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Structural and Thermal Analysis of Steam Turbine Blade Using FEM



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ABSTRACT

In the present work the first stage rotor blade of a two-stage Steam turbine has been analyzed for structural, thermal using ANSYS 15, which is a powerful Finite Element Software. In the process of getting the static and thermal stresses, the fatigue life in the rotor blade has been improved using this software.

The first stage rotor blade of the Steam turbine has been analyzed for the static and thermal stresses resulting from the tangential, axial and centrifugal forces. The Steam forces namely tangential, axial were determined by constructing velocity triangles at inlet and exist of rotor blades. The rotor blade was then analyzed for the temperature distribution. For obtaining temperature distribution, the convective heat transfer coefficients on the blade surface exposed to the Steam have to feed to the software. After containing the temperature distribution, the rotor blade was then analyzed for the combined mechanical and thermal stresses and also the fatigue life.

Steam turbine is an important functional part of many applications. Reducing the stresses and increasing the fatigue life is the major concern since they are in high temperature environment. Various techniques have been proposed for the increase of fatigue life and one such technique is to have axial holes along the blade span. Finite element analysis is used to analyze thermal and structural performance due to the loading condition, with material properties of structural steel.

Three different models with different size of holes (2, 3, and 4mm) were analyzed to find out the optimum number of holes for good performance. Graphs are plotted for stresses for existing design (7 holes) and for fatigue sensitivity against size of the holes (2, 3, and 4 mm). It is found that when the number of holes of the blades is increased, the stresses are reduced and number of cycles are increased. Thus, the blade configuration with 7 holes of 2mm size is found to be optimum solution.

1. INTRODUCTION

Turbo machine rotor blades are subjected to different types of loading such as fluid or gas forces, inertia loads and centrifugal forces. Due to these forces various stresses are induced in rotor blades. So stress and strain mapping on a rotor blade provide a vital information concerning the turbo machine design and lead to the detection of critical blade section. Analysis of static and dynamic behavior of a rotor blade is a basic problem in aero elasticity of turbo machine blades. The present paper deals with the stress analysis of a typical blade made up of nickel super alloy, which is subjected to centrifugal loading. The analysis results shows that stress is sever due to centrifugal forces compared that due to dynamic gas forces. Here in this case the effect of thickness, twist and taper of the blade was considered at the root of the blade where generally failure is occurring. The various blade shapes viz. rectangular, aerofoils with some angle twist, taper aerofoil are taken into consideration. In this paper linear static analysis for determining von-mises stresses, deformation in Z direction was determined using Finite element analysis software. The Solid brick 20-node element is used.

Aero engine turbine and compressor blades operate at speed range 5000 to 15000 r.p.m with temperature ranging from 50 to 900 degree centigrade. Hence depending on the stage of operation, blade material is usually an AL alloy, stainless steels, titanium alloys and nickel-based alloys. The tolerances on the blades are usually in the range of 0.05 mm to 0.15 mm on the aerofoil. The blades have a complex aerofoil structure and with varying aerofoil shape at different sections along the length of blade. There is always a twist in the aerofoil sometimes of the order 60 degrees. These complex configurations are required as the gases are to be smoothly guided along the different stages of the compressor and turbine without turbulence to achieve maximum thrust from the engine. Fatigue failure result from a combination of steady stress, vibratory stress, and material imperfections. However, the size of microscopic imperfections is difficult to control. Hence, stress-range diagrams are used to quantify the allowable vibratory stress amplitudes to avoid fatigue damage. Advanced turbo machinery blading is designed to have high steady stress levels. Thus, HCF occurs because of high mean stress - low amplitude vibratory loading of the airfoils. It is often initiated by the formation of small microscopic cracks.

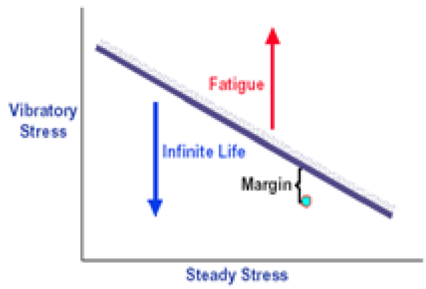

Fig. 1.1 Stress Range diagram

Advanced turbo machinery blading is designed to have high steady stress levels. Thus high cycle fatigue occurs because of high mean stress low amplitude vibratory loading of aero foils, as shown.

