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EXPERIMENTAL INVESTIGATION ON DESIGN OF HIGH PRESSURE STEAM TURBINE BLADE

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Abstract: The Experimental 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. The capable of thermal and chemical conditions in blade substrate from to prevent the corrosion when exposed to wet steam. The efficiency of the steam turbine is a key factor in both the environmental and economical impact of any coal-fired power station. To increasing the efficiency of a typical 500MW turbine by 1% reduces emissions of CO_2 from the turbine station, with corresponding reductions in NOx and SOx. In this connection an attempt is made on steam turbine blade performance is important criterion for retrofit coal fired power plant. Based on the research presented modifications to high pressure high pressure steam turbine blades can be made to increase turbine efficiency of the turbine. The results and conclusions are presented for a study concerning the durability problems experienced with steam turbine blades.

Keywords: Steam Turbine, High Pressure Blade.

I. 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 parameters are, Inlet and outlet angle of the blade, Blade Materials, Profile of the blade and Surface finishing of the blade e.t.c.

The most critical aspect of steam turbine reliability centres on the bucket design. Since buckets, or rotating blades, are subjected to unsteady steam forces during operation, the phenomenon of vibration resonance must be considered. Resonance occurs when a stimulating frequency coincides with a natural frequency of the system. At resonance conditions, the amplitude of vibration is related primarily to the amount of stimulus and damping present in the system. High bucket reliability requires designs with minimum resonant vibration. The design process starts with accurate calculation of bucket natural frequencies in the tangential, axial, tensional, and complex modes, which are verified by test data. In addition, improved aerodynamic nozzle shapes and generous stage axial clearances are used to reduce bucket stimulus. Bucket covers are used on some or all stages to attenuate induced vibration. These design practices, together with advanced precision manufacturing techniques, ensure the necessary bucket reliability. Almost all of the blading used in modern mechanical drive steam turbines is either of drawn or milled type construction. Drawn blades are machined from extruded airfoil shaped pieces of material stock. Milled blades are machined from a rectangular piece of bar stock.

The purpose of this paper is to examine the causes for these seemingly contradictory results. An attempts will be made here to review the previous studies to look into future possibilities of high pressure blade from the view point of datum and modified design.

TK Ghosh et.al [1] et al states that the limited primary energy sources and awareness of environmental pollution has led to ever increasing end over to develop new steam turbine power plants with the highest possible efficiency. Considering their output, even small step increase in efficiency can result in major saving for the customers. As overall cycle efficiency is strongly dependent on steam turbine performance, Continuous improvement are sought to increase the turbine efficiency. These effectors are directly primarily towards improvements are blading as the key component of the turbine. This paper presents the BHEL realness to meet the requirement of higher efficiency by adapting newer balding, which can substantially improve stages efficiency and hence overall performance of the turbine. Also it

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contains the new developments in the area of shrouds and sealing of flow of the part art manufacturing facilities establish at BHEL, Haridwar for manufacturing advance class blading.

Kenji Nakamura et.al [2] response to global environmental issues, higher efficiency and improved operational reliability are increasingly being requested for steam turbines, essential equipment for thermal power generation. By increasing the temperature and pressure of the steam turbine operating conditions, more efficient power generation is realized, and in order to realize a turbine applied with the higher temperature conditions of 700˚C for the future, Fuji Electric is participating in the METI-sponsored development of advanced ultra-supercritical power generation, and is evaluating and verifying the reliability of materials used for high-temperature valves. In addition, for geothermal steam turbines, Fuji has developed surface coatings and other technology for enhancing corrosion resistance in order to improve reliability. Fuji is also moving ahead with the development of geothermal binary power-generating turbines that utilize a low boiling point medium.

Zachary Stuck et.al [3] addresses steam turbine efficiency by discussing the overall design of steam turbine blades with a specific focus on blade aerodynamics, materials 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 paper 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 paper 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 will also be 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 reaction turbine blades are compared and contrasted, and the effect that these designs have on turbine efficiency are discussed. Based on the research presented herein, this paper presents a detailed summary of what modifications to existing steam turbine generator blades can be made to increase turbine efficiency. Finally, the overall sustainability of steam turbine generators is discussed and the impact that the design of the blades has on the sustainability of these generators is presented.

