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Failure Analysis of Last Stage Low Pressure Steam Turbine Blade

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Abstract*: Objective of this work is to study the failure analysis of low pressure turbine last stage blades; this can be ensured by carrying out the modal analysis of the steam turbine blades. Steam turbines are subject to the number of starts and stops during their useful life. This means that it is subject to repetitive cyclic loading conditions that cause fatigue failure of the moving blades. It is important to design the blades of the steam turbine so that the natural frequencies of the blades are keep away from the harmonic excitation of the excitation forces. The modal analysis of the blade of the steam turbine is done with the help of FEM techniques to generate different modes of operation at different frequencies. From Modal analysis of blade and validation with design data through generation of Campbell Diagram. And also analyse the deflection of the blade to estimate the dynamic behaviour and stresses occurring.
Keywords: FEM, Modal Analysis, Natural frequencies, Campbell diagram, Steam Turbine blade

I. INTRODUCTION

Steam turbine blades are subjected to centrifugal loads from high rotational speeds apart from steam bending forces. High cycle fatigue plays a significant role in many turbine blade failures. During operation, periodic variations in the steam force occur at frequencies corresponding to the operating speed and harmonics and cause the bladed disk to vibrate. The amplitude of these vibrations depends in part of the natural frequencies of the bladed disk to the forcing frequency. Large amplitude vibration can occur when the forcing frequency approaches or becomes resonant with the natural frequency of the blades. Dynamic stresses related with near resonant or resonant vibration produce high cycle fatigue failure and can initiate and propagate cracks very quickly. Steam turbine manufacturers typically design and manufacture blades with tolerable margins between the forcing frequencies and the fundamental natural frequencies to avoid resonance.

A rotating turbine blade is the component, which transforms the energy of the flowing fluid into mechanical energy. Thus the consistency of these blades is very important for the successful operation of a turbine. Metallurgical examinations of failed blades show that almost all the failures can be attributed to the fatigue of metal. Blade failures due to fatigue are predominately vibration related.

It has been recommended that these high stresses in a particular blade might be subjective by these factors. One factor is the non-uniform circumferential fluid flow and gear meshing. These may excite a specific blade or packet of blades to vibrate at high stress levels leading to fatigue failure. The second factor is the flexibility of the disc. Due to this flexibility, forces may be conveyed between any blade and its neighbors. This inter-blade coupling may thus excite a particular blade or a group of blades thus causing failures due to extensive fatigue. The last factor is that blades are, in practice, slightly different, with the significance that their natural frequencies differ with a specific rotor speed, some of these blades may be excited, with consequence that they fail due to fatigue.

A. Factors Effecting for Blade Failure

1) High temperature
2) High stresses
3) High vibrations
4) Material used
5) Environmental effects
6) Blade design
7) Manufacturing Effects
8) Stage environment & Operating conditions
9) Maintenance effect

B. Generally Effecting Factors for Failure of Blades

1) Corrosion failure
2) Corrosion failure
3) Stress corrosion Crackin
4) Pitting and erosion-corrosio
5) Fretting fatigue failure
6) Fatigue creep failure

Steam turbine blades are one of the most critical components in power plants, the blade is an important component of the turbine, which takes the impulse continuous from the steam jet and converts into the driving force. In any steam turbine, the last stage blades are more susceptible to damage due to periodic loading conditions.

The basic design concern the failure stresses that have occurred by the fluctuating forces is to avoid or to minimize. Since these forces are cyclic, it should be careful that these harmonics coincide with any of the natural frequencies of the blades. Generally, a Campbell diagram is plot to determine this and estimate the dynamic behavior of a blade and stresses occur in the blade due to centrifugal forces. From modal analysis, the natural frequencies and the mode shapes at different speeds were obtained and Campbell diagram were plotted.

II. LITERATURE SURVEY

Although some significant advances have been made during past 25 years in the design technology of the blade from vibration point of view, blade failures still continue to take place, and hence efforts are still on to understand in totally the blade dynamics which has resulted in large number of technical papers. Some of the important work is presented here.

In this survey emphasis is placed on papers dealing with general structural analysis of blade by analytical modelling, blade excitation and its response and experimental evaluation of turbine blades.

A. Analytical Modeling – General

There has been a continuing progress in the analytical modelling for the determination of natural frequencies of the system comprising of a set of blades mounted on the bladed disk. The early attempts to model the blade as a beam element have progressively led to a more comprehensive finite element representation. This finite element representation of real blade profile becomes necessary especially when plate or shell type of vibratory modes is induced.Udai Kumar Singh, K.Ch. Peraiah, J.S. Rao, [1] Estimate the dynamic stresses in last stage steam turbine blades under reverse flow conditions, last stage steam turbine blades are known to suffer high alternating stresses under low volumetric flows of steam. The low volumetric flows results in a reverse flow condition and the resulting response is unstable vibration that occurs predominantly at fundamental mode of the blade. Besides the high alternating stresses, the last stage blades are also subjected to severe centrifugal load stress that when combined with the alternating stress is responsible for fatigue failures.

