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Design and Vibrational Analysis of Steam Turbine High Pressure Moving Blade

K. Kumaraswamy[1], A. Siva Naga Raju[2]
[1]*M.Tech. Student, Dept. of Mechanical Engineering, Chinthalapudi Engineering College, Chinthalapudi, India*
[2]*Assistant Professor, Dept. of Mechanical Engg., Chinthalapudi Engineering College, Chinthalapudi, India*

***Abstract*: This Project addresses the design issue of steam turbine moving blade and vibrational analysis of the moving blade using ANSYS. A specific focus on blade profile, material used in the production of steam turbine blades, and the factors that cause turbine blade failure. This project enumerates and describes the currently available technologies. In particular, this project evaluates the effectiveness of certain titanium alloys and steels in resisting creep and fracture in turbine blades. The effectiveness of chemical and thermal coatings in protecting the blade substrate from corrosion when exposed to wet steam is addressed. The stresses developed in the blade as a result of steam pressure, steam temperature, and the centrifugal forces due to rotational movement are delineated, current designs calculated to counter the fatigue caused by these stresses are presented. The aerodynamic designs of both impulse and turbine blades are compared and contrasted and the effect that these designs have on turbine efficiency are discussed. The results and conclusions are presented for a study concerning the durability problems experienced with steam turbine blades. The maximum operational VonMises Stresses are within the yield strength of the material but the deformation is comparatively better for material AISI 422(martensitic stainless steel). Based on the research presented here in this project presents a detailed summary of what modifications to existing high pressure steam turbine blades can be made to increase turbine efficiency.**

***Keywords*: Steam Turbine, High Pressure Blade**

1. Introduction

Blades are the heart of a steam turbine, as they are the principal elements that convert the thermal energy into kinetic energy. The efficiency and reliability of a turbine depend on the proper design of the blades. It is therefore necessary for all engineers involved in the steam turbines engineering to have an overview of the importance and the basic design aspects of the steam turbine blades, Blade design is a multi-disciplinary task. It involves the thermodynamic, aerodynamic, mechanical and material science disciplines. A total development of a new blade is therefore possible only when experts of all these fields come together as a team.

- Efficiency of the turbine is depends on the following parameters.
- Inlet and outlet angle of the blade
- Blade Materials.
- Profile of the blade.
- Surface finishing of the blade.

The major cause of break down in turbo machine is the failure of Turbine blades like as

1. Creep
2. Blade Corrosion
3. Accumulation failures etc.

Proper design of the turbo machine blade plays a vital role in the proper functioning of the turbo machine.

Efficiency of the turbine Blade Depending upon following factors:

1. Blade materials.
2. Inlet and outlet angle of the blade.
3. Profile of the blade.
4. Surface finishing of the blade.

A. Assumptions

1. Blade "Materials: CA-6NM and AISI 422
2. Design of "High Pressure blade" : Vector Diagram
3. A specific focus on "Blade profile: Airfoil Design Workshop
4. Blade Modelling: Pro/E Wildfire V5.
5. Analysis- Ansys 13.0

B. Objects

1. The material of the blade was specified as CA-6NM and AISI 422. This materials is an iron based super alloy.
2. To analyse the mechanical and radial elongations resulting from the Tangential, Axial and Centrifugal forces.
3. To analyse structural analysis.

For Modeling and analysis of the Steam turbine blade, CAD Software packages namely Pro/E Wildfire 5.0 and ANSYS 13.0 have been used.

C. Blade materials

1. Blade material must have some or all of the following properties, depending on the position and role.
2. Corrosion resistance (Especially in the wet LP stage)
3. Tensile strength (To resist centrifugal and bending stresses)
4. Ductility (To accommodate stress peaks and stress concentrations)

5. Impact strength (To resist water slugs)
6. Material damping (To reduce vibration stresses)
7. Creep resistance

2. Materials compositions and microstructures

Table 1

Chemical Composition Limits: CA-6 NM (Cromium-Nickle)								
Composition, %	Cr	Ni	Mo	Si	Mn	P	S	C
CA-6NM	11.5-14.0	3.5-4.5	0.40-1.0	1.00	1.00	0.04	0.03	0.06