T. Mısek et.al [4] have developed 3000 rpm 1220 mm blade for a steam turbine was developed with application of new design features. The last stage moving blade is designed with an integral cover, a mid-span tie-boss connection and a fir-tree dovetail. With this configuration the blades are continuously coupled by the blade untwist due to the centrifugal force when the blades rotate at high speed, so that vibration control and increased structural damping are provided. Blade was tuned in order to Eigen frequencies were safely far from possible excitation. Because of connection members, the number of the resonant vibration modes can be reduced by virtue of the vibration characteristics of the circumferentially continuous blades. The last stage airfoil was optimalized from view of mineralization of its centrifugal force. In order to develop the 3000 rpm 1220 mm blade, the advanced analysis methods to predict dynamics behaviour of the bladed structure were applied. Coupled rotor-blade analysis was also aim of the attention. To validate calculated results the verification measurement such as rotational vibration tests was carried out in the high-speed test rig. Relation of the friction damping of the bladed structure on amount of excitation level was also monitored and evaluated.

Tulsidas.D et.al [5] have said large variety of turbo-machinery blade root geometries used in industry prompted the question if an optimum geometry could be found. An optimum blade root was defined, as a root with practical geometry which, when loaded returns the minimum fillet stress concentration factor. The present paper outlines the design modification for fillet stresses and a special attention is made on SCF of the blade root (T-root) which fails and to guarantee for safe and reliable operation under all possible service conditions. Finite Element Analysis is used to determine the fillet stresses and Peterson's Stress Concentration Factor chart is effectively utilized to modify the blade root. The root is modified due to the difficulty in manufacturing the butting surface of the tang which grips the blade to the disk crowns having small contact area.

Yasutomo Kaneko et.al [6] to improve the reliability and the thermal efficiency of HP (High Pressure) end blades of steam turbine, new standard series of HP blades has been developed. The new HP blades are characterized by the ISB (Integral Shroud Blade) structure. In the ISB structure, blades are continuously coupled by blade untwist due to centrifugal force when the blades rotate at high speed. One of the probable failure modes of the ISB structure seems to be fretting fatigue, because the ISB utilizes friction damping between adjacent shrouds and stubs. Therefore, in order to design a blade with high reliability, the design procedure for evaluating the fretting fatigue strength was established by the model test and the nonlinear contact analysis. This paper presents the practical design method for predicting the fretting fatigue strength of the ISB structure, and the some applications are explained.

B. Stanisaa et.al [7] has suggest to the erosion caused by wet steam flow reduces the efficiency of the last stage rotor blades of condensing steam turbines, and makes their service life shorter. To date there has been insufficient data on the erosion process which the steam turbine rotor blades are subject to during the operation, data which could be a basis for development and verification of mathematical models to estimate the service life of eroded rotor blades. This paper reviews the results of many years monitoring and researching of the laws of the erosion process and its mechanism for rotor blades of condensing steam turbines. On the basis of the obtained laws of the rotor blades erosion process and a simplified model their service life is estimated.

Christoph-Hermann Richter et.al [8] to provide an overview of the structural design of modern steam turbine blades at Siemens power generation using the finite element method. The different types of blades are described in detail regarding their geometry and loading. The modular building block approach of modelling is shown to be of essential importance. For the different analyses a fatigue post-processor has been implemented as well as an optimization tool. Both of these in-house codes will be briefly presented.

Pavlos K. Zachos [9] has investigates the effect of blade deformation, caused by manufacturing inaccuracies, on the performance of a 2-stage axial steam turbine. A high fidelity 3D coordinate Measurement Machine has been employed to obtain the exact geometrical model of the blades. A Streamline Curvature solver was used to predict the overall performance of the turbine. During the manufacturing process of the casts and of the blades themselves, several types of errors can occur which lead to a different geometry from that envisaged by the designer. The main objective of this study is to investigate the effect of those errors on the performance of a 2-stage experimental axial steam turbine. A high fidelity measurement of the actual geometry of both stator and rotor blades has been carried out, using a 3D Coordinate Measurement Machine. The cross sections of the blades obtained by the measurement were compared with those produced by the design process to evaluate the change in blade inlet/exit angles. In addition, the geometrical deviations from the initial design have been subjected to a statistical study in order to locate the nature of the error. The actual (measured) model has been used as input into a Streamline Curvature solver to evaluate its performance. Finally, a comparison with the performance plots of the original geometry has been carried out. A measurable change of efficiency as well as in the total power delivered by the turbine was found. This suggests that the accumulated error caused during the manufacturing procedure plays a significant role in the overall performance of the machine by making it less efficient by more than 1%. Reverse engineering techniques can be applied to predict and alleviate these errors leading thereby to a final design of each stage with improved performance.