Ernst Plesiutschnig, Patrick Fritzl, Norbert Enzinger and Christ of Sommitsch, [2] cracks were analyzed at the root of the third blade row of low-pressure steam turbine blades of different natural frequencies. The root cause of the fatigue crack initiation was pitting corrosion of the forged ferritic/martensitic X20Cr13 material. Metallographic investigations, finite element analysis and fracture mechanics analysis combined with experimental data from the literature are used to evaluate crack propagating stresses to discuss the operating conditions. The calculations show that corrosion pits at the root of the turbine blade increase the local stresses above yield strength. Excitation of natural frequencies by changing the rotor speed is not responsible for the crack propagation. The centrifugal load and superimposed bending load caused by unsteady steam forces are responsible for the crack propagation.

Akash Shukla, S.P. Harsha, [3] made comparative study of the model analysis of steam turbine blade with the analytical method. In any steam turbine the last stage blades are most prone for the failure due to severe dynamic conditions of loading. First the modal analysis of last stage blade is carried out with the help of FEM techniques and the natural frequency calculated. Finally the variation in natural frequency and mode shapes for cracked and un-cracked blade is studied.

B. Blade Excitation and Response

The major source of blade excitation arises out of the interaction between the moving blade rows and the stationary blade row. A logical approach towards the design of turbo machine blade is to study the nature of these excitation forces and analyse the dynamic stresses. Many researchers have worked to develop the basic theories of isolated aerofoil and have studied the flow interference in a turbo machinery stage.

S.P. Harsha, Akash Shukla [4] Find the variation in vibration response of blades with the crack size of fir tree root free standing blades. Generation of crack in blade root causes the loss of stiffness in the vicinity of blade root. It results in shifting of natural frequencies and redistribution of dynamic and static stress in the blade root which may cause failure of the blade. A 3D finite element model of a blade and its fir-tree roots has been analyzed. Results of FEM study are validated by experimental results of cracked blades. Finally the variation in natural frequency and mode shapes for different sizes cracked blades are studied. Pavel, Polach, [5] a suitability of the bladed disk design regarding the possibility of the resonant vibration excitation can be assessed on the basis of several approaches. Most information concerning the evaluation of the bladed disk design suitability is provided in the SAFE diagram but the possibility of exciting the action wheel resonant vibration can also be evaluated from the Campbell diagram. Further criterion is the assessment of the design suitability on the basis of a critical speed of the bladed disk. It especially causes breakdowns of relatively flexible disks. The suitability of the bladed disk with the blades of the ZN340-2 type was evaluated. Zdzislaw Mazur, RafealGarcia_Illescas and Jorge Aguirre-Romano, [6] a last stage turbine blades failure was experienced in two units of 660MW. These units have one high-pressure turbine and two tandem-compound low-pressure turbines with 44-in. last-stage blades. The blades that failed were in a low pressure (LP) turbine connected to the high pressure turbine (LP1) and in LP turbine connected to the generator (LP2). The investigation included a metallographic analysis of the cracked blades, natural frequency test and analysis, blade stress analysis, unit's operation parameters and history of events analysis, fracture mechanics and crack propagation analysis. This paper provides an overview of this failure investigation, which led to the identification of the blades torsional vibrations near 120Hz and some operation periods with low load low vacuum as the primary contribution to the observed failure.

C. Experimental Evaluation

Rotating blades have been recognized as one major cause of failure in many turbines and jet engines. They are, usually rotating at high speeds, interacting with the erosive environment, have complicated shapes, and undergo severe dynamic and thermal loadings. These operating conditions expose blades to many vibration excitation mechanisms and at the same time make the vibration measurement process of blades a very complicated task.

Experiments are done to evaluate the frequencies of blades.

Akash Shukla, S. P. Harsha, [7] comparative study of the model analysis of steam turbine blade Experimental Method. In any steam turbine the last stage blades are most prone for the failure due to severe dynamic conditions of loading. NFT of these blades in lab conditions is mandatory to detune these blades.

First the modal analysis of last stage blade is carried out with the help of FEM techniques then the blade is tested in NFT lab and the natural frequency calculated by both the methods is compared. After the validation of model through experimental results the similar study is also carried out for a cracked blade. Finally the variation in natural frequency and mode shapes for cracked and un-cracked blade is studied. R.S. Mohan, A. Sarkar, A.S. Sekhar, [8] Blade failure is a common problem of a steam turbine and its failure in-service results in safety risks, repair cost and nonoperational revenue losses.

Thus, there liability of these blades is very important for the successful operation of a steam turbine. Dynamic analysis of a steam turbine blade in computational environment is carried out in the present work. In order to gain physical insight into the flexural dynamics of such turbine blades with the inclusion of the rotor dynamic effect, the turbine blade was approximated as a twisted cantilever beam with an asymmetric aerofoil cross-section fixed on a rigid rotor disk. Methods to validate the computational procedures for cantilever beam were established. Similar computational procedures were leveraged for the turbine blade. Critical speeds were obtained for different excitations.

