



International Journal of Engineering, Business and Enterprise Applications (IJEBA)

www.iasir.net

DESIGN OPTIMIZATION AND STATIC & THERMAL ANALYSIS OF GAS TURBINE BLADE

Ganta Nagaraju¹, Venkata Ramesh Mamilla², M.V.Mallikarjun³

¹M.Tech Student, ²Associate Professor, ³Professor

Department of Mechanical Engineering
QIS College of Engineering & Technology
Ongole – 523272, Andhra Pradesh, India.

Abstract: This paper studies about the design and analysis of Gas turbine blade, CATIA is used for design of solid model and ANSYS software for analysis for F.E. model generated, by applying boundary condition, this paper also includes specific post-processing and life assessment of blade. How the program makes effective use of the ANSYS pre-processor to mesh complex turbine blade geometries and apply boundary conditions. The objective of this paper is to get the natural frequencies and mode shape of the turbine blade.

Keywords: Design; static Analysis; Thermal analysis; Gas Turbine Blade; ANSYS

I. INTRODUCTION

The break down and failures of turbo machineries have been influencing such as consequential damages, hazards to public life and most importantly the cost to repairs. To avoid these, it is obvious that the blading of turbo machinery must be made structurally stronger, that means not in dimensions and/or use of materials of construction, but keeping the operating stresses well within the limits.

Turbo machinery blades are classified into two categories depending on their manner of operation as either impulse or reaction blades. Impulse blades function by redirecting the passing fluid (steam or gas) flow, through a specified angle. A work producing force is developed by resulting change of momentum of passing fluid flow.

Reaction blades function as airfoils by developing a gas dynamic lift from the pressure difference, which the airfoil causes, between the blades upper and lower surfaces. High-pressure stages are generally impulse stages and low-pressure stages are reaction stages. Thus, a single free standing blade can be considered as pre-twisted continuous beam with an asymmetric airfoil cross-section mounted at a stagger angle on a rotating disc.

Reasons for failure of bladed disc:

- Excessive stresses
- Resonance due to vibration
- Operating environmental effects.

Ever increasing demands of high performance together with reliability of operation, long life and lightweight necessitate consistent development of almost every part of steam turbine blades from a vital part of a turbo machine. Apart from their shape and geometry, on which the performance characteristics of the machine largely depend, their dynamic strength is of considerable importance as far as the reliability operation and life of the engine are concerned. High cycle fatigue plays a significant role in many turbine blade failures. During operation, periodic fluctuations in the steam force occur at frequencies

Corresponding to the operating speed and harmonics and cause the bladed disk to vibrate. The amplitude of these vibrations depends in part of the natural frequencies of the bladed disk to the forcing frequency. Large amplitude vibration can occur when the forcing frequency approaches or becomes resonant with the natural frequency of the blades. Dynamic stresses associated with near resonant or resonant vibration produce high cycle fatigue damage and can initiate and propagate cracks very quickly. Steam turbine manufacturers typically design and manufacture blades with adequate margins between the forcing frequencies and the fundamental natural frequencies to avoid resonance.

A. FAILURES IN STEAM TURBINE BLADES

In the following paragraphs various failure modes of the turbine blade are discussed along with different kinds of stresses in the blade and the nature of aerodynamic excitation. A brief discussion of each of the above failure mechanisms follows in order to understand their significance.

EXCESSIVE STRESS:

The total stress at any location of the blade is, sum of the centrifugal tension, centrifugal bending, steady steam bending and the alternating bending. The amplitude of alternating bending depends on the dynamic bending force, damping factor and the resonant frequency. Each of these is briefly discussed below to highlight their importance.

CENTRIFUGAL STRESS:

In steam turbine, centrifugal stress is never the main cause of a blade failure, except in the rare cases of turbine run-away or due to low cycle fatigue caused by frequent start ups/shut downs. However, centrifugal stress is an important contributing factor with fatigue failure, corrosion fatigue failure and stress corrosion failures. The level of centrifugal stress is kept at such a level so as to have enough margins for alternating stress. The blade configuration is designed so as to keep the center of gravity of shroud, airfoil and root attachments, on a common radial axis. This prevents centrifugal induced torsion stresses. Using any of the standard FEM packages can best carry out analysis of centrifugal stress.

STEAM INDUCED STRESS – STEADY STATE:

Steam being the driving force, exerts loads on the blade due to steam pressure from the pressure side of the blade profile thereby inducing bending stresses. This bending stress is superimposed on the centrifugal tension. Although the net value of steady bending load and centrifugal stress alone is not normally a cause of blade failure, this is the basis for the majority of failures arising due to vibratory stresses superimposed upon them.