Microstructure - Tempered Martensitic

Table 2

Chemical Composition Limits: AISI 422 ALLOY											
Weight %	C	P	Si	Ni	V	Fe	Mn	S	Cr	Mo	W
Alloy 422	0.20-0.25	0.040 max	1.0 max	0.50-1.0	0.20-0.50	Bal	1.0	0.030 max	11.5-13.5	0.75-1.25	0.75-1.25

Microstucture - Martensitic standard

Table 3
Mechanical properties

Alloy	CA-6NM	AISI 422
Heat treatment	>955 °C	>1038 °C
Tensile Strength	827 MPa	965 MPa
Yield strength (0.2% offset)	689 MPa	760 MPa
Elongation in 50 mm (2 in.), %	24	13
Reduction in area, %	60	30
Hardness, (HB)	269	290

Overall, it is the material properties that make a blade reliable to failure. The yield strength, tensile strength, corrosion resistance, and modulus of elasticity all play a role in determining whether or not a blade will fail under operating loads.

A. *Theory behind static analysis*

In the static analysis we calculate the Centrifugal stresses, Axial stresses and Radial stresses.

Centrifugal stresses: The centrifugal forces exert the tensile stresses at the blade root, which pulls the blade away from the disc or the rotor. So sufficient section must be provided to the blades at the root and the material capable of withstanding the stresses without fatigue must be selected.

Blades of area A, with angular frequency ω and density ρ exert centrifugal

force, $F_c = \rho\ A\ H$

$\omega^2\ r$

It is also given by following

equation, $F_c = \rho\ A\ A_a / 2\pi(2\pi/60)^2$

Where,

A= annulus area = $[\pi r^2/(t\text{-}r2n)]$

The centrifugal forces at the blade root section is the centrifugal force divided by area of the blade section at the root.

$\sigma_c = \rho A_a / 2\pi(2\pi n/60)^2$

B. *Theory behind thermal analysis*

The design features of the turbine segment of the steam turbine have been taken from the “Preliminary design of a power turbine. It is absorbed that in the above design, after the rotor blades begins designed they are analyzed only for mechanical stresses but there is no evaluation of thermal stresses. As he temperature has a significant effect on the overall stresses in the rotor blades a detailed study is carried out on the temperature effects to have a clear understanding of the combined mechanical and thermal stresses and the Radial elongation resulting from the Axial and Centrifugal forces.

The gas forces namely Tangential and Axial are determined by constructing velocity triangles at the inlet and exit of the rotor blades. For obtaining the temperature distribution, the convective heat transfer coefficients on the blade surface exposed to the gases are fed in to the software. The radial elongation in the blade is also calculated. Temperature distributions and elongations are evaluated at several sections in the rotor blade.

The blades are designed for strength on the basis of the total effects of both static and dynamic stresses since the blades are designed to these stresses at one and the same time. The centrifugal forces causes tensile and bending stresses of constant magnitude whereas gas pressure causes bending stresses, due to centrifugal forces are known as static stresses and those due to gas pressure are known as dynamic stresses.

The most dangerous of a constant section is the one at the root since it is weakened by the presence of reverting holes etc. If a blade is acted up on by instantaneous forces free vibrations are setup. The frequency of these vibrations depends on the dimensions of the blade or blade assembly and their mounting on the disc. There is a lot of stress concentration entailed in the root portion of the blade, so care should be taken to reduce this concentration. For blades with constant blade section along the length, the stresses at the weaker section are:

$$\sigma = C_o/F_o = (C_b + \Sigma C_s)/F_o$$

Where, C_o = =Centrifugal forces of the blade, shroud etc, F_o = area of the weakest blade section (root section).
The centrifugal forces of a constant section blade will be:

$$C_b = G_b \gamma_{xy} \omega 2/g = F_o \gamma_{xy} \omega 2h/g$$

Where,
G is the weight of the blade, h is the height of the blade,
γ_{xy} is the mean diameter, ω is angular velocity.

The centrifugal forces of the shrouding is obtained as

$C_s = G_s\ r_s \omega^2/g = F_s \gamma r_s \omega^2 I_s/g$

Where,

γ is the specific weight of the material from which the blades are made.

r is radius of the strip centroids.