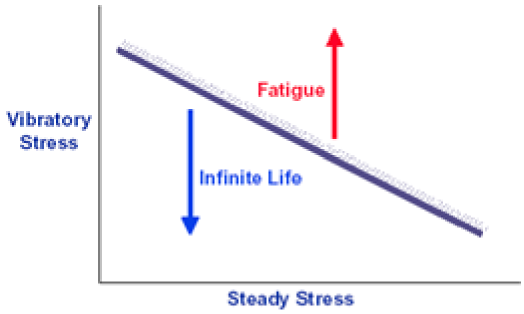

Fig. 1.2 Advanced Airfoil Stress Range diagram

Due to development of computers and subsequent development of numerical methods, it is now possible to model the components, simulate the conditions and perform testing on computer without actual model making, one of the most popular numerical methods used is the Finite Element (FEM) offered by the existing CAD/CAM/CAE. The most popular software, which is based on Finite Element Analysis, is "ANSYS" package, which is used in this work.

2. METHODOLOGY

The purpose of turbine technology are to extract the maximum quantity of energy from the working fluid to convert it into useful work with maximum efficiency by means of a plant having maximum reliability, minimum cost, minimum supervision and minimum starting time. The gas turbine obtains its power by utilizing the energy of burnt gases and the air which is

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at high temperature and pressure by expanding through the several rings of fixed and moving blades. To get a high pressure of order 4 to 10 bar of working fluid, which is essential for expansion a compressor, is required. The quantity of ten working fluid and speed required are more so generally a centrifugal or axial compressor is required. The turbine drives the compressor so it is coupled to the turbine shaft. If after compression the working fluid were to be expanded in a turbine, then assuming that there were no losses in either component, the power developed by the turbine can be increased by increasing the volume of working fluid at constant pressure or alternatively increasing the pressure at constant volume. Either of these may be done by adding heat so that the temperature of the working fluid is increased after compression. To get a higher temperature of the working fluid a combustion chamber is required where combustion of air and fuel takes place giving temperature rise to the working fluid.

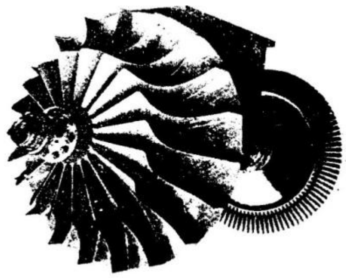

Fig.2.1 Turbine Blade Coupled to Centrifugal Compressor

WHAT IS THE SOLUTION?

In the solution phase of the analysis, the computer takes over and solves the simultaneous equations that the finite element method generates. The results of the solution are:

a) Nodal degree of freedom values which form the primary solution and

b) derived values, which form the element solution. The element solution is usually calculated at the element integration points.

Several methods of solving the simultaneous equations are available in the ANSYS program, frontal solution, sparse direction solution, Jacobi Conjugate Gradient solution, Precondition Conjugate Solution and an automatic iteration solver option. The frontal solver is the default.

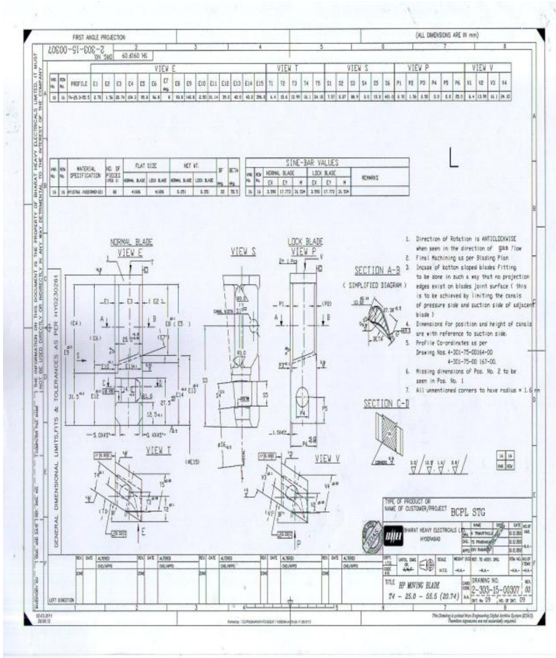

Fig.2.2 Turbine Blade Sheet (reference by BHEL)

Evaluation of Gas Forces on the First Stage Rotor Blades

At the inlet of the first stage rotor blades,

Absolute flow angle $\alpha_2 = 23.85^0$

Absolute velocity $V_2 = 462.21$ m/s

The velocity triangles at inlet of first stage rotor blades were constructed as shown.