III. THEORY &METHODOLOGY

A. Natural Frequency and Mode Shape

Natural frequency is the frequency at which an object vibrates when excited by a force, such as a sharp blow from a hammer. At this frequency, the structure offers the least resistance to a force and if left uncontrolled, failure can occur. Mode shape is the way in which the object deflects at this frequency. An example of natural frequency and mode shape is given in the case of a guitar string. When struck, the string vibrates at a certain frequency and attains deflection shape. The frequency can be noted by the pitch coming from the string. Different string geometries lead to different natural frequencies or notes. By nature of its structure, a turbine blade has many natural frequencies and mode shapes. These frequencies and mode shapes are somewhat further complicated by the use of shroud to connect group of blades together.

B. Vibratory Forces

Some alternating forces must exist to excite a structure to vibrate. These forces have inherent frequencies and shapes just as bladed disk do. In a steam turbine, the most common sources of excitation are nozzle passing frequencies and running speed harmonics.

Running speed harmonics occur due to interruptions in the fluid flow path. Frequencies of running speed are multiples of rotor operating speed. For example, the tenth harmonic of running speed would be a force that occurs ten times for every revolution of the wheel. For example, a turbine rotor running at a speed of 3600 RPM (60 cycles/sec or Hz) would have running speed harmonics occurring at 120HZ, 180Hz, 240HZ…

C. Resonance

Each blade on a rotating turbine disk experiences a dynamic force when it rotates through a non-uniform flow from stationary vanes. The dynamic response (e.g. stress, displacements, etc.) levels experienced by the bladed depend on:

1) The natural frequencies of the bladed disk and tier associated mode shapes.
2) The frequency, the shape and the magnitude of the dynamic force which are ion of the turbine speed, number of stationary vanes and their location around the annulus and/or the number of interruptions in the flow passage e.g. struts and their location around the annulus.
3) The energy dissipating properties called damping – provided by blade material, frictional slip between joints, aerodynamic damping from steam, etc.

A turbine bladed disk may get into a state of vibration where the energy build up is a maxima in its response (stress, displacement, etc.) and minima in its resistance to the exciting force. This condition is called a state of 'RESONANCE'. There are two simultaneous for the energy built up per cycle of vibration to be a maximum.

These conditions areThus for a resonance to occur, both of the above conditions must be met.

D. Static Analysis of the Blade

Most of the mechanical and structural engineers are familiar with structural static analysis. It is used to determine the displacement, stress, strains and forces that occur in a structure or component as result of applied loads.

[K]{q}= [F]

Where [K] = Structural stiffness.
{q} = Nodal displacement.
[F] = Load matrix.

In the solution phase we really end up with governing equations for each element. By solving these equations at each node, we obtain the degrees of freedom, which would give the approximate behavior of complete model.

Von Misesstressσ_v, or simply Mises stress, is a scalar function of the components of the stress tensor that gives an appreciation of the overall 'magnitude' of the tensor. This allows the onset and amount of the plastic deformation under triaxial loading to be predicted from the results of a simple uniaxial tensile test. It is most applicable to ductile materials.

Plastic deformation, or yielding, initiates when the mises stress reaches the yield stress in uniaxial tension and, for hardening materials, will continue provided the mises stress is equal to the current yield stress and tending to increase. Mises stress can then be used to predict failure by ductile tearing. It is not appropriate for failure by crack propagation or fatigue, which depend on the maximum principal stress.

In 3-d, the Mises stress can be expressed as:

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}}$$

Whereσ_1,σ_2, $\sigma_3$are the principal stresses.

Or, in terms of a local coordinate system:

$$\sigma_v = \frac{1}{\sqrt{2}}\sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)}$$

Structural components can be determined to fail by various modes determined by, deflection, natural frequency, stress, or strain. Strain or stress failure criteria are different depending on whether they considered as brittle material or ductile materials. Most finite element systems, default to assuming a ductile material and display the distortional energy failure theory which is usually called the von Mises stress.

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E. Modal Analysis of the Blade

Modal analysis is the process of determining the modal parameters, which are then sufficient for formulating a mathematical dynamic model. Modal analysis is used to determine the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component while it is being designed. Modal analysis can be accomplished through experimental techniques. It is the most common method for characterization of the dynamic properties of a mechanical system. It is also a starting point for another, more detailed, dynamic analysis, such as transient dynamic analysis, a harmonic response analysis, or spectrum analysis. The modal parameters are the modal frequency, the modal damping and the mode shape.

Because of the importance of natural frequencies and mode shapes in the design of a structure for dynamic loading conditions, the modal analysis is used to determine the same. Modal analysis can also be used to do analysis on a pre-stressed structure, such as spinning of the turbine blade. Modal analysis is a linear analysis. Any non-linearity's, such as plasticity and contact (gap) elements, are ignored even if they are defined.

The procedure for modal analysis consists of four main steps:

1) Building the model.
2) Applying loads and obtains the solution.
3) Expand the modes.
4) Review the results.

The modal analysis procedure is done to the steam turbine blade for obtaining mode shapes and natural frequencies. The modal analysis is carried out for different RPM to obtain different natural frequencies. Through these natural frequencies the Campbell diagram should be plotted.