The centrifugal forces of the binding wire

is $C_w = G_w r_w \omega^2/g = F_w\ I_w\ r_w \gamma\ \omega^2/g$

C. Bending and twisting calculations

Maximum tangential stress produced in shaft is equal to

TMax= ½ w (Mb_2+Mt^2)w = 32πd

Where, d is diameter of the shaft, and Mb, Mt are bending and twisting moments.

Twisting moment at the chosen section is given by Mt = 97300 Ni/N

Ni is total power developed and N is RPM of turbine.

Maximum bending moment can be calculated graphically i_r by shear force bending moment diagram. For obtaining the stresses the T shaped root node degree of freedom are constrained in the UX, UY and UZ directions and tangential, axial and centrifugal forces are applied at the centroids. The axial and tangential forces result from the gas momentum changes and from pressure differences across the blades, which are evaluated by constructing velocity triangles at the inlet and outlet of the rotor blades.

D. Inlet velocity triangle

The axial and tangential forces result from the gas momentum changes and from pressure difference across the blades, which are evaluated by constructing velocity triangles at the inlet and outlet of the rotor blades.

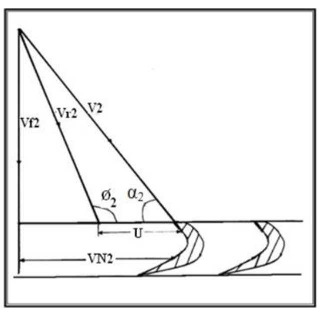

Fig. 1. Inlet triangles for 1st stage rotor blade

From the inlet velocity triangle of a rotor blade we get, Whirl velocity (Vw2) = 422.74m/s.

Flow velocity (Vf2) = 186.89m/s. Relative velocity (Vr2) = 265.09m/s.

Blade angle at the inlet (θ3) = 135.170

The axial and tangential forces results from the gas momentum changes from pressure difference across the blades, which are evaluated by constructing velocity tangles at the inlet and outlets of the rotor blades.

E. Exit velocity triangle

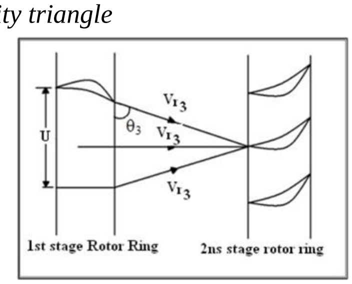

Fig. 2. Exit velocity triangles for 1st stage rotor blade

At the exit of first stage rotor blades, Flow velocity (Vf3) = 180.42m/s.

Relative flow angle (ψ3) = 37.88o Whirl velocity (Vw3) = 2.805m/s. Relative velocity (Vr3) = 293.83m/s.

F. Evaluation of Tangential (F_t), Axial force (F_a) and Centrifugal force (F_c) on each rotor

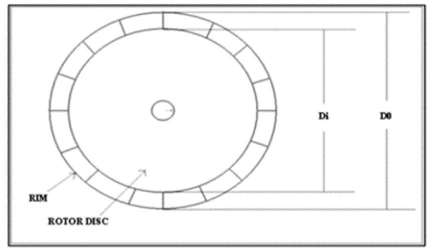

Fig. 3. 1st Stage Rotors

Evaluation of gas forces on first stage rotor:

At the inlet of first stage rotor blades,
Absolute flow angle (α2) = 22.85o
Absolute velocity (V2) = 462.21m/s.
Dia of blade mid span (γ) = 1.3085m.
Design speed of turbine (u) = πDN/60.

Tangential force Ft = m(Vw2 + Vw3)Newton.
Axial force Fa = m(Vf2 + Vf3)Newton.

Where,
m is mass flow rate of gases through the turbine. Referring to the above figure
M =(ρ2 (D0 − Di)Vf2)/4

Where,

ρ_2 is the density of gases at the entry of first stage rotor,
ρ_2 = 0.8900 kg/m^3,
m = 70.925 kg/s.

Total axial force on first stage rotor Fa=458.88N.
Total tangential force on each rotor Ft=29783.88N.
Number of blade passages in first stage rotor = 120.
Tangential forces on each rotor blade,
Ft=Ft/no. of blade passages

Ft=248.199N
Axial force on each rotor blade, Fa=Fa/no. of blade passages

Fa =3.82N.