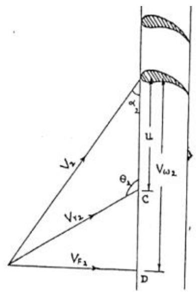

Fig.2.3 Inlet Velocity Triangles of I-stage Rotor Blades

Diameter of blade midspan, D = 1.3085m.

Design speed of turbine, N = 3426 r.p.m.

Peripheral speed of rotor blade at its midspan,

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$U = \pi DN/60$ ....(1)

From the velocity triangles in Fig. 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 inlet $\theta_2 = 135.017^0$

At the exit of first stage rotor blades,

Flow velocity $V_{f3} = 180.42$ m/s
Relative flow angle $\theta_2 = 37.88^0$

The velocity triangles were constructed at the exit of the first stagerotor blades as shown.

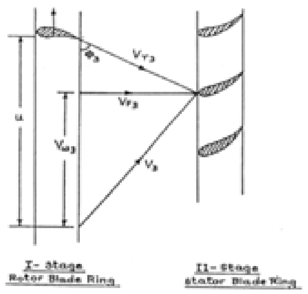

Fig.2.4 Exit Velocity Triangles of I-Stage Rotor Blades

From the velocity triangles (Figure), we get

Whirl velocity $V_{w3} = 2.805$ m/s
Relative velocity V_{r3} = 293.83 m/s

Finding Tangential Force (F_t) and Axial force (F_a) on each Rotor

Tangential force in N

$F_T = M (V_{w2} - (+V_{w3})]$ ....(2)

Axial Force in N

$F_A = M (V_{f2} - (+V_{f3})]$ ....(3)

Where M represents mass flow rate of gases through the turbine in kg/s.

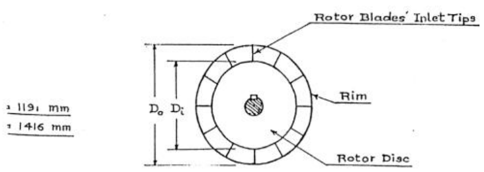

Fig.2.5 First stage Rotor

Referring to Figure

$M = \rho_2 \times \pi (D_o\text{-}D_i)/4 \times V_{f2}.$ ....(4)

Where ρ_2 is the density of gases at the entry of first stage rotor

$\rho_2 = 0.8234$ kg/m^3

Using equation (4),

M = 70.925 kgs/sec

Using equations (2) and (3)

Total tangential force on first stage rotor

$F_T = 29783.88$ N.

Total axial force on first stage rotor

$F_A = 458.88$ N.

Number of blade passages in first stage rotor = 120

Tangential force on each rotor blade

$$F_t = \frac{F_T}{\text{No. of blade passages}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

Axial forced on each rotor blade

$$F_a = \frac{F_A}{\text{No. of blade passages}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

Using equation (5)

$F_t = 248.199$ N

Using equation (6)

$F_a = 3.82$ N

3.Evaluation of Gas Forces on the Second Stage Rotor Blades

The gas forces and power developed in second stage rotor blades were evaluated using the same procedure and similar equations that were used for first stage rotor blades

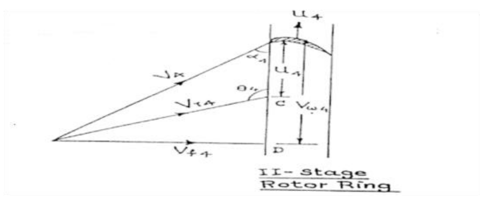

Fig. 3.1 Inlet Velocity Triangles of II-Stage

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ROTOR BLADES
We get,
Tangential force F_t = 244.49 N
Axial force F_a = 0.944 N