F. Modelling of blade

Solid modelling is the first step for doing any analysis and testing and it gives physical picture for new products. FEM models can be easily created from solid models, by the process of meshing. Here we will consider the application of ANSYS software to model a low-pressure turbine blade with its root so as to generate macro.

Fig: 1 Last stage low pressure turbine blade

G. Governing Equation for Single Blade Vibration

1) Assumptions
a) The structure has constant stiffness and mass effects.
b) There is no damping.
c) The structure has no time varying forces, displacements, pressure or temperature applied.

The model considered as cantilever and its simple form is shown in figure below.

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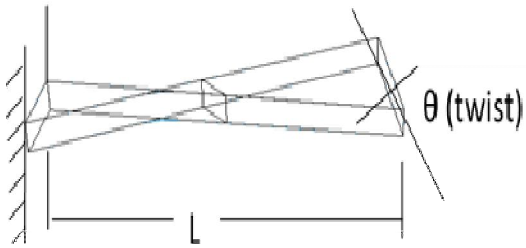


Fig: 2 Blade model consider as cantilever its simple form

At fixed end *u* (longitudinal), *v* and *w* (transverse) are taken as zero. At free end the longitudinal (i.e., *u* =0) displacement is taken as zero. It is implied that there are no longitudinal vibrations. Only transverse and torsional displacements are considered.
The forms of equations for motion are
$[M]\ddot{X} + [K]X = 0$ and
$[\,I\,]\ddot{\Theta} + [K]\Theta = 0$.
These equations are solved in Ansys, and it gives the Eigen values, which are nothing but the natural frequencies.

H. Meshing for Blade

The meshing of the blade is carried out in Hypermesh.

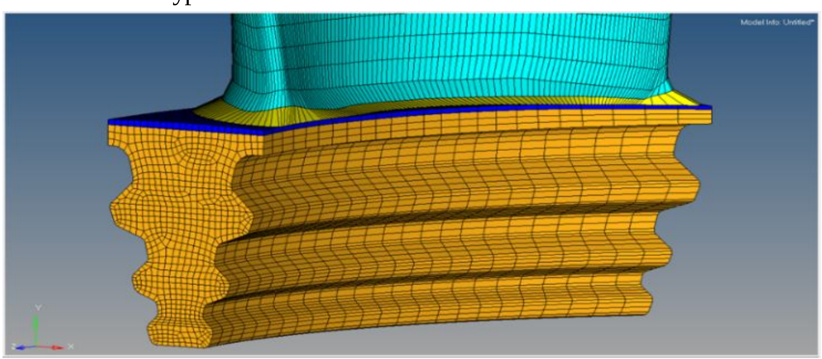


Fig: 3 Meshed blade model with root

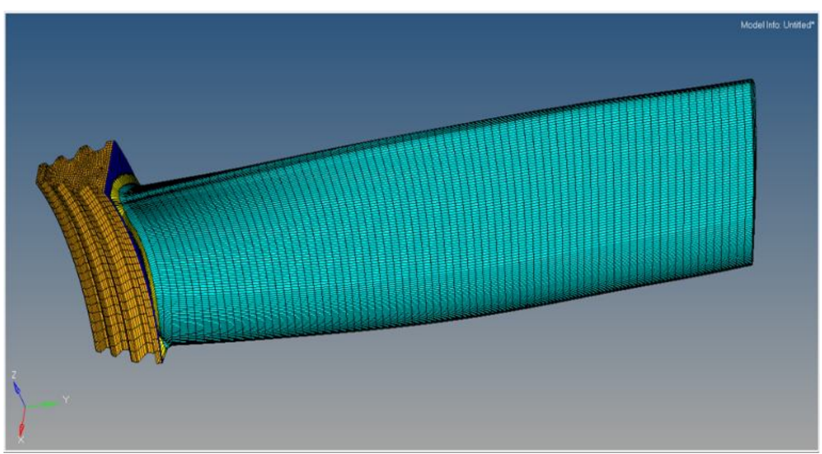


Fig: 4 Hexa mesh of blade in Hypermesh

Total model is divided in 46764 elements which in turn computed the results at 57519 nodes.

I. Analysis for Blade

The Analysis of blade is carried out in Ansys.
Importing the meshed blade model from Hypermesh to Ansys.