STEAM INDUCED STRESS – ALTERNATING:

Steam induced alternating stress can be induced by interrupted arc of admission, nozzle wake at nozzle passing frequency, mismatch of diaphragm nozzle at horizontal split and/or missing diaphragm blades, nozzle pitch variation, poor nozzle profile etc.

Estimation of the vibratory stress caused by these requires the analysis of the response of the blade to the excitation forces caused by these. Obviously the inputs to such an analysis are the excitation levels, the damping, natural frequency and the mode shape of the blade vibration. Disk vibrations, which can get excited due to variety of reasons, can also cause high stresses in the blade roots leading to failure. This aspect of blade stressing is quite involved and many a time difficult to compute correctly.

IMPACT STRESS:

Impact stress arises due to entrapment of a foreign body such as broken valve spindles, strainers etc. This may result in a chain reaction and can cause failure in many downstream rotating and stationary blades. Impact stress also results due to water ingress and at times due to steam hammer and from rubbing of blades in the event of failure of the thrust bearing.

LOW CYCLE FATIGUE:

Low cycle fatigue is caused by frequent start/stop operations, thermal cycling and frequent water slugging or water washing, due to inadequate water drainage in the casing and can cause failures within few hundreds to a few thousands stress cycle.

THERMAL FATIGUE:

Thermal fatigues is a low cycle fatigue process as a result of thermal stress caused due to quick starting, rapid and frequent load changing, steam temperature cycling and water slugging. Considerable thermal stresses are generated due to the temperature differential since blade foil is in direct contact with the nozzle upstream temperature whereas much cooler spent steam cools shroud, root and disk.

CREEP STRESS:

Creep stress failures are rare but cracking at locations with high stress concentrations may take place, if design is inadequate. Hence high temperature stages should have larger fillet radii at the root.

RESONANT VIBRATION:

Vibration is important in designing turbine blades/disk since resonant vibratory stresses, sustained over a period of time, can cause fatigue failures. The period of time need not be large since a 500Hz vibration accumulates 2 million cycles in an hour and 108 cycles in 2 days. Bladed disk poses a worst vibratory fatigue problem since it is directly exposed to a wide range of aerodynamic excitation and failure can result when any of the following matching takes place.

A rotating turbine blade (bucket) is the components, which converts the energy of flowing fluid into mechanical energy. Thus the reliability of these blades is very important for successful operation of turbine. Metallurgical examinations of failed blades show that almost all the failures can be attributed to the fatigue of metal. Fluctuating forces in combination with the steady forces cause fatigue failure. Turbo machine experience fluctuating forces when they pass through non-uniform fluid from stationary vanes (nozzle). The basic design consideration is to avoid or to minimize the dynamic stresses due to fluctuating force. Based on the vibration of the mechanical structures, the dynamic behavior of turbine blade, blades or the bladed disc assembly can be predicted. The present work presents an approach for modeling of blades, bladed disc & for its vibration analysis. Generally turbo machine low-pressure stage blades are long, twisted & tapered, so it needs lot of input data to accurately define the complete geometry. Blade geometry is defined by giving different profile data at Different heights. To reduce the pains for creation of solid model of this type of blades, a program file called as macro, was developed in ansys command. From these macros solid model was created.

Finite element method for the single blade solid model was created with eight noded quadrilateral brick element in ansys.

Two types of analysis was done for this present work

- Free standing blade static analysis
- Free standing blade model analysis.

The maximum number of nodal diameters in bladed disc assembly is half the number of blades (for an even number of blades). For the disc having an odd number of the blades, the maximum nodal diameter is (number of blades-1)/2

For free standing blade root supports stiffness effect was studied and frequency at different rotating speeds also calculated Analytical work to determine the blade characteristics calls for accurately modeling the geometry.

II. DESIGN PROCEDURE

A turbo machine blade is usually a cantilever beam or plate is tapered and twisted with an airfoil cross-section. Typically a turbo machine has several stages, each stage with a stator and rotor. In the stator, they are all inserted as diaphragms or nozzles in a ring to guide the flow medium at an appropriate entry angle into rotor blades. The rotor blades are mounted on a disc at a stagger angle to the machine axis and they convert the thermal energy into mechanical energy in turbine. In turbine steam enters at high pressure and temperature in the first stage and expands while passing through the several stages before it is let out from the last

Stage with low temperature and pressure after extracting as much as thermal energy as possible. Hence, the short blades in high pressure have high frequency of the order of 1000Hz, which becomes progressively lower about 100Hz in the last stage long blades. In the compressor stage, the operation principle is reversed to compress the gases utilizing the supplied mechanical power.