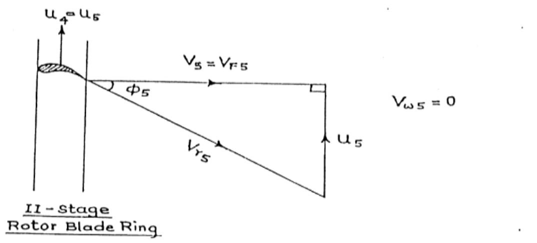

Fig. 3.2 Exit Velocity of Stage -II Rotor Blades

CETRIFUGAL FORCES EXPERIENCED BY THE ROTOR BLADES

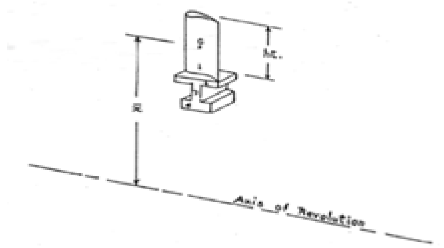

Fig. 3.3 3-D view of Rotor Blades showing position of centroid 'G'

4. CATIA DESIGN

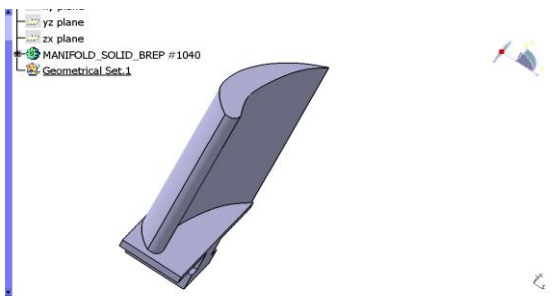

Fig.4.1 MODELING OF TURBINE BLADE

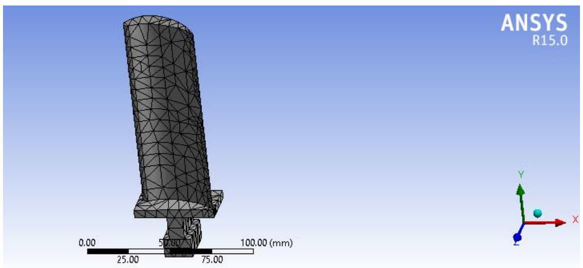

Fig.4.2 MESHING OF BLADE