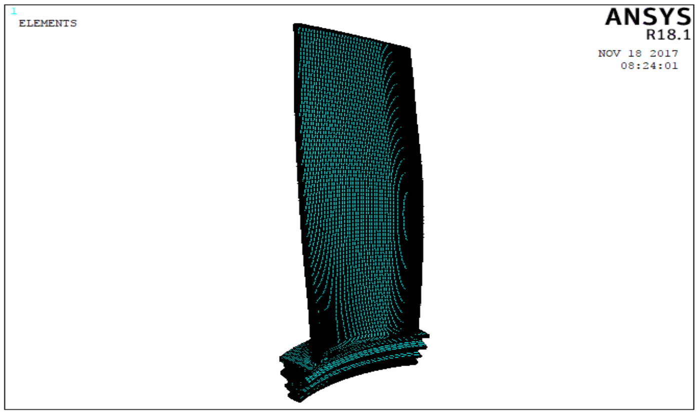


Fig: 5 Blade model in Ansys with elements

J. Boundary Conditions

The Boundary Conditions for the blade are loads means centrifugal forces because it is a self-weighted. The various types of the boundary conditions applied here are

1) Load - centrifugal force
2) Constraint equation between nodes & elements of Blade root top surface & aerofoil bottom surface
3) Displacement constraint on nodes in axial direction
4) Displacement constraint on nodes in 6 contact surface normal
5) For blade tip Analysis Displacement constraint on nodes in axial direction

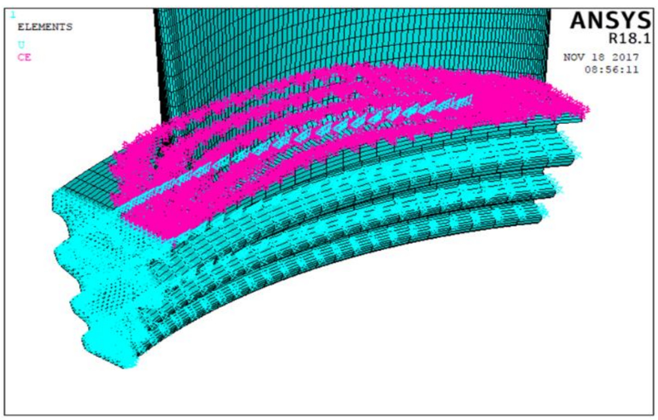


Fig: 6 Blade model with boundary conditions

The single blade with its fir-tree type root is considered for the stress analysis to obtain the stress distribution in the blade as well as the root. The stress analysis is carried out by constraining the all landing of the root and Z-direction of the root. The blades are run at the speed of 3000 rpm and stresses are compared. From Modal analysis frequencies are obtained for the four modes.

K. Analysis for blade

1) FEM Model Details

Assumptions

Material is isotropic.

Modulus of elasticity is 2.15e5 N/mm^2.

Poisson's ratio is 0.3.

Density of the material is 7.7e-9 Kg/mm^3.

Proof Stress 1130 N/mm^2.

2) *Static Analysis:* From static analysis we have to find out the maximum stress area of the blade at running RPM trough von Mises stress.

Here we observe the maximum stress is 907.359 N/mm^2, as per the given material property, the proof stress is 1130 N/mm^2. So, the running speed of the steam turbine blade at 3000 RPM is within the yield limit. And the blade failure because of crack occurs at max stress that is at blade root suction side, Z face 1st concave surface as shown in fig.7 and fig.8

