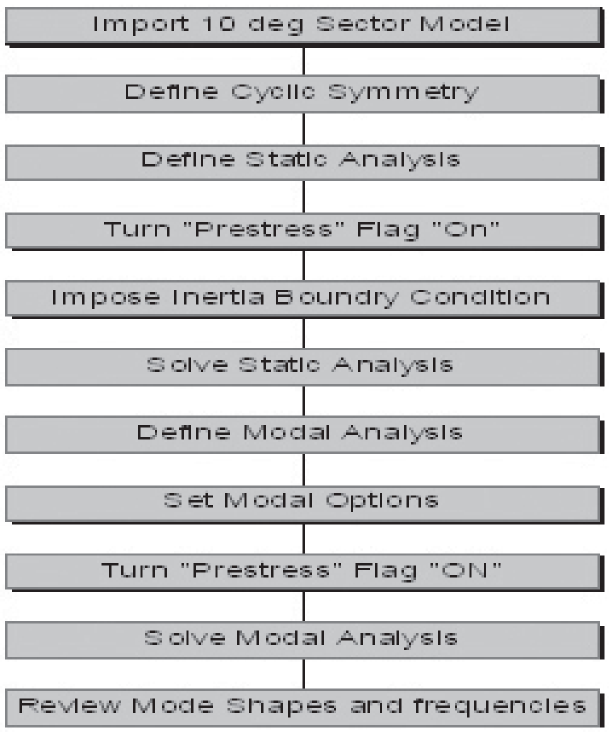

FIG 4 PRESTRESSED MODAL ANALYSIS PROCEDURES IN ANSYS

4.6 Campbell Diagram

A common way to identify forced response regions of a bladed disc is the Campbell diagram, which is a key plot in the dynamic design process [7]. It is a plot of Frequency vs. Rotational speed. It is essential in eliminating the critical speeds that could be encountered during operation for different vibration modes of the component.

Resonance is caused when the nozzle passing frequency coincides with one of the natural frequencies of the bladed disc. Nozzle passing frequency is defined as the number of times steam from the nozzles impinges on a blade per second. It is a function of the rotor running speed and the number of nozzles. NPF = t N/60, where t is the number of nozzles=10, N is the rotor running speed.

Engine order excitations [8] are caused by non uniformities on the stator blade row. The excitations can induce vibrations in fundamental blade modes. These excitation frequencies are computed at different disc speeds (N) using a generalized emperical formula EOF = nN/60 Hz, where n=1, 2, 3, etc.

The critical speeds for the bladed disc can be evaluated by extrapolating the points of intersection of the nozzle passing frequency and the engine order excitations lines with the eigen-frequency lines onto the speed axis.

When the frequency of external excitation (engine order and nozzle passing) coincides with any of the natural frequencies, the amplitude of vibration increases excessively, and this condition of the structure is called resonance.

A typical Campbell diagram [9], showing eigen-frequency lines of the bladed disc along with engine order and nozzle passing excitation lines is shown in Figure 5. The black spots in the Figure represent resonance due to coincidence of the nozzle passing and engine order frequencies with the natural frequencies of the bladed disc.

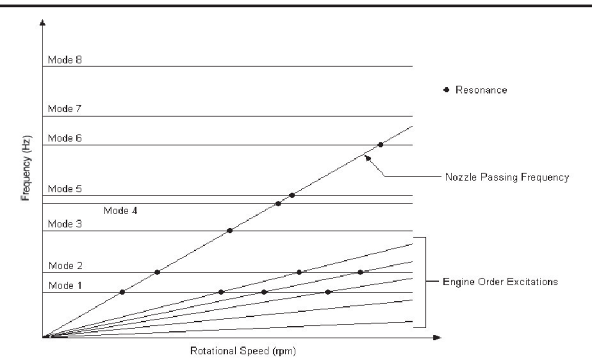

FIG. 5 A TYPICAL CAMPBELL DIAGRAM-SHOWING POINTS OF RESONANCE

5.0 RESULTS AND DISCUSSION

5.1 Prestressed Modal Analysis

As explained in the previous section, a detailed analysis has been carried out on a steam turbine rotor assembly to study the critical speeds and mode shapes of the bladed rotor disc under prestressed condition.

Table 1 presents the variation of natural frequency with different rotational speeds of the rotor from 0 to 50000 rpm in steps of 5000 rpm. Also, listed are the nozzle passing frequency (npf) extracted from the rotational speeds (rpmx10/60) and first five engine order excitations extracted from nozzle passing frequency (npfx1/10, npfx2/10, npfx3/10, etc).

Figure 6 presents the Campbell diagram of natural frequency (Hz) with rotational speed for the first four modes of natural frequencies, nozzle passing frequency, and the first five engine order excitations.