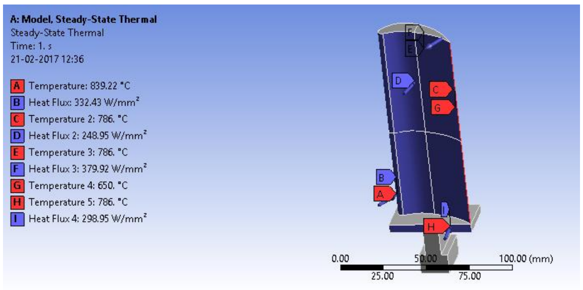

Fig.4.3 Steady State Thermal Analysis

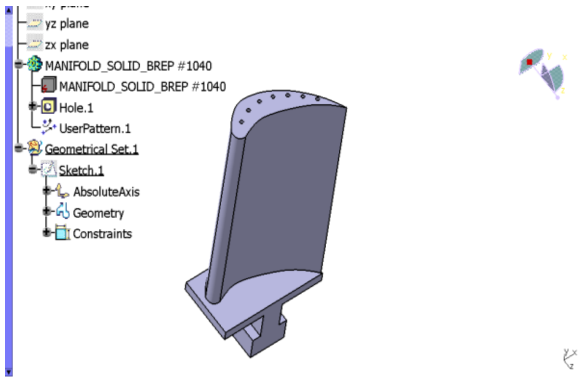

Fig.4.4 Optimization of blade by using Catia

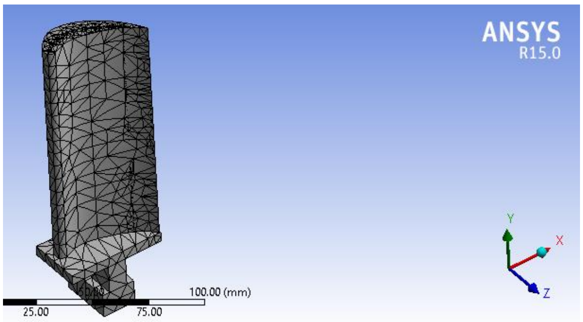

Fig.4.5 Meshing of Optimized Blade

5. COMPARISION OF BLADES

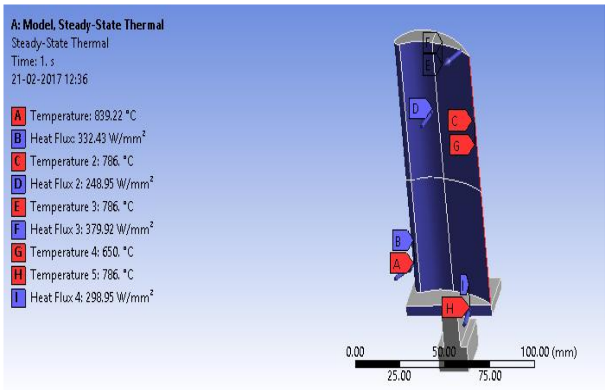

Fig.5.1 Geometric view of blade without hole

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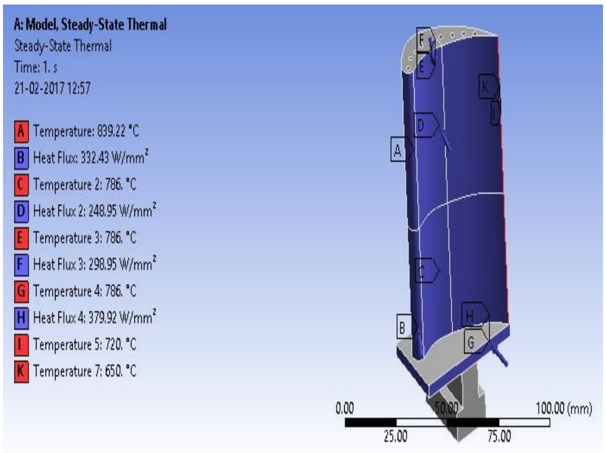