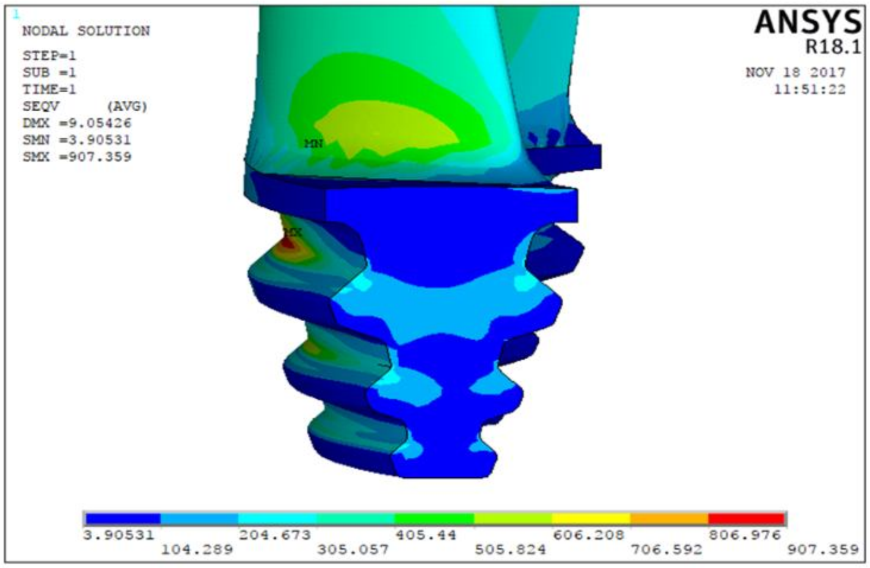


Fig: 7 Von mises stress at blade operating speed

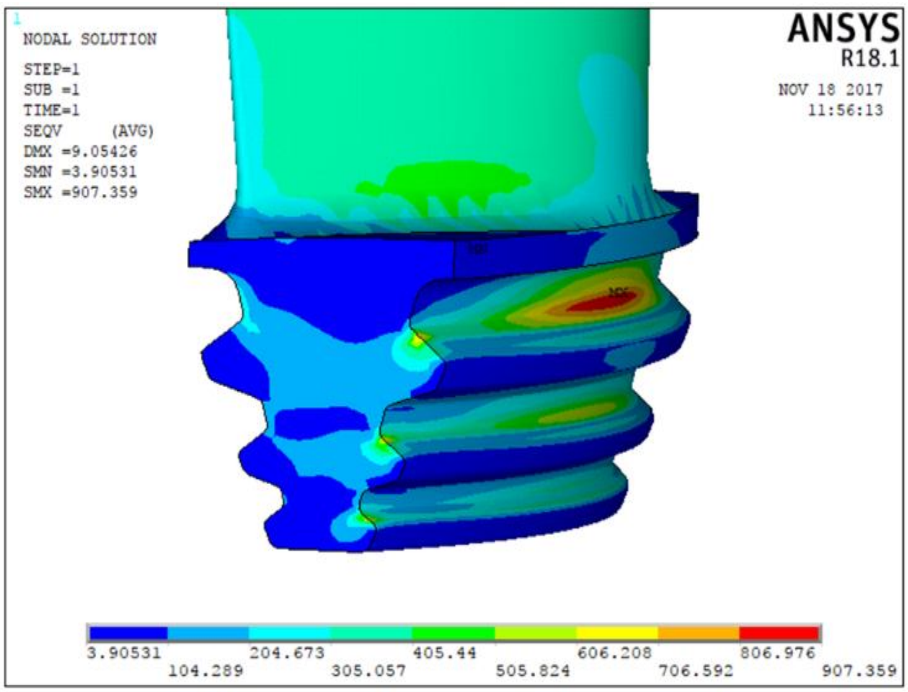
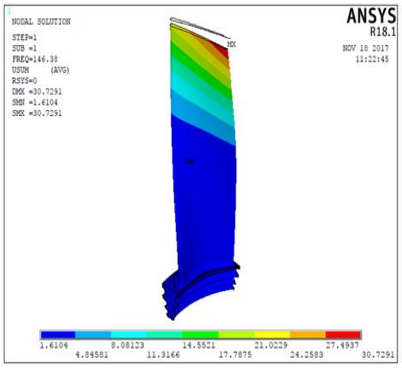
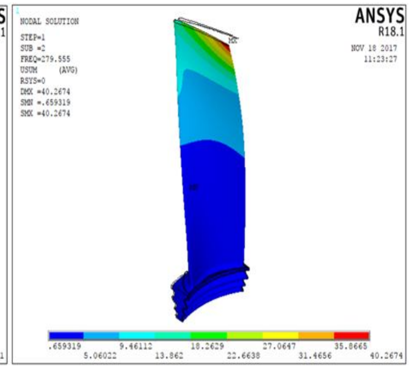


Fig: 8 Von mises stress at blade operating speed, max at blade root suction side

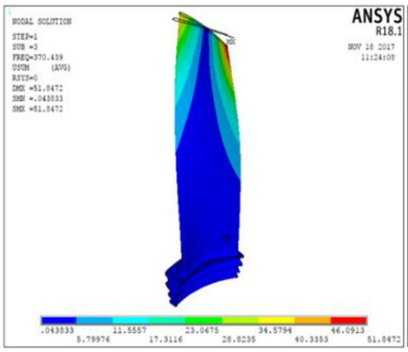
3) *Modal Analysis:* From Modal Analysis we extract 4 mode shapes of blade of different natural frequencies at different RPM, The natural frequencies of the blade plays an important role while increasing and decreasing the speed of rotor i.e., unsteady centrifugal forces during stats up and shut down of the turbine machine often will effect to identify the crack at root of the blade. We observe the modal shapes with bending and twisting at different RPMs and different frequencies, the bending will effect failure of blades because of bending unsteady centrifugal loads and steam flow. And from natural frequencies we plot a Campbell diagram to know the critical frequencies of the blade.



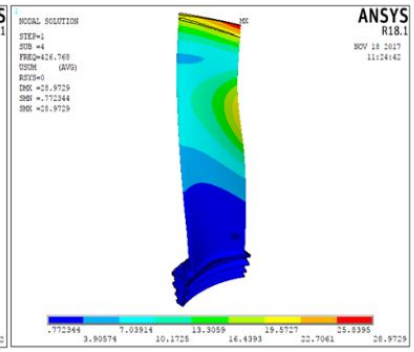
Mode 1



Mode 2



Mode 3



Mode 4

Fig: 9 Mode shapes with pre-stress effect at 3000 RPM

IV. RESULT AND DISCUSSIONS

Static and modal analysis is carried out for the root with blade for different speeds here pre stress effects is also included and von Mises stress is obtained for 3000 rpm as shown in Fig: 7 to fig 9. Natural frequencies for root with blade were obtained from modal analysis. Campbell Diagram is drawn for root and blade by taking the frequencies obtained for different speeds to see whether the blade is resonating in the operating region in table 1. It is observed that the first two modes are not intersecting the engine order lines as shown in the Fig 10. Thus they are free from resonance. Only first two bending modes are considered because they are only critical and cause failures in turbine blade.