From Euler's Energy Equation, Power developed in First stage rotor P = m(Vw2 U + Vw3 U)
Using the above Equation P = 6.991MW
The distance

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X =(m1x1 + m2x2 + m3x3)/(m1 + m2 + m3)

Where,
m1, m2 and m3 are masses of volume 1,2 and 3. x1, x2 and x3 distances of the centroids of volumes, 2 and 3 from
The density of material ρ is graphically measured to be: ρ = 8900kg/m2
m1 = 0.382
kg m2 = ρ x
V2 m3= ρ x
V3

Where,
V2 and V3 are volumes of portions 2 and 3 of rotor blades,
The distance X is calculated as 648.85mm.

Total mass M = m1+ m2+m3.
Centrifugal force Fc =M(2πn/60)2X and its value is found to be 38038.33N

3. Blade modelling and analysis

A. *Blade modelling*

Sequential Steps for Modeling by Using Pro/Engineer

- Firstly, select the datum in part module where the model has to be generated
- Go to main menu/File/New/Part Name.”*./ok”
- Click on insert > extrude> placement>sketch/Select plane to sketch> click ok
- Points to be plotted with the required dimensions in X-axis and Y-axis to get the required shape. Check all points and extrude the object to required height.
- Again select the plane to extrude the I-section on top of the blade
- Extrude the I-section to complete the required shape.
- The PRO-E file is saved in *.igs format.
- “*.igs” format is good format for the data translation between any packages.

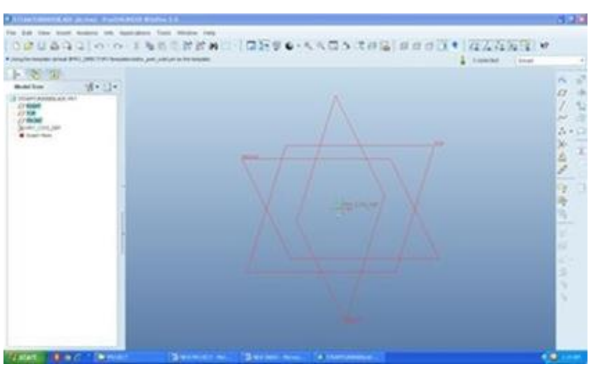

Fig. 4. The geometry under consideration is generate in the PRO-E modeling package

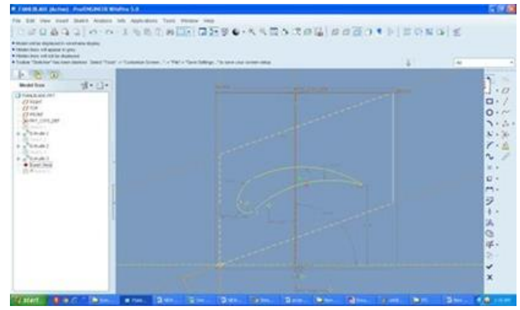

Fig. 5. Blade Airfoil design by using Pro/E

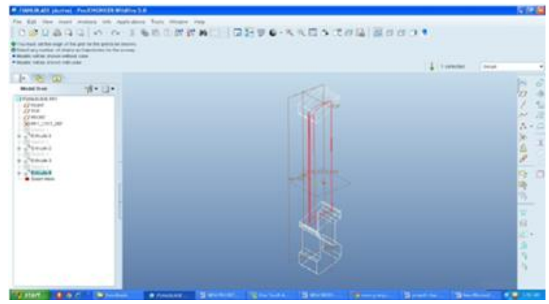

Fig. 6. Wireframe model of blade

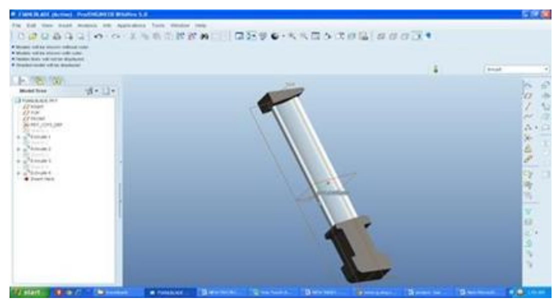

Fig. 7. Steam turbine blade generated in pro/engineer

B. *The results file*

The ANSYS solver writes results of an analysis to the results file during solution. The name of the result file depends on the analysis discipline:

- Job name.RST for a structural analysis
- Job name.RSH for a thermal analysis
- Job name.RMG for a magnetic field analysis
- Job name.RFL for a FLOTRAN analysis

For a FLOTRAN analysis, the file extension is .RFL, for other fluid analysis, the file extension is .RST or RTH, depending on whether structural degrees of freedom are present.(using different file identifiers for different disciplines helps you in coupled-field analysis where the results from one analysis are used as loads for another. This presents a complete description of coupled-field analysis).

Analysis of the model is done in ANSYS software

- The model is imported into ANSYS as File>import>.ges…..
- Once file is imported, need to check all volumes and areas for any data loss generally occurs.\
- Once model is free from the geometric errors, than save the file as File>save as>filename.db

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- *.db is the extension of the ANSYS file
- Preferences >structural>ok
- Preprocessor>element type >brick solid45>ok
- Preprocessor>material prop>structural>linear>elastic>isotropic>modulus of Elasticity (2.15e5)>Poisson Ratio(0.3)>ok
- Preprocessor>material prop>structural>density(7.7e-9)>ok
- Meshing>mesh tool > Assign element size(2)>volume mesh
- Solution>defineloads>apply>Structural
- >Displacement >on areas> (select appropriate area)
- >ok
- Solution > define loads > Angular velocity >ok
- Solution >Analysis type >static >per stressed condition on > ok
- Solution > analysis type > modal Analysis > no of Modes to extract >per-stressed condition on > ok
- Solution > solve>current L.S >ok
- General post processor >nodal solution > deform shape > def+ unreformed shape
- General post processor > read results >by pick >gives the set of natural frequencies
- Once frequencies are known to us, we can read particular frequency and we can go for mode shapes
- Frequencies vs. speeds gives the Campbell diagram which is helpful to find out the critical frequencies, thereby we avoid the resonation effect.

C. Creative phase

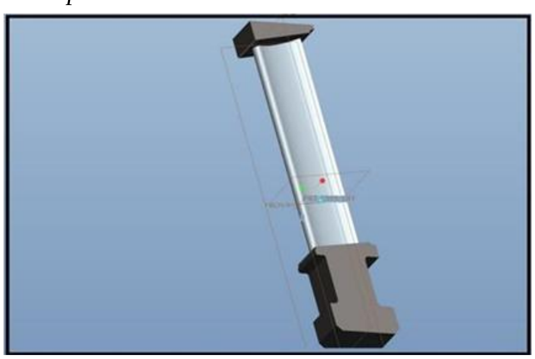

Fig. 8. HP blade file is to to be export format of "*.igs"

