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Design and Vibratory Analysis of High Pressure Steam Turbine Moving Blade

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ABSTRACT: In this paper addresses the issue of steam turbine efficiency by discussing the overhaul design of high pressure steam turbine blades. A specific focus on blade profile, material used in the production of steam turbine blades, and the factors that cause turbine blade failure and therefore the failure of the turbine itself. This project enumerates and describes the currently available technologies that enhance the overall efficiency of the generator and prevent turbine failure due to blade erosion and blade cracking. 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 the wet steam will also be addressed.

KEYWORDS: Steam turbine, High pressure blade, Blade design.

I. INTRODUCTION

A turbine is a device that converts chemical energy into mechanical energy, specifically when a rotor of multiple blades or vanes is driven by the movement of a fluid or gas. In the case of a steam turbine, the pressure and flow of newly condensed steam rapidly turns the rotor. This movement is possible because the water to steam conversion results in a rapidly expanding gas. As the turbine rotor turns, the rotating shaft can work to accomplish numerous applications, often electricity generation.

In a steam turbine, the steam energy is extracted through the turbine and the steam leaves the turbine at a lower energy state. High pressure and temperature fluid at the inlet of the turbine exit as lower pressure and temperature fluid. The difference is energy converted by the turbine to mechanical rotational energy, less any aerodynamic and mechanical inefficiencies incurred in the process. Since the fluid is at a lower pressure at the exit of the turbine than at the inlet, it is common to say the fluid has been expanded across the turbine. Because of the expanding flow, higher volumetric flow occurs at the turbine exit leading to the need for larger turbine exit areas than at the inlet. This is how one can tell a turbine symbol from a compressor symbol. In a turbine with a high inlet pressure, the turbine blades convert this pressure energy into velocity or kinetic energy, which causes the blades to rotate. Many green cycles use a turbine in this fashion, although the inlet conditions may not be the same as for a conventional high pressure and temperature steam turbine. Bottoming cycles, for instance, extract fluid energy that is at a lower pressure and temperature than a turbine in a conventional power plant. A bottoming cycle might be used to extract energy from the exhaust gases of a large diesel engine, but the fluid in a bottoming cycle still has sufficient energy to be extracted across a turbine, with the energy converted into rotational energy.

II. LITERATURE SURVEY

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

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In this work, an effort is made to minimize the stress by changing datum blade angle also along with other measures. The blade design is modified as per the inlet and outlet velocity triangle, In addition to this to improve the turbine and blade efficiency. In view of the above an attempt is made to explain in details about the improvement evolution, blade excitation and response, overhaul life estimation and results obtain of different researchers in their experimental investigations.

Subramanyam Pavuluri et.al. [1] predicted investigation on design of high pressure steam turbine blade addresses the issue of steam turbine efficiency. A specific focus on aerofoil profile for high pressure turbine blade, and it evaluates the effectiveness of certain Chromium and Nickel in resisting creep and fracture in turbine blades.

Rao J.S et.al. [3] made a two - dimensional analysis of free vibrations in the tangential direction. The first step is to develop the potential and kinetic energies for the tangential motion of the blades and shrouds. Second, Hamilton's principle is applied to derive the differential equation of motion and the boundary conditions. Then these equations are solved to determine natural frequencies.

Leissa.A.W et.al. [4] made a comprehensive study of the numerous previous investigations on the free vibration of twisted cantilever plates of rectangular platform which are results of a joint industry, government and university effort. Theoretically results received from different FEM programs utilizing shell theory and beam theory are compared with two independent sets of data obtained from experiments. Reasonable agreement among the theoretical results is found but it is recommended that further improvement in analysis method is necessary for increased reliability.

Park et.al [5] deal with the failures of turbine blades. Failures of turbine blades are identified as the leading causes of unplanned outages of steam turbine. Accidents of low - pressure turbine blade occupied more than 70% in turbine components. Therefore for preventing failures they had implemented the case study.