Blade	RPM	Mode 1	Mode 2	Mode 3	Mode 4
Blade with root	0	88.68	177.97	361.94	388.22
	1000	93.47	182.61	363.27	392.57
	2000	106.25	195.64	367.09	405.36
	3000	123.99	214.88	373.10	425.53
	3600	136.03	228.37	377.68	440.82

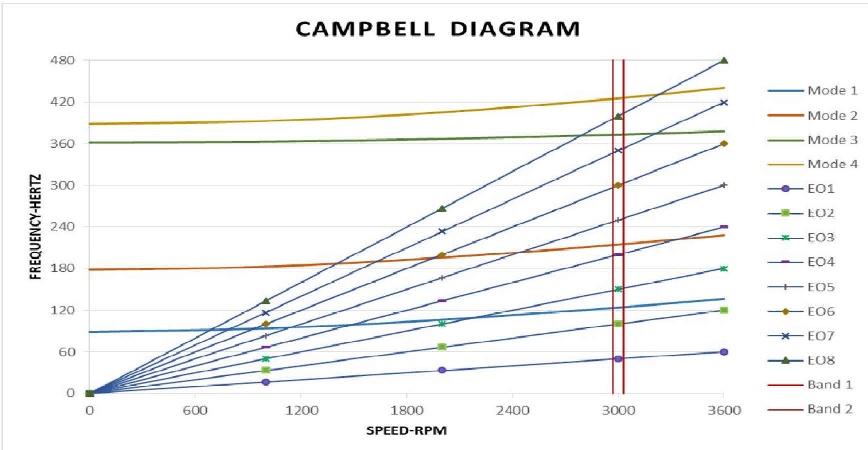
Table: 1 Natural frequencies of different mode shapes at different RPM

A. Campbell Diagram

The Campbell diagram is a pictorial representation of the variation of blade fundamental and harmonic frequencies as a function of the speed of rotation. This diagram is used to determine the adequacy of a blade in avoiding operation at various conditions, where it could be operating at risk. The Campbell diagram or the interference diagram indicates the vibratory stress levels in a given stage. Since almost all the blade failures are caused by vibratory stress, many reliability conscious purchasers are requesting Campbell diagrams with turbine quotes or orders. A Campbell diagram is a graph with turbine speed (rpm) plotted on the horizontal axis and frequency (Hertz) in cycles per second plot on the vertical axis. The blade frequencies and the frequencies of exciting forces are plotted on the Campbell diagram. This diagram predicts where the blades natural frequencies coincide with the exciting frequencies. When a blade frequency and an exciting frequency are equal, or intersect, it is called a resonance.

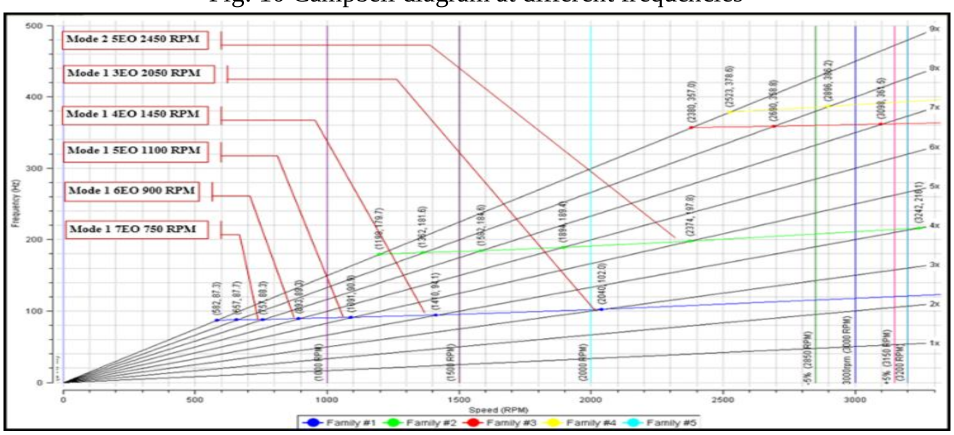
The frequency that a turbine blade will have is a function of its mass and stiffness. There are three basic types of frequencies that exist in a blade vibration, tangential, axial and torsional. Blade frequency can be determined analytically or by testing. The main advantage of testing is that the exact frequencies are found. The blade properties are not being approximated as in analytical determination. The disadvantage of testing are that it is time consuming and tedious and the tests are done on stationary rotor, which does not include centrifugal stiffening and a possible change in boundary conditions in the root.

If the Campbell diagram shows that a resonance exists in the operating range several design changes can be made to avoid it. Since frequency is the function of mass and the stiffness. The blade frequency can be changed by going to a stronger stiffness blade section. If the resonance is with nozzle passing frequency, the number of nozzles can be changed to avoid resonance.



A. Analytical Campbell Diagram

Fig: 10 Campbell diagram at different frequencies



B. Experimental Campbell Diagram

Fig: 11Campbell diagram at different frequencies

Summary Campbell for LP blade from experimental and analytical figures shows all the significant synchronous vibratory modes observed. 1st bending mode is observed as 3rd EO crossing at 2050 RPM, 4th EO at 1450 RPM and 2nd bending mode is observed as 5th EO at 2450 RPM, on both generator and turbine side blades.

