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Investigation on Last Stage High Pressure Steam Turbine Blade for Producing Electricity

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Abstract

The investigation on design of high pressure steam turbine blade addresses the problems of steam turbine efficiency. A precise focus on aerofoil profile for high pressure turbine blade and it gages the effectiveness of certain like Chromium and Nickel in resisting creep and fracture in the turbine blades. The thermal and chemical conditions in blades are, 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 collisions of any coal-fired in power stations. The increasing efficiency of a typical 500 MW turbine by 1% reduces emissions of CO_2 from the turbine location, 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 in power plant. Based on the research presented modifications to high pressure steam turbine blades can be made to increase turbine efficiency a 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; Stresses in blade

Introduction

Blades are the heart of a steam turbine, as they are the elements that convert the thermal energy into kinematic energy. The efficiency and consistency of a turbine depends on the proper design of the blades. It is therefore necessary for engineers involved in the steam turbines field to have an overview of the importance and the basic design aspects of the steam turbine blades. The blade design is a multi-disciplinary task. It involves like thermodynamic, aero-dynamic, mechanical and material science restraints. The total development of a new blade is thus possible only when experts of all these fields come together as a panel. Efficiency of the turbine is depends on the parameters like, Inlet and outlet angle of blade, blade Materials, blade profile and Surface finishing of the blade and etc.

The most critical aspect of steam turbine reliability centre on design of buckets. Buckets or rotating blades are subjected to unsteady steam forces during operation, the phenomenon of vibration significance must be measured. Resonance happens when an exciting frequency coincides with a natural frequency of the system. At timbre conditions, the amplitude of vibration is related primarily to the amount of stimulus and damping present in this system. A high bucket reliability requires design with minimum resonant vibration. The design procedure starts with precise calculation of bucket natural frequencies in the tangential, axial, tensile, and complex modes, which are verified by in the data. In addition, improved aerodynamic nozzle shapes and generous stage axial clearances are used to reduce stimulus bucket. Bucket covers are used on some of stages or all stages to attenuate prompted vibration. These design practices and composed with advanced precision manufacturing techniques, ensured the necessary bucket reliabilities. Almost all of the blading used in modern mechanical drive steam turbines is either of drawn or milled type construction and drawn blades are machined from extruded airfoil shaped pieces of material stock of material. 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 attempt 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.

Literature Survey

Ghosh and Bansal [1] 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 increase in efficiency can result in saving for the customers. As overall cycle efficiency is strongly dependent on steam turbine performance, Constant development are sought to increase turbine efficiency. These effects are directly primarily towards improvements are blading as the key component of the turbine. This paper presents the BHEL to meet the requirement of higher efficiency by adapting newer blading, which can substantially improve stages efficiency and hence overall performance of the turbine

Kenji Nakamura [2] response to global environmental productions, higher efficiency and improved operating reliability are increasingly being requested for steam turbines are vital role for thermal power plants, by increasing the temperature and pressure of the steam turbine working conditions, more efficient power generation is recognized, and in order to realize a turbine applied with the higher temperature conditions of 700°C for the upcoming, Fuji Electric is contributing in the METI-sponsored development of advanced ultra-supercritical

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power generation, and is evaluating and verifying the reliability of materials used for high temperature controllers. In addition to geothermal steam turbines, Fuji has developed surface coatings and other technology for enhancing corrosion resistance in order to develop reliability. Moreover Fuji is moving ahead with the development of geothermal binary power-generating turbines that utilize a low boiling point medium.

Zachary Stuckl [3] addresses steam turbine efficiency by discussing the overall design of steam turbine blades with a specific focus on blade aerodynamics, materials are used in the manufacture 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. 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 existing. The aerodynamic designs of impulse and reaction turbine blades are compared and contrasted, the effect of those designs have on turbine efficiency are debated. Based on the research unfilled herein, this paper presents a complete summary of what modifications to existing steam turbine generator blades can be made to increase turbine efficiency.

Mısekl [4] have developed 3000 rpm 1220 mm blade for a steam turbine was developed with application of new design structures. The last stage stirring blade is designed with an integral cover, a mid-span tie-boss connection with a fir-tree dovetail joint. 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 structural damping are provided. The last stage airfoil was optimalized from view of mineralization of its centrifugal force. In order to develop a speed of 3000 rpm 1220 mm blade, the advanced analysis methods to predict dynamics behaviour of the bladed structure were applied. To validate calculated results the verification measurement such as rotational vibration tests was carried out in the high speed test rig. The relation of the friction damping of the bladed structure on amount of excitation level was also monitored and evaluated.