The Experimental investigation of M. Ahmad et.al [10] on a droplet size influence on low pressure steam turbine blade erosion In the last stages of steam turbines, large droplet is generated from the wet steam flow. These droplets collide with the following rotating blades with almost the peripheral speed of the rotor. This high speed impact is perceived in the form of erosion of low pressure steam turbine blades. Among others, impacting droplet size is a key parameter contributing to the erosion of low-pressure steam turbine blades. At the Institute of Thermal Turbo machinery and Machinery Laboratory (ITSM) Stuttgart, the effect of droplet size on the erosion of steam turbine blade has been investigated with the help of an erosion test rig. The experiments confirm that the erosion increases with increasing droplet sizes. It is also found that volume loss per droplet impact increases with droplet size with a simple power law relation Erosion where value of n is found to be 3.2 up to 3.5 for common blade materials.

Behaviour of steam turbine blades of glass-epoxy Sandeep Soni et.al [11] erosion caused by wet steam flow reduces the efficiency of the last stage rotor blades of steam turbines and makes their service life shorter. Water droplet erosion is one of the major concerns in the design of modern steam turbine because it causes serious operational problems such as performance degradation and reduction of service life. An erosion model has been used in the present study for the prediction of water droplet erosion of rotor blades operated in wet steam conditions. It is used to analyze the erosion behavior of nickel coated glass epoxy steam turbine blades. The major erosion parameters to find incubation time is rate of mass loss under varying conditions of dryness fraction of steam (x) ,steam temperature (T), coating thickness (hc) and size of the water droplets(d) are involved in the model so that it can also be used for engineering purpose at the design stage of rotor blades. Results are showing greater improvement in the erosion characteristics like incubation period and rate of mass loss due to application of Ni coating on the glass epoxy blades. According to that suitable operational factors have been defined to obtain the best possible performance of steam turbines.

The results of investigation E. K. Sevidova et.al [12] to evaluation of the protective properties of multilayer coatings for steam turbine blades protective properties of multilayer ion-plasma coatings relative to the conditions of their exploitation on steam turbines are described. It was established that the protection properties of coatings on 20X13 steel in an aggressive NaCl environment of various concentrations increase according to the sequence [Cr + (Cr,Ti)N]10 < (Ti + TiN)10 < (Cr + CrN)10. It was also found that a breach in the coating integrity can lead to the appearance of macrogalvanic couples. Their activity considerably increases (by 4–5 times) during the mechanical passivation of the surface under the conditions of drop-collision erosive wear. The greatest values of the EMF in stationary conditions are generated between the 20X13 steel and Ti + TiN coating.

II. BLADE MATERIAL

Composition and Microstructure of Corrosion-Resistant High-Alloy Cast Steel

Alloy	**CA-6 NM (Chromium and Nickel)**
Microstructure	**Tempered Martensitic**

Room-Temperature mechanical properties of cast corrosion –resistant alloy.

Table I: Mechanical Properties of Cast Corrosion Resistant Alloys

Alloy	**CA-6NM**
Heat Treatment	>955OC, Air cool, Tempering
Tensile strength	827 MPA
Yield strength (0.2% offset)	689 MPA
Elongation in 50mm(2 in.),%	24
Reduction in area, %	60
Hardness (HB)	269
Charpy impact energy	94.9 J

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.

III.EXPERIMENTAL HIGH PRESSURE BLADE DESIGN

A. THEORY BEHIND STATIC ANALYSIS

In the static analysis we calculate the Centrifugal 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,

$$A = \text{annulus area} = [\pi r2/(t\text{-}r2n)]$$

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 was absorbed that in the above design, after the rotor blades begins designed they were analysed only for mechanical stresses but there was 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 were 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:

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

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:

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

Bending and twisting calculations:

Maximum tangential stress produced in shaft is equal to

$$T_{Max} = ½\, w \sqrt{}(Mb^2+Mt^2)w = 32\pi d$$

Where,

d is diameter of the shaft, and **Mb, Mt** are bending and twisting moments. Twisting moment at the chosen section is given by

$$Mt = 97300\ Ni/N$$

Where,

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

Maximum bending moment can be calculated graphically 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.

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C. 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.