Fig.5.2 Geometric view of blade with 2mm hole

6. COMPARISON OF EQUIVALENT STRESSES

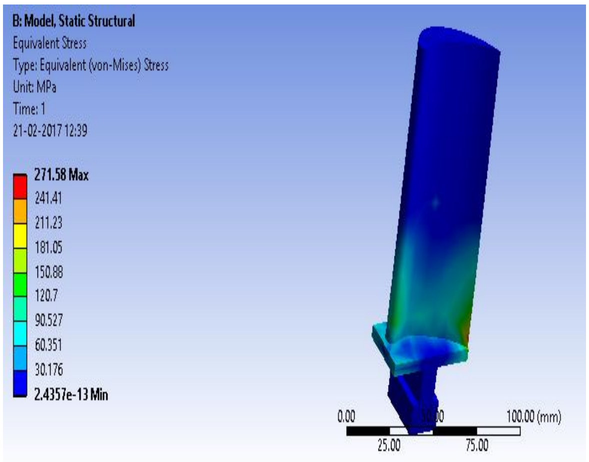

Fig.6.1 Blade without hole

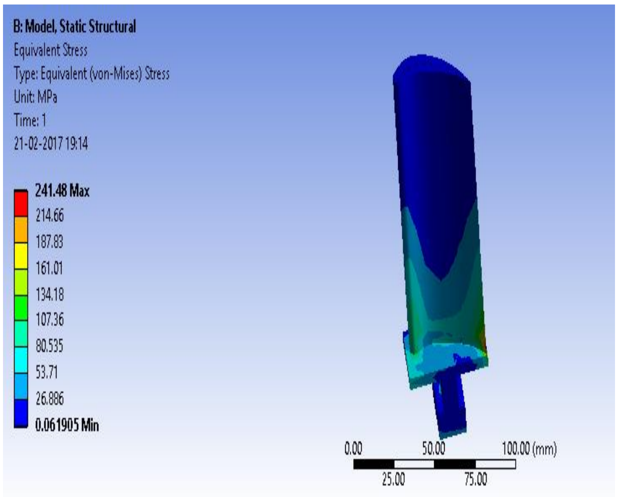

Fig.6.2 Blade with 2mm hole

Case 2: COMPARISON OF FATIGUE LIFE

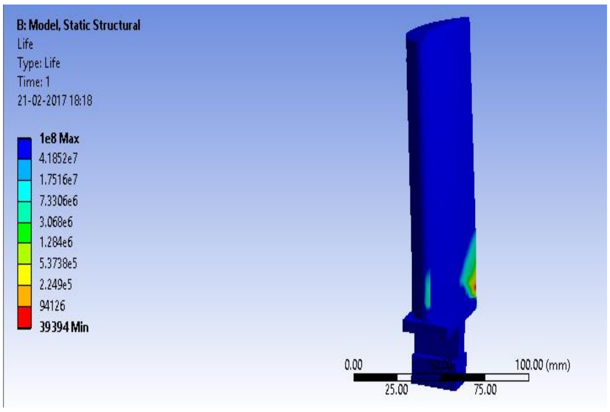

Fig.6.3 Blade without hole

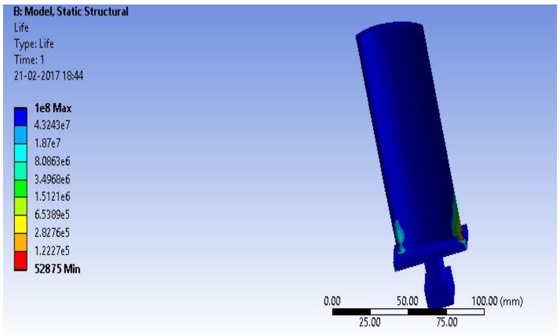

Fig.6.4 Blade with 2mm hole

Case 2: COMPARISON OF DAMAGE

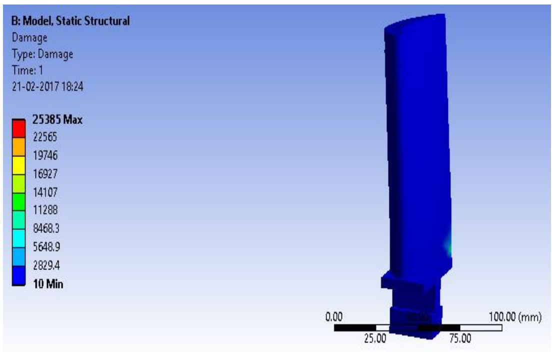

Fig.6.5 Blade without hole

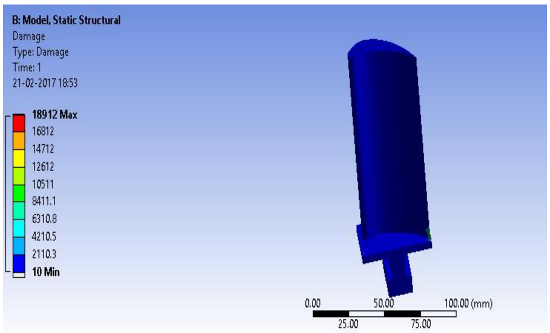

Fig.6.6 Blade with 2mm hole

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Case 3:COMPARISON OF FATIGUE SENSITIVITY