Tulsidas [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 stood defined, as a root with useful geometry when loaded returns the minimum stress concentration factor at the fillet. The present paper outlines the design modification for fillet stresses and a special attention is made on SCF of the blade root. Finite Element Analysis is used to determine the fillet stresses and Peterson's Stress Concentration Factor chart is effectively utilized to modify root of blade. The root is modified due to difficulty in manufacturing the butting surface of the tang which grips the blade to the disk crowns having small contact area.

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

Stanisaa [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 tinier. Today there is insufficient data on the erosion process which the steam turbine rotor blades are subject to during the working data which could be an origin for development and verification of mathematical models to estimate the service life of eroded rotor blades. This rag 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 beginning of the obtained regulations of the rotor blades erosion process and a simplified model their service life is estimated.

Christoph-Hermann Richter [8] to provide an overview of the structural design of modern steam turbine blades at Siemens power generation using the FE method. The altered types of blades are described in detail regarding their geometry of loading. The segmental building block approach of modelling is shown to be importance for the different analysis, a fatigue post-processor has been applied as well as an optimization tool. Both of these are in-house codes be briefly presented.

Pavlos K. Zachos [9] investigated the effect of blade twist, caused by developed inaccuracies, on the performance of a two stage axial steam turbine. A high reliability 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 turbine. In the manufacturing process of the casts and of the blades themself several types of errors can happen which lead to a different geometry envisaged by the designer. A high fidelity measurement of the actual geometry of both stator and rotor blades has been carried out. Finally, a comparison with the performance plots of the original geometry has been accepted. A assessable change of efficiency as well as in the total power delivered by the turbine was originated. This proposes 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.

Ahmad [10] on a droplet size influence on low pressure steam turbine blade erosion in the last stage of steam turbines, huge droplets is generated from the flow of wet steam. These droplets collide with the following rotating blades with almost the peripheral speed of the rotor. This high speed impact is observed 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 Institute of Thermal Turbo machinery and Machinery Laboratory Stuttgart, the effect of droplet size on the erosion of steam turbine blade has been investigated with the help of a corrosion test rig. The experiments confirmed that the erosion increases with increasing droplet sizes.

Sandeep Soni [11] wet steam flow reduces the efficiency of the last stage rotor blades of steam turbines and makes their service shorter life. Water droplet corrosion 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. A 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 behaviour of nickel coated glass epoxy steam turbine blades. The major erosion parameters to find growth time is rate of mass loss under varying conditions of dryness fraction of steam (x) ,steam temperature (T), coating thickness 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 and these results are showing better improvement in the erosion characteristics like incubation period and rate of mass loss due to application of Ni coating on the glass -epoxy blades. Accordingly to that suitable operational factors have been defined to obtain the best possible performance of steam turbines.

Sevidova [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 defined. 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 maximum values of the EMF in stationary conditions are generated between the 20X13 steel and Ti + TiN coating.

Blade Material

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

Alloy: CA-6 NM (Chromium and Nickel)

Microstructure: Martensitic Tempered Overall (Table 1), it is the material properties that make a blade consistent 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.

Experimental high pressure blade design: 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, $F_c = \dfrac{\rho A A_a}{2\pi(\omega)^2}$

Where, A=annulus area= $\dfrac{2\pi r}{(t-2rn)}$

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

Table 1: Mechanical Properties Alloys.

The centrifugal forces at the blade root section is the centrifugal force divided by area of the blade section at the root. $\sigma_c = \dfrac{\rho A_a}{2\pi(\omega)^2}$

Theory behind thermal analysis

The design features of the turbine segment of the steam turbine have been taken from the "Preliminary design of a control turbine. It was observed 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 the temperature takes 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 [13-15].

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 the 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 its length, the stresses at the weaker section are:

$$\sigma = \frac{C_o}{F_o} = \frac{C_b + \Sigma C_s}{F_o}$$

Where, Co=Centrifugal forces of blade, shroud etc,

Fo=area of the weakest blade section (root section).

Centrifugal forces of constant section blade will be:

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

Where, G=weight of the blade

h=height of the blade,

$\gamma_{xy=}$mean diameter

ω=angular velocity.

The centrifugal forces of the shrouding are obtained as:

$$C_s = G_s r_s \frac{\omega^2}{g} = F_s \gamma r_s \omega^2 \frac{I_s}{g}$$

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

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r is radius of the strip centroids.

The centrifugal forces of binding wire: $C_w = G_w r_w \frac{\omega^2}{g} = F_w I_w r_w \gamma \frac{\omega^2}{g}$

Bending and twisting calculations: Maximum tangential stress produced in the shaft

$$T_{\max} = \frac{1}{2} W \sqrt{(M_b)^2 + (M_t)^2 W} = 32\pi d$$

Where,

d=diameter of the shaft

M_b=bending moment

M_t=twisting moment

Twisting moment at the chosen section is given by

$$M_t = 97300 \frac{N}{n}$$

Where,

N=total power developed in KW and n speed of turbine in RPM.