V. CONCLUSION

1) The study presents the technique and methodology for studying the vibration behavior of turbine blade. The study highlights the fact that although the major advances have been made in the blade vibration design technology, failures still take place, there by emphasizing that blade behavior is complex.
2) Finite element results for free standing blades give a complete picture of structural characteristics, which can be utilized for the improvement in the design and optimization of the operating conditions. The results are correlated with the experimental results

and varying the profile to get the optimum results. The final blade profiles obtained after complete analysis was free from resonance in the operating region. The profile point coordinates are generated and send to manufacturer for manufacturing the blade.

3) The above studies can give more accurate values of dynamic stress levels, if the magnitude of dynamic force experienced by the blade is exactly known. The blade is subjected to such forces due to flow excitation and flow disturbances under various operating conditions. The results obtained from the analytical work matches quite closely with the experimental results thus showing the accuracy of the model and adequacy of boundary conditions.
4) An atempt was made to model the blade for wide frequency operation. Considerable work in this area is still required and should be attempted to completely analyze the blade behavior.

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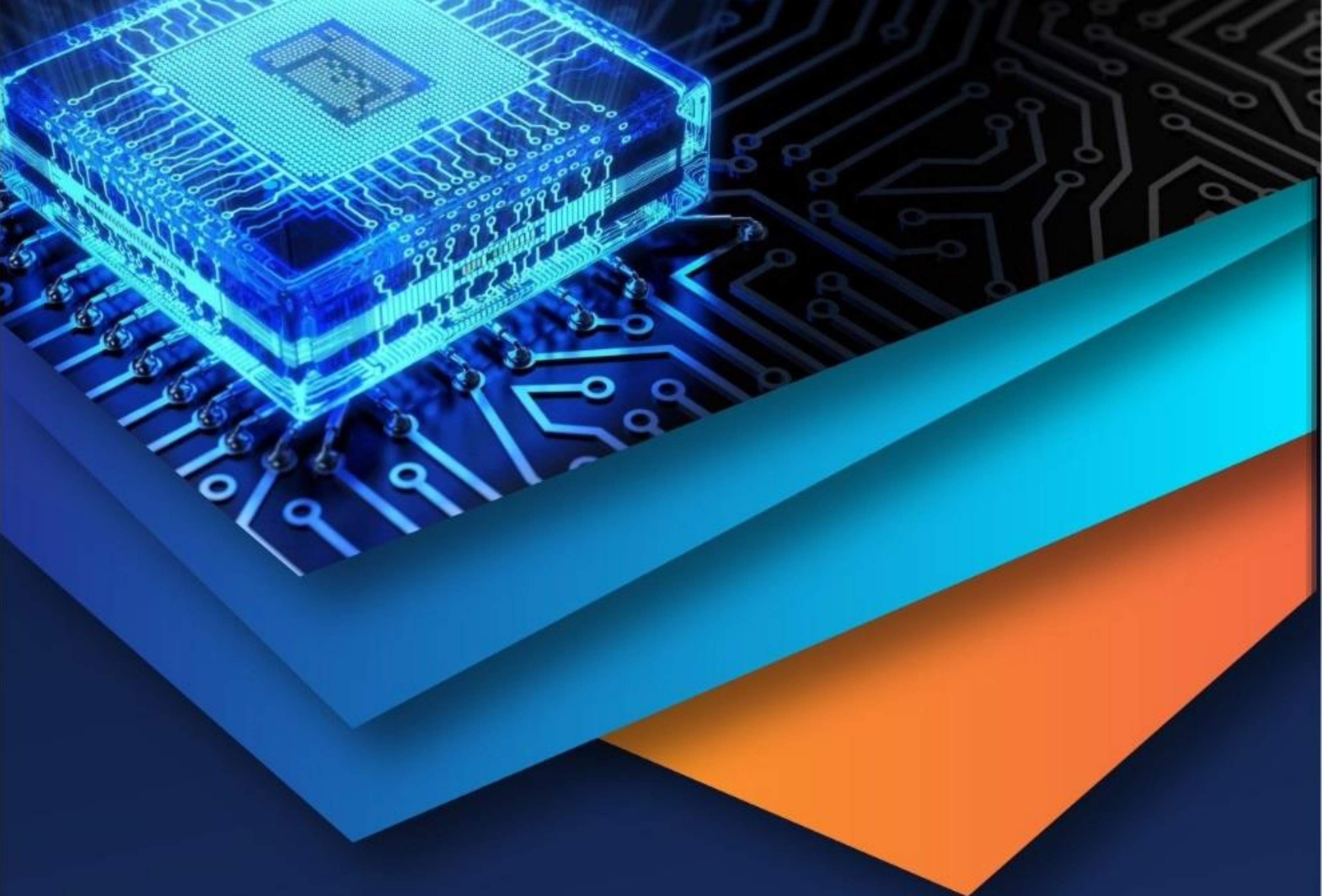

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