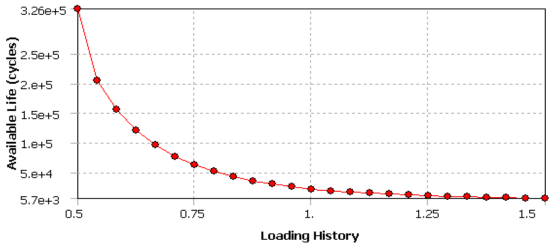


Fig.6.1 Blade without hole

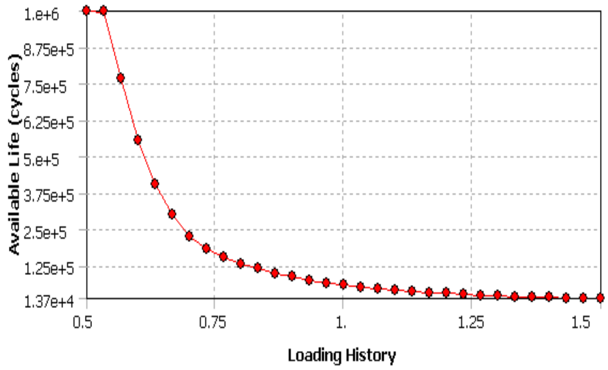


Fig.6.2 Blade with 2mm hole

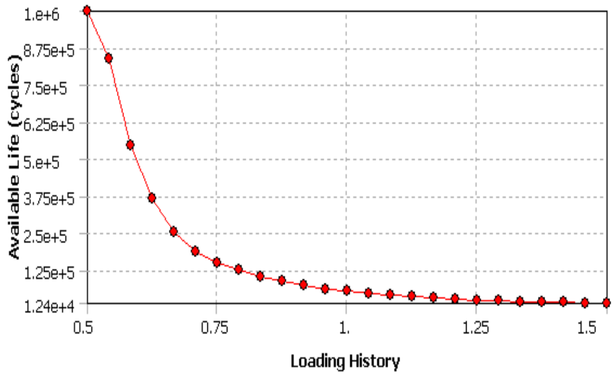


Fig.6.3 Blade with 3mm hole

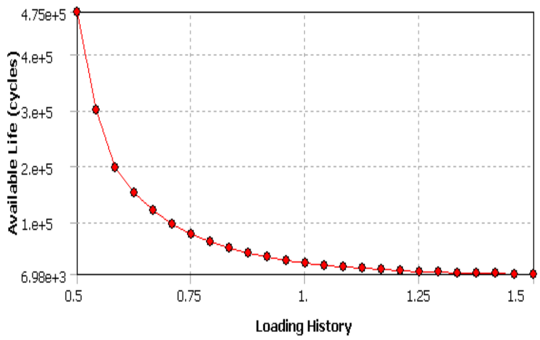
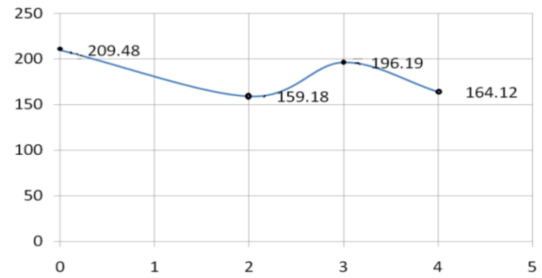


Fig.6.4 Blade with 4mm hole

7. RESULTS AND DISCUSSION

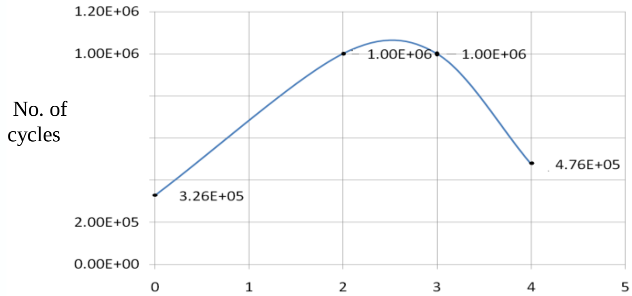
Table.7.1 : Results

Blade Specifications	Equivalent stress	Fatigue Life	Fatigue Damage	Fatigue sensitivity
Blade without hole	209.48N/mm^2	6.531E5	46232	3.26E5
Blade with 2mm hole	159.18N/mm^2	7.301E5	16884	1E6
Blade with 3mm hole	196.19N/mm^2	7.214E5	18888	1E6
Blade with 4mm hole	164.012N/mm^2	6.708E5	36354	4.76E5



Size of the holes (mm)

Fig.7.1 Graphical Representation for Stresses



Size of the holes (mm)

Fig.7.2 Graphical Representation for Fatigue Sensitivity

CONCLUSIONS

The finite element analysis of gas turbine rotor blade is carried out using 20 nodes brick element. The static and thermal analysis is carried out. The temperature has a significant effect on the overall stresses in the turbine blades. The following conclusions are made from the above analysis:

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1. Stress obtained for the turbine blade without holes is 209.48N/mm^2
2. When the size of the hole is 2mm stress obtained is 159.18N/mm^2 therefore stresses are reduced by 50N/mm^2
3. For blade with 3mm hole stress increased by 37N/mm^2 whereas for the blade with 4mm holes stress results were quite different, it decreased by 35N/mm^2
4. The no. of cycles for blade without hole is 3.26 x 10^5, for blade with 2mm and 3mm hole no. of cycles remains constant i.e. 10^6 whereas for 4mm blade its 4.76 x 10^5
5. Stress are reduced and fatigue life increased for blades with holes
6. On the whole it is noticed that the equivalent stresses are reduced up to 23% for the blades with holes as compared to blade without holes

Thus blade with 2mm hole is better for suing because the stress obtained is less and the number of cycles increased when compared to blades with 2, 3and 4mm holes.

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