Maximum bending moment can be calculated graphically by shear force bending moment diagrams. For obtaining the stresses the T shaped root node degree of freedom are constrained in the U_X, U_Y and U_Z directions and tangential, axial and centrifugal forces are applied at the centroid. The axial and tangential forces results from the gas momentum change and from pressure differences through the blades, which are evaluated by creating velocity triangles at the inlet and outlet of the rotor blades (Figure 1).

Inlet velocity triangle

The tangential and axial 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.

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

Whirl velocity V_{w2}=422.74 m/s.

Flow velocity V_{f2}=186.89 m/s.

Relative velocity V_{r2}=265.09 m/s.

Blade angle at the inlet (θ_3)=135.17^0

The tangential and axial force results from the gas momentum changes from pressure difference through the blades, which are

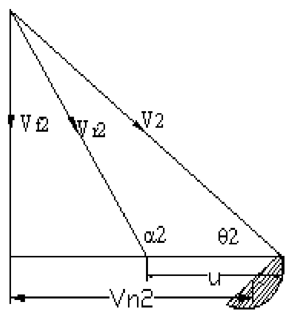

Figure 1: Inlet triangles for 1st stage rotor blade.

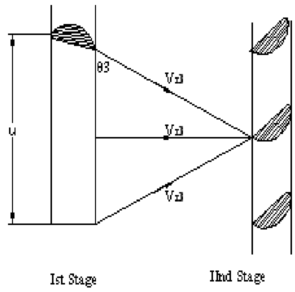

Figure 2: Exit velocity triangles for 1st stage rotor blade.

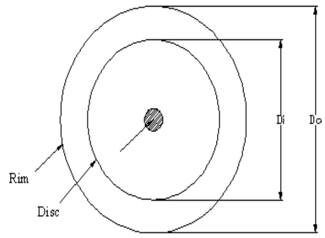

Figure 3: 1st Stage Rotors.

calculated by creating velocity tangles at the inlet and outlets of the rotor blades.

Exit velocity triangle

At the exit of first stage rotor blades,

Flow velocity V_{f3}=180.42 m/s.

Relative flow angle ψ_3=37.88°

Whirl velocity V_{w3}=2.805 m/s.

Relative velocity V_{r3}=293.83 m/s.

Please refer above Figure 2 for more understanding.

Evaluation of Tangential (Ft), Axial force (Fa) and Centrifugal force (Fc) on each rotor:

(A) Calculation of gas forces on first stage rotor:

At the inlet of first stage rotor blades,

Absolute flow angle (α_2)=22.850

Absolute velocity (V_2)=462.21 m/s.

Dia of blade mid span (γ)=1.3085 m.

Design speed of turbine (u)=πDN/60.

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

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

Where, m is mass flow rate of gases through the turbine.

Referring to the above Figure 3.

$$M = \frac{\rho_2 (D_0 - D_i) V_{f2}}{4}$$

Where, ρ_2=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, $F_t = \dfrac{F_t}{No.of.blade.passages}$ =248.199N

Axial force on each rotor blade, $F_a = \dfrac{F_a}{No.of.blade.passages}$ =3.82N.

From Euler's Energy Equation,

Power developed in First stage rotor

P=m (V_{w2} U + V_{w3} U)

Using the above Equation

P=6.991MW

The distance $X = \dfrac{(m_1 \times x_1 + m_2 \times x_2 + m_3 \times x_3)}{(m_1 + m_2 + m_3)}$

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, 1, 2 and 3 from the axis of revolution.

The material density ρ is graphically measured to be:

ρ=8900 kg/m^3

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 and is 648.85 mm.

Total mass M=m_1+ m_2+m_3.

Centrifugal force Fc =M(2πn/60)^2

X=38038.33N.

Conclusions

The implementation of robust turbine blades are designed in accordance with the modern material technologies and able to withstand the most of circumstances, in combination with the use of clean and renewable fuel presents an efficient method of generating substantial amount of electricity. An enhanced blade design, focused on resisting the effects of stresses, corrosive agents, and creep-inducing temperatures, will raise the turbine efficiency, consequently leading to an increase in the power plant overall efficiency reduction of the quantity of fuel consumed, and ultimately a decrease in operational costs. To improve efficient blade design, will help to reduce operating costs even further and minimize the environmental impacts of steam turbines. Generally such a combination of technologies would benefit society by providing an effective, viable, and invulnerable means of generating electrical energy.

This example demonstrates that under normal operating conditions, stresses are sound and tinny 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 viewpoint, 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 Von-Mises 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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