Improvement evolution:

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

Al-Bedoor.B.R [6] measured the blade vibrations in turbo-machinery. A blade vibration has been recognized as failure of turbo machinery, which has developed enormous efforts towards developing reliable techniques. They can be classified into broad categories namely the direct approach, such as using strain gauges bonded at the blade, and optical/laser methods to monitor the blade motion directly at one or more points on the blade span and the indirect approach, by extracting vibration, casing/bearing cap vibration, pressure fluctuations, performance monitoring of torsional vibration. By using these techniques he measured the vibrations in turbo machinery.

Rao.J.S, K.Gupta, N.Vyas[7] has done experiment to determine overall damping in a rotating turbine disc-blade system. A test spin rig has been designed and fabricated. Transient excitation of rotating blades is caused by suddenly shutting off the excitation to the rotating blades. Frequency analysis of the transient blade response gives the information about the modal damping with the speed of rotation and strain amplitude is obtained.

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 analyze the dynamic stresses. Many researchers have worked to developed the basic theories of isolated airfoil and have studied the flow interference in a turbo machinery stage.

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Rao.J.S [8] have been estimated the life of turbine blades. The practice of during blades from possible resonance near operation conditions is no more an acceptable criterion in the design of modern steam and gas turbines. With the advent of high-capacity, low-weight and high-speed design of these machines it is becoming essential to estimate the blade life at the design stage itself..

Rao.J.S [2] derived the equations of motion of blade on a disk rotating with angular velocity. The acceleration of the disk is taken as constant. They have obtained the forced vibration response of turbine blades during operations such as step-up and down, involving variation in angular velocity with time. They have shown that the acceleration values have significant influence on the blade response. the governing equation of motion of a cantilever blade on a rotating disk with variable angular velocity have been derived. Corolis forces are included in the derivation. The equation is a non-linear integral partial differential equation. A solution of this equation is obtained by Ritz averaging principle.

Overhaul life estimation:

Vibration induced fatigue is very common for turbo machine blading and this aspect is undergoing continues investigation. Fatigue crack normally initiates from the zone f high stress, having metallurgical or structural discontinuity and this may grow and lead to the failure due to operating conditions.

Delaneuvile.R.E, Cunningham.S.E[10] predicted the life of turbine engine blades under vibratory high cycle fatigue. A novel fracture mechanics approach has been used to predict crack propagation lives in gas turbine engine blades subjected to vibratory high cycle fatigue (HCF). The vibratory loading included both a resonant mode and a non-resonant mode, with one blade subjected to only the non resonant mode and another blade to both modes.

Murari P Singh and Terry Mathews [11] highlighted the fatigue damage of steam turbine blade caused by frequency shift due to solid build up. They discussed how due to the solid built up at the blade root shifted the natural frequency into interference, and changed the response characteristic of the bladed disk assembly.

Vyas.N.S, Rao .J.S [9] developed the fatigue life estimation procedure for a turbine blade under transient loads. they presented a technique for fatigue damage assessment during variable-speed operation. Transient resonant stresses for a blade with non-linear damping have been determined using a numerical procedure. A fatigue damage procedure is described. The fatigue failure surface is generated on the S-mean stress axes and miner's rule is employed to estimate the blade stiffness and other operating parameters.

a)High Pressure Steam Turbine Blade angles

The design vector angles for the turbine blade rows are listed in Table 1 for root and tip reference diameters, and the leading dimension of the existing blade design are shown In Table.1, As mentioned earlier these blades are constructed in the conventional manner using circular arcs. The profiles are repaired to have low curvature on the suction surface between the throat and trailing edge, and the channels between adjacent blades are made smoothly convergent up to the throat plane. The blade pitch/chord ratios are selected on the basis of general loss correlations.

b)Modified High Pressure Steam Turbine Blade angles

The following parameters are selected to provide the means for translating the blade design of the large reference turbine into the requirement of the smaller turbine.

- High Pressure Steam Turbine Blade
- Thickness of High Pressure Steam Turbine Blade Profile
- High Pressure Steam Turbine Blade loading

i) High Pressure Steam Turbine Blade Profile shape

The thickness distributions of the reference turbine are shown in table and these are adopted for the new rotor blades. The parabolic chamber line is also retained, the chamber angles being modified to suit the existing vectors. Whereas the existing blading had been designed with approximately zero incidences at all sections the new blades are required to incorporate the same local increscences as the reference turbine.