A typical rotor blades sees upstream disturbances from the stator row and as it rotates, receives a corresponding number of increasing and decreasing lift and moment alternating periodically depending on the number of stator blades/nozzles/guide vanes. A stator blade can also be imagined to rotate in an opposite direction to the rotor relative to the moving row and thus receives a corresponding number of periodic forces and moments equal to the rotor blades. An ideal placement of blades in the stator is not feasible in practice. Firstly, the blades are not all identical in their cross section along the length, their pitch distance from blade to blade varies, and the axial and angular locations will have some errors in mounting them in the stator housing. Because of these errors in the stator, mechanical excitation at rotational speed and its harmonics occurs on the rotor blades.

NATURAL FREQUENCY AND MODE SHAPE: Natural frequency is the frequency at which an object vibrates when excited by a force, such as a sharp blow from a hammer. At this frequency, the structure offers the least resistance to a force and if left uncontrolled, failure can occur. Mode shape is the way in which the object deflects at this frequency. An example of natural frequency and mode shape is given in the case of a guitar string. When struck, the string vibrates at a certain frequency and attains deflection shape.

The frequency can be noted by the pitch coming from the string. Different string geometries lead to different natural frequencies or notes. By nature of its structure, a turbine blade has many natural frequencies and mode shapes. These frequencies and mode shapes are somewhat further complicated by the use of shroud to connect group of blades together.

III. STATIC ANALYSIS

A. GRAPHITE

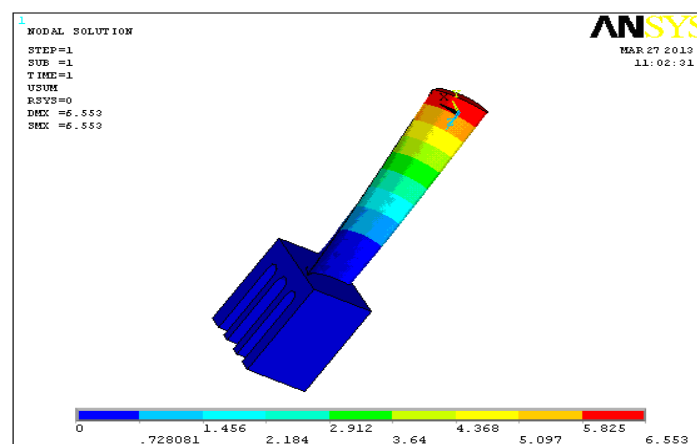


Figure.1: VONMISES STRESS VALUE

B. STAINLESS STEEL

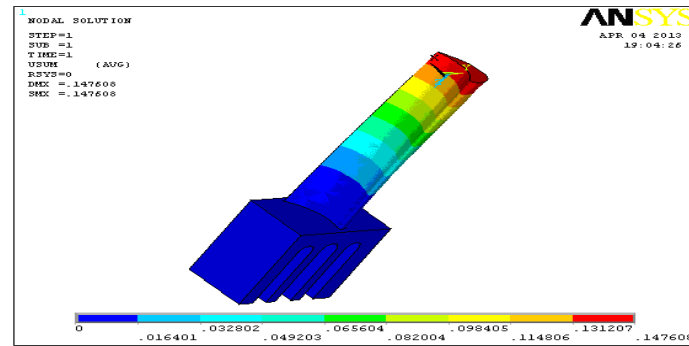


Figure.2 : VARIATION OF GEOMETRY SHAPE AFTER APPLYING LOADS.

C. TITANIUM

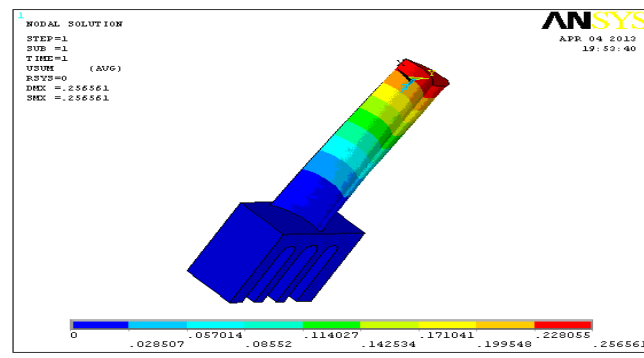


Figure.3 : VARIATION OF GEOMETRY SHAPE AFTER APPLYING LOADS.