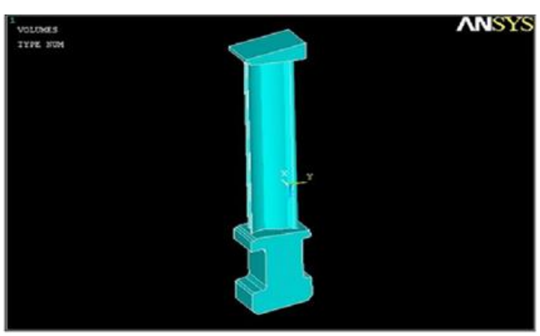

Fig. 9. HP blade import in Ansys software from .igs file

Fig. 10. HP blade meshing in by sing Meshing tool

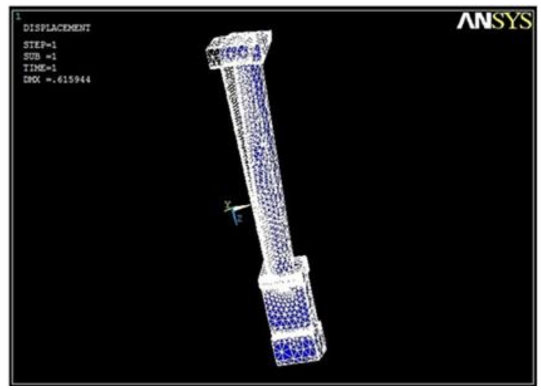

Fig. 11. Displacement model under load as per calculations

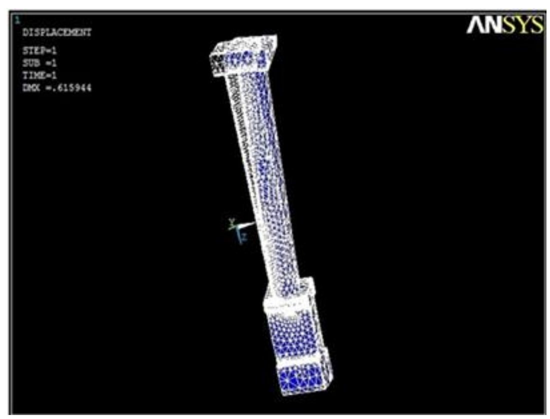

Fig. 12. Displacement model with changing position

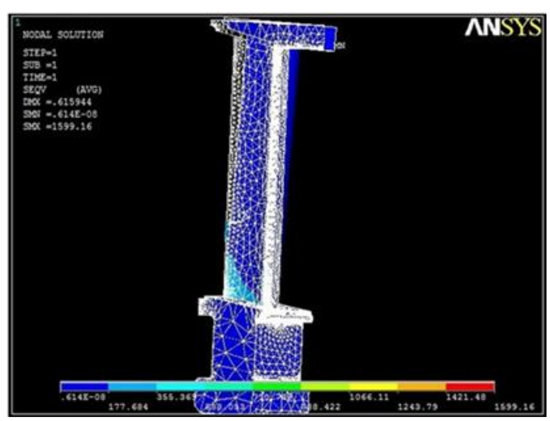

Fig. 13. Von misses stress analysis nodal solution

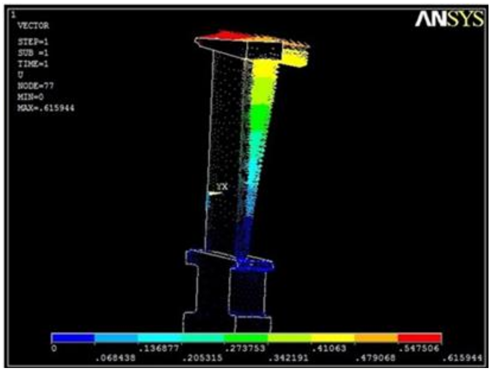

Fig. 14. HP blade vector mode analysis

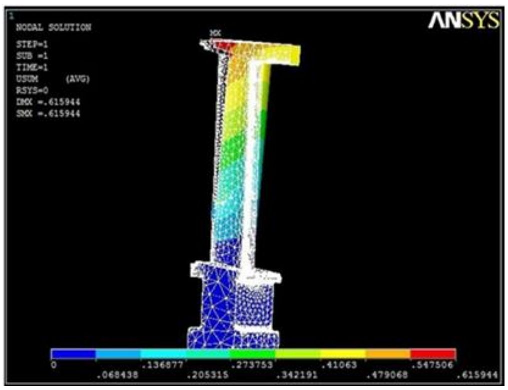

Fig. 15. HP blade vector sum nodal analysis

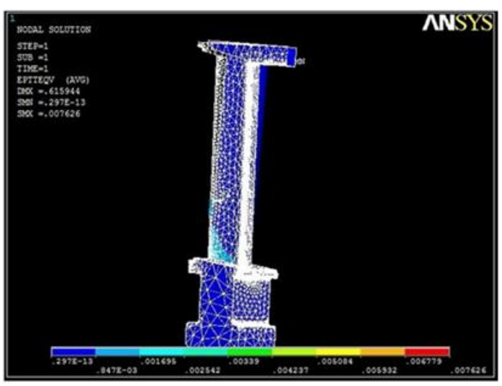

Fig. 16. Von misses total mechanical and thermal strains

4. Conclusion

The implementation of robust turbine blades, designed in accordance with the latest material technologies and able to withstand the most trying of circumstances, in combination with the use of clean, renewable fuels presents an efficient method of generating substantial amounts of electricity. An improved blade design, focused on resisting the effects of stresses, corrosive agents, and creep-inducing temperatures, will elevate the turbine efficiency, consequently leading to an increase in the power plant's overall efficiency, a reduction of the amount of fuel consumed, and ultimately a decrease in operating costs. To improve efficient blade design, will serve to reduce operating costs even further and lessen the environmental impacts of steam turbines. Overall, such a combination of technologies would benefit society by providing an efficient, viable, and sustainable means of generating electrical energy.