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ii) Thickness of High Pressure Steam Turbine Blade Profile

It is now well established that turbine efficiency is sensitive to the thickness of the blade trailing edges, and various analyses of this effect indicate that the ratio trailing edge thickness/blade pitch is the significant parameter against which loss in efficiency may be correlated. For this reason the (t_m/s) values of the reference blade sections are used to define the thickness of the new blades in preference to the more conventional maximum thickness/chord ratio. However it is found that the resulting blade geometry at the root of the second-stage rotor gave a channel throat upstream of the trailing edge are shown In Table 3. This is alleviated by reducing the originally calculated value of (t_m/c) from 13.5 to 11.2 %.

iii) High Pressure Steam Turbine Blade loading

The loading coefficient for a blade in cascade in terms of the pitch is maximum chord c_a and flow angles α_1 and α_2 is expressed as:

$$\Psi_t = 2s/C_a(\tan \alpha_1 + \alpha_2) \cos^2 \alpha_2$$

Thus expresses the tangential lift force experienced by the blade section in incompressible flow as a proportion of the exit dynamic head. Thus for a given form of pressure distribution Ψ_t provides a rough measure of the diffusion imposed upon the blade surface boundary layer. The loading coefficients which are evaluated for the large reference turbine blades at the root and tip stations are compared with corresponding values for the existing design in the following Table 4.

Table 1: High Pressure Steam Turbine Blade Angles at stage I & II

		Reference diameter (in.)	Inlet angle (deg.)	Outlet angle (deg.)
At Stage I	Stator	21.00	0	63.50
		27.45	0	56.85
	Rotor	21.00	47.30	50.51
		27.45	18.00	54.95
At Stage II	Stator	19.925	18.40	56.55
		28.525	13.0	46.68
	Rotor	19.925	37.40	35.50
		28.525	63.50	46.20

Table 2: Details of High Pressure Steam Turbine Blade at stage I & II

	At Stage I				At Stage II			
	Stator		Rotor		Stator		Rotor	
Reference diameter (in.)	1	27.45	21	27.45	19.925	28.525	19.925	28.525
Blade Number	59		83		66		62	
Pitch(s) in.	1.12	1.46	0.795	1.04	1.358	1.010	1.445	
Chord (c) in.	1.68	1.88	1.4	1.34	1.48	1.76	1.63	1.70
Axial chord (ca) in.	1.36	1.56	1.38	1.06	1.27	1.63	1.61	1.35
Throat (o) in.	0.5	0.798	0.466	0.538	0.522	0.936	0.767	0.938
Maximum thickness (t_m) in.	0.346	0.33	0.311	0.163	0.286	0.315	0.216	0.097
Trailing edge thickness (t_e) in.	0.032	0.032	0.032	0.032	0.032	0.032	0.032	0.032
Average rotor tip clearance	0.030 in.				0.020 in.			

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Table 3: Steam Turbine High Pressure Blade factors

				Leading edge radius		Trailing edge radius	
I	Lead and Trailing Edge of Root and Tip	Root		8% t_m		6% t_m	
		Tip		8% t_m		12% t_m	
II	Incidence angles for rotors	Rotor 1				Rotor 2	
		Incidence (deg.)	Root	Tip	Root	Tip	
			+9.0,	-4.0	+4.5	-6.0	
III	Rotor Blade Thickness Distributions	Station on camber line	Root section semi-ordinates		Tip section semi ordinates		
		25	1.375		1.375		
		2.5	1.94		1.94		
		5	2.675		2.675		
		10	3.6		3.6		
		20	4.55		4.55		
		30	4.95		4.95		
		40	4.82		4.9		
		50	3.98		4.175		
		60	3.08		3.345		
		70	2.26		2.62		
		80	1.54		1.975		
90	0.97		1.51				

Table 4: Co-Efficient of High Pressure Steam Turbine Blade Loading

	Existing Turbine		Reference Turbine	
	Root	Tip	Root	Tip
1st Stator	0.65	0.855	0.580	0.81
1st Rotor	1.07	1.14	0.910	0.093
2nd Stator	0.88	0.97	0.85	1.23
2nd Rotor	1.30	1.03	1.05	0.92

Original cascade work suggested that a loading coefficient of 0.8 is the optimum value but other test data delivered from turbine test rigs has shown that the optimum mean loading coefficient for a given blade row may lie between 0.7 and 1.10.