IV. THERMAL ANALYSIS

A. GRAPHITE

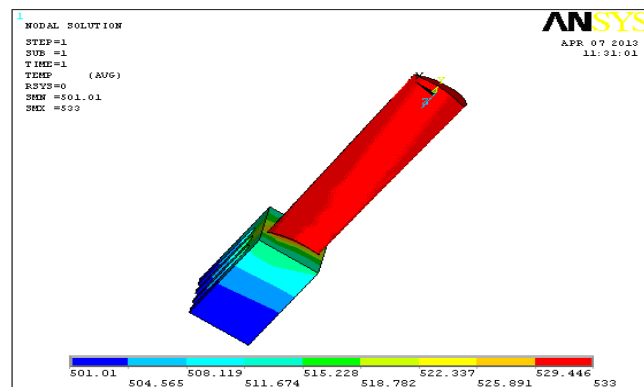


Figure.4: SHOWS THE TEMPERATURE DISTRIBUTION

B. STAINLESS STEEL

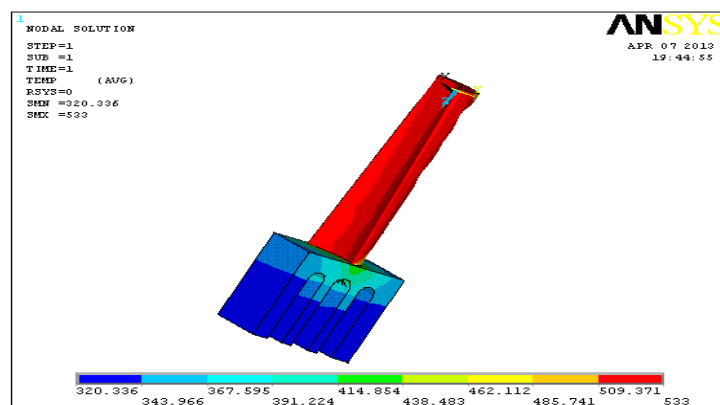


Figure.5: SHOWS THE TEMPERATURE DISTRIBUTION

4.3 TITANIUM

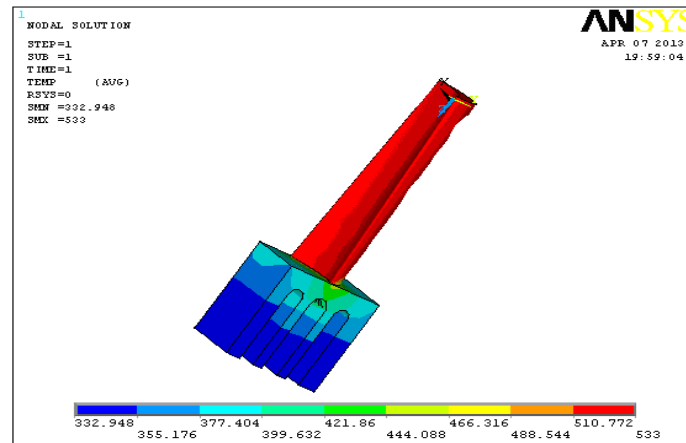


Figure.6: SHOWS TEMPERATURE DISTRIBUTION

V. RESULTS

Structural analysis				
	Graphite	stainless steel	Titanium	With coating
Stress	352147	339.07	327.817	258.892
displacement	6.553	0.147608	256561	.0430927
Thermal analysis				
Temperature	533	533	533	533
Thermal gradient	76.638	715.055	605.119	72.591
Thermal flux	19.16	11.727	10.287	15.97

VI. CONCLUSION

In this paper, analyzed previous designs and generals of turbine blade to do further optimization, Finite element results for free standing blades give a complete picture of structural characteristics, which can utilized for the improvement in the design and optimization of the operating conditions.

- Designed turbine blade using CMM data from existing model.
- Study on different materials which are suitable for the improvement of turbine blade.
- Different materials for turbine blade to suggest best material.

From the above results ,it conclude that using cast iron with partially stabilized zirconium coating is more beneficial than previous materials, due to low stress displacement, good thermal strength, low cost and easy to manufacture.

REFERENCES

1. Rao.J.S; application of variational principle to shrouded turbine blades, proceedings of 19th cong. ISTAM, 1974, pp: 93-97.
2. Leissa .A.W,Macbin.J.C and Keilb.R.E,Vibration of twisted cantilever plates summary of provisions; current studies, Journal of sound and vibration, 1984,vol-96(20),pp:159-167.
3. Tsuneo Tsuji,Teisuke sueoka , vibrational analysis of twisted thin cylindrical panels by using Raleigh-Ritz method ,JSME international Journal, Series iii,1990,volume 33,pp:501-505.
4. Le-chung shiau , Teng -yuan Wu,Free vibration of buckled laminated plates by finite element method ,Transactions of the ASME ,journal of vibrations and Acoustics October 1997,volume 111, pp:635-644.
5. Hu,XX and T.Tsuiji free vibrational analysis of curved and twisted cylindrical thin panels ,journal of sound and vibration, Jan 7,1999,vol-219(1) ,pp:63-68.
6. Yoo .H.H,J.Y.Kwak and J.chung ,Vibrational analysis of rotating pre twisted blades with a concentrated mass, Journal of sound and vibration, mar 2001,vol 240(5),pp:891-908.