Steam Turbine blades are designed with a significant factor of safety, the operating forces and stresses calculated above are normal for a last stage blade and under normal conditions this should not pose any risk to the blades integrity. Large steam turbine blades are manufactured from 12% chrome high alloy steels where maximum design stress values of 200-300 N/mm2 are permissible. This exemplar demonstrates that under normal operating conditions, stresses are well thin acceptable limits. However, what the exemplar also shows are the large forces and stresses involved in such rotating machinery and how important factors such as design philosophy, manufacture and maintenance strategy are to ensure safe operation.

The results and conclusions are presented for a study concerning the durability problems experienced with steam turbine blades. The maximum operational VonMises Stresses are within the yield strength of the material but the deformation is comparatively better for material AISI 422 (martensitic stainless steel). Modernization solutions are the application of the Steam turbine blade technology to existing machines to maximize their efficiency, improve reliability, and reduce life cycle costs.

The discussion of these manufacturing solutions focused on the technology and this project is made on the basic manufacturing technique of steam turbine blades. The procedure involved in this manufacturing leads to achieve the best surface quality and structure. This method uses two separate cutting steps to reach the full depth of cut, which allows the cutting force to be reduced more effectively than by reducing the feed per tooth, as it allows the chip thickness to be modified towards the recommended target values.

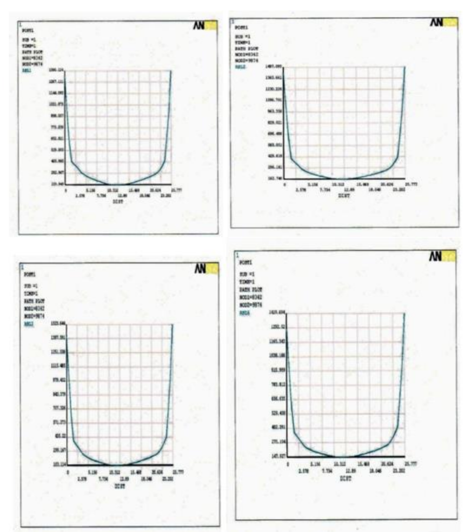

Fig. 17. Stress vs. Root Axial Distance in MM

Stress analysis is carried out for the root with blade for different speeds here pre stress effects is also included and

Table 4
Natural Frequencies of Different Mode Shapes at Different RPM

Blade	RPM	1st MODE	2nd MODE	3rd MODE	4th MODE
250NEWLP3	0	88.668	177.97	361.92	388.22
	1000	93.466	182.61	363.27	392.57
	2000	106.25z	195.64	367.09	405.36
	3000	123.99	214.88	373.1	425.53
	3600	136.03	228.37	377.68	440.53

Table 5
Von Mises Stresses at 3000 rpm for different Deflection

S. NO	Deflection (mm)	VON Mises Stress (MPa)			
		UX		UZ	
		Minimum	Maximum	Minimum	Maximum
1	0	3.90133	906.435	3.90133	906.435
2	0.1	4.20005	907.688	4.15046	922.043
3	1	4.20815	911.019	4.16555	939.074
4	5	3.2306	925.832	3.47329	1014.82
5	10	1.8457	1109.57	2.46175	1437.48

vonmises stress is obtained for 3000 rpm as shown in Fig. 4. And the Table 4 Natural frequencies for root with blade are 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 3 and Fig. 5. It is observed that the first two modes are not intersecting the engine order lines as shown in the Fig. 5. Thus they are free from resonance. Only first two bending modes are considered because they are only critical and cause failures in turbine blade. Graphs are plotted at the mid-section of the root i.e. where stress is maximum. From graphs it is observed that stress is within yield stress limits.

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