The comparison of the loading factors showed that the major differences between the two designs concerned the rotor blade rows, the loading of the existing turbine being higher for both stages. The mean diameter values for the first stator rows are very similar and the average loading for the second stator row of the existing blades is slightly below that of the reference turbine. In the light of this comparison it is decided to limit the scope of the redesign to the rotor blades only. The method employed for each section is to combine the loading coefficient of the corresponding section of the reference turbine blade with the local vector angles of the existing turbine. In this way the appropriate values of pitch/axial chord for the root and tip sections of each rotor blade row are established.

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III. MODELLING AND ANALYSIS OF BLADE

The blade is modeled using Pro-E, in which CAD model is prepared using Pro-E template and saved in .igs or .iges format. So that it is easy to import the CAD geometry to import into ANSYS software for analysis.

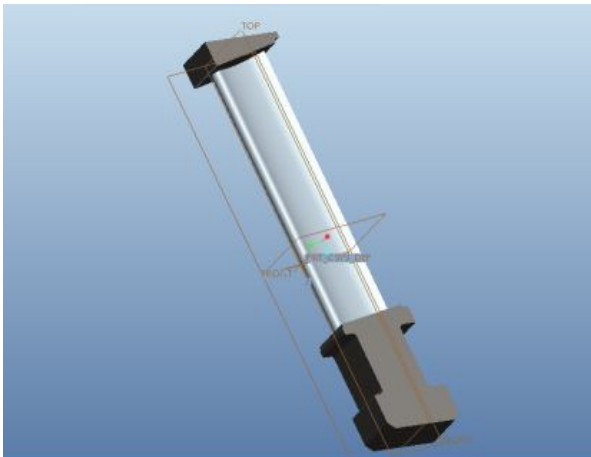


Fig.6 Steam turbine blade generated in Pro/Engineer V5.0

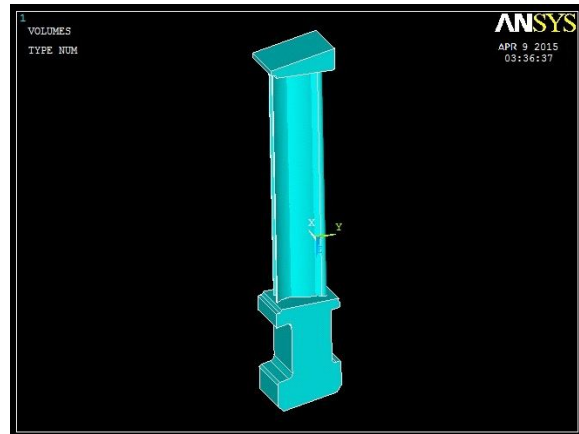


Fig.7 HP blade import in Ansys software from .igs

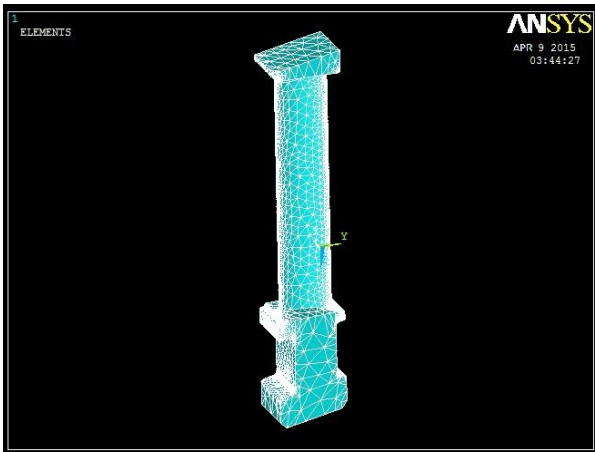


Fig.8 HP blade meshing in by sing Meshing tool

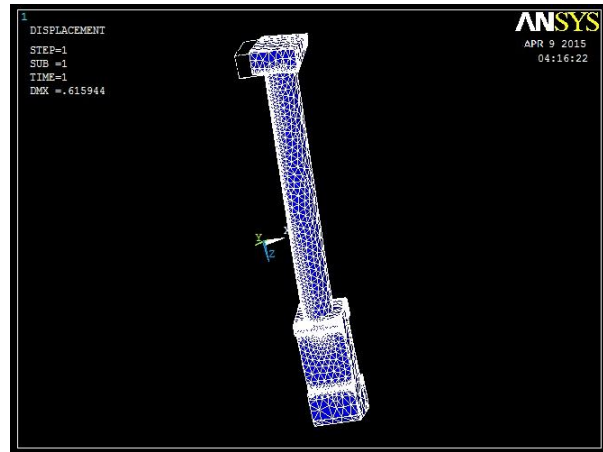


Fig.9 Displacement model with changing position

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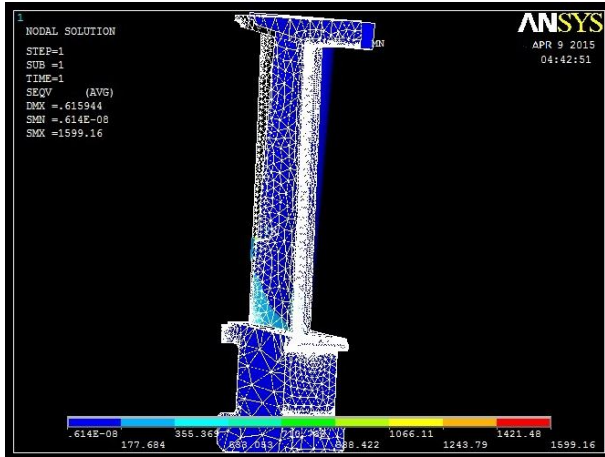


Fig.10 Von misses stress analysis Nodal solution

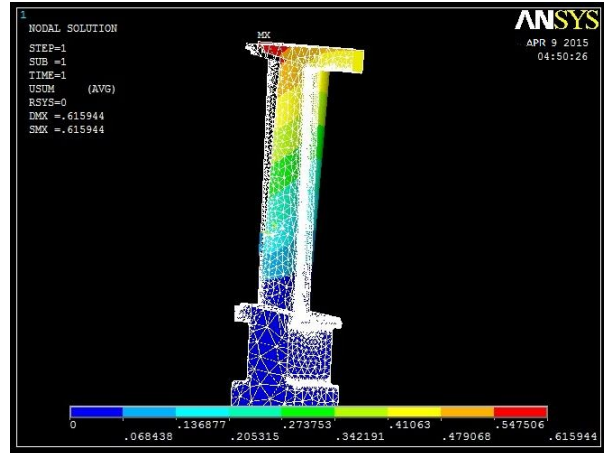


Fig.11 HP blade Vector Mode Analysis

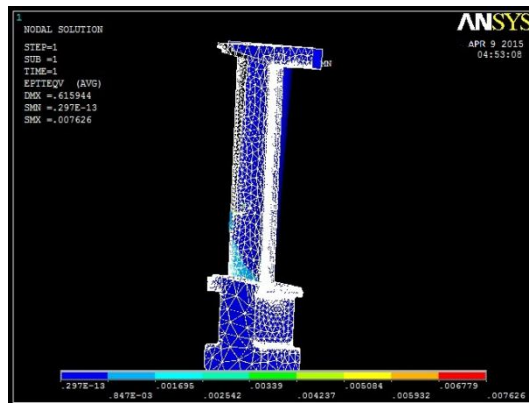


Fig.12 Von Misses Total Mechanical And Thermal Strains

IV. CONCLUSIONS

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.

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Table 5. Natural Frequencies of Different Mode Shapes at Different RPM

Blade	RPM	1 st MODE	2 nd MODE	3 rd MODE	4 th 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 6. 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

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