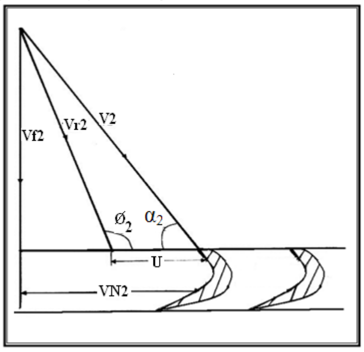

Fig.1: Inlet triangles for 1st stage rotor blade

From the inlet velocity triangle of a rotor blade we get,

Whirl velocity (V_{w2}) = 422.74m/s.
Flow velocity (V_{f2}) = 186.89m/s.
Relative velocity (V_{r2}) = 265.09m/s.
Blade angle at the inlet (θ_3) = 135.17^0

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.

D. EXIT VELOCITY TRIANGLE

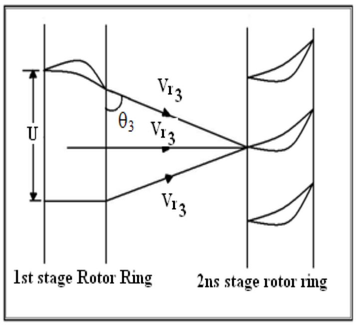

Fig.2: Exit velocity triangles for 1st stage rotor blade

At the exit of first stage rotor blades,

Flow velocity (V_{f3}) = 180.42m/s.
Relative flow angle (ψ_3) = 37.88^o
Whirl velocity (V_{w3}) = 2.805m/s.
Relative velocity (V_{r3}) = 293.83m/s.

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

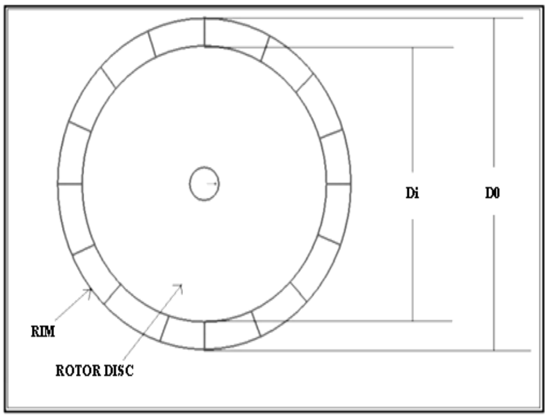

Fig.3: 1st Stage Rotors

Evaluation of gas forces on first stage rotor:

At the inlet of first stage rotor blades,

Absolute flow angle ($\alpha_{2)}$ = 22.85^{o}

Absolute velocity ($V_{2)}$ = 462.21m/s.

Dia of blade mid span (γ) = 1.3085m.

Design speed of turbine (u) = πDN/60.

Tangential force $F_t = m(V_{w2} + V_{w3})$Newton.

Axial force $F_a = m(V_{f2} + V_{f3})$Newton.

Where,

m is mass flow rate of gases through the turbine.

Referring to the above figure

$$M = \frac{(\rho 2\ (D0 - Di)Vf2)}{4}$$

Where,

ρ2 is the density of gases at the entry of first stage rotor,

$\rho 2 = 0.8900\ kg/m^3$,

m = 70.925 kg/s.

Total axial force on first stage rotor F_a=458.88N.

Total tangential force on each rotor F_t=29783.88N.

Number of blade passages in first stage rotor = 120.

Tangential forces on each rotor blade,

$$F_t = \frac{Ft}{no.of\ blade\ passages}$$

F_t=248.199N

Axial force on each rotor blade,

$$F_a = \frac{Fa}{no.of\ blade\ passages}$$

F_a =3.82N.

From Euler's Energy Equation,

Power developed in First stage rotor

$$P = m(V_{w2}\,U \pm V_{w3}\,U)$$

Using the above Equation

$$P = 6.991MW$$

The distance

$$X = \frac{(m1x1 + m2x2 + m3x3)}{(m1 + m2 + m3)}$$

Where,

m_1, m_2 and m_3 are masses of volume 1,2 and 3. $x_{1,}$ x_2 and x_3 distances of the centroids of volumes, 2 and 3 from the axis of revolution.

The density of material ρ is graphically measured to be:

$\rho = 8900kg/m^2$

$m_1 = 0.382$ kg

$m_2 = \rho \times V_2$

$m_3 = \rho \times V_3$

Where,

V_2 and V_3 are volumes of portions 2 and 3 of rotor blades,

The distance X is calculated as 648.85mm.

Total mass $M = m_1 + m_{2+}m_{3.}$

Centrifugal force $F_c = M(2\pi n/60)^2X$ and its value is found to be 38038.33N.

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IV. CONCLUSIONS

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.

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 CA-6 NM (Chromium Nickel).

Modified solutions for Steam turbine blade values to machines to maximize their reduce life cycle costs, efficiency, , and improve reliability.

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