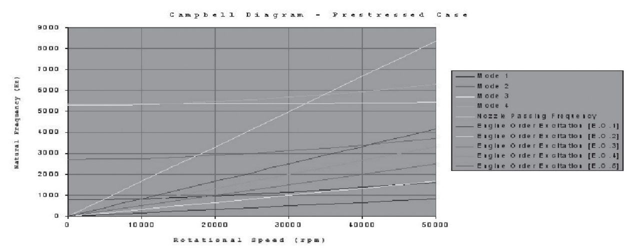

FIG. 6 CAMPBELL DIAGRAM: PRESTRESSED MODAL ANALYSIS

TABLE 1

TABLE OF PRESTRESSED MODAL ANALYSIS RESULTS

Rotational speed (rpm) 0	0	5000	10000	15000	20000	25000	30000	35000	40000	45000	50000
Natural Frequencies (Hi)	785.11	797.76	83454	802.42	967.56	1056.2	1155.2	1261.0	1374.6	1491.8	1612.5
	2720.6	2732.7	2768.7	2827.4	2907.3	3006 2	3121.9	3252.2	3395	3548.2	3710.1
	5291.3	5302.3	5335.2	5365.5	5370.1	5376	5383.2	5391.7	5401.6	5412.6	5425
	5359.6	5360.2	5362.2	5389.6	5464.4	5558.9	5671.7	5801.6	5947.1	6106.8	6279.4
	9319 7	9333 7	9375 5	9444.6	9540.3	9661 8	9807 9	9977 2	10168	10380	10611
	12034	12035	12040	12049	12060	120’5	12093	12115	12139	1 2167	1 2197
	14513	14515	14519	14525	14534	14544	14554	14566	14577	14587	14596
	14737	14752	147S7	14872	14976	15108	15265	15440	15555	15601	15642
	15442	15444	15451	15462	15478	15493	15525	15565	15692	15916	16170
	10400	10403	10413	10427	10440	10408	16491	16514	16535	16553	16570
Nozzle Passing Frequency (Hz) 0	0	833.33	1666.87	2500	3333.33	4166.67	5000	5833.33	6666.67	7500	8333.33
Engine Order Excitations (Hz) 0	0	83.33	166.67	250	333.33	416.37	500	583.33	668.67	750	833.33
	0	166.67	333.33	500	060.67	033.33	1000	1106.07	1333.33	1500	1066.67
	0	250	500	750	1000	1250	1500	1750	2000	2250	2500
	0	333.33	666.67	1000	1333.33	1666.67	2000	2333.33	2666.67	3000	3333.33
	0	416.67	833.33	1250	1666.67	2083.33	2500	2016.67	3333.33	3750	4166.67

Figure 7 presents displacement contours of disc at mode 1.

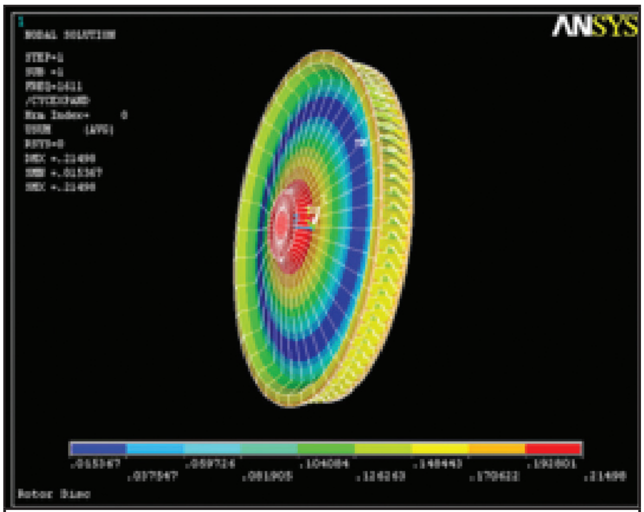

FIG. 7 DISPLACEMENT CONTOUR PLOT FOR MODE 1

5.2 Inference and Interpretation of Results

Since the nozzle passing frequency excites only the first four modes of the bladed disc in the speed range of 0–50000 rpm, only the first four modes were considered for analysis. Also, since the blade geometry is such that it is not only shrouded along its full length by the disc, but also substantially inferior in size in comparison with the disc, blade resonances occur at very high frequencies, which are very unlikely to be excited by either nozzle passing frequency or engine order frequencies, as the rotor is never run at high speeds.

From Figure 6 and Table 2, it can be seen that the first critical speed, i.e. the excitation of the first mode can occur at any of these speeds due to their respective sources of excitation.

TABLE 2

LISTING OF FIRST CRITICAL SPEED WITH RESPECTIVE EXCITATION SOURCE

First Critical	Excitation source
4500	Nozzle
10000	Engine Order
13000	Engine Order
20000	Engine order
45000	Engine Order

The second critical speed, i.e. the excitation of the second mode can occur at any of these speeds due to their respective sources of excitation and is shown in Table 3.

TABLE 3

LISTING OF SECOND CRITICAL SPEED WITH RESPECTIVE EXCITATION SOURCE

second critical speed (rpsi)	Excitation source
17500	Nozzle Passing
41500	Engine Order 5

The third critical speed, i.e. the excitation of the third mode occurs at 32500 rpm due to nozzle passing frequency excitation.

The fourth critical speed, i.e., the excitation of the fourth mode occurs at 35000 rpm due to nozzle passing frequency excitation.

It is observed from the Campbell diagram [10] for the prestressed modal analysis of Figure 6 that spin stiffening behavior is exhibited by the rotor disc. As the rotational speed is increased from 0 to 50000 rpm, the natural frequencies are observed to increase significantly for all the modes.

This is the result of the structure being stiffened due to static prestressing as the spinning speed is increased.

6.0 CONCLUSIONS

A novel method that is based on ANSYS Modal analysis procedures has been presented for estimating resonant speeds that could be encountered during the operation of turbine rotors for different vibration modes.

It is found to be very effective in finding forced response regions of the bladed disc of small-and medium-capacity steam turbine rotors up to 2.5 MW.

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