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Analysis of First Stage Gas Turbine Blade for Evaluating Damping Energy and Failure Mode

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Abstract

Objectives: The turbine undergoes high temperatures, high stresses, and a potentially high vibration environment while in service. These vibrational stresses lead to fatigue failures. The percentage of failures gets reduced by dissipating vibratory energy. An alternate method that employed for reducing the high level of stresses within permissible limits is incorporation of damping. The objective of the paper is to evaluate total damping energy of real case of first stage gas turbine blade collected from the site. **Methods/Analysis:** The total damping energy of turbine blade is evaluated through Lazan's law. The CAD model of turbine blade is generated by using 3D scanner. The 3D data set of the scanned turbine blade is converted to a solid model. The modal and fatigue analysis of CAD model of turbine blade has been performed using ANSYS® software. The mass, volume, yield stress, resonant frequencies and total deformation at resonant frequencies are obtained through modal analysis. Equivalent alternating stresses is obtained from fatigue analysis. **Findings:** Equivalent alternating stress is used to calculate fatigue stress, which is further used with yield stress and volume to evaluate total damping energy of turbine blade. The total damping energy of the blade is used to evaluate loss factor and further equivalent damping co-efficient of turbine blade. From modal analysis results, it is found that there are stress concentration areas on leading and trailing edge of turbine blade at sixth resonant frequency. This shows that the turbine blade would be most susceptible to fatigue fracture at sixth resonant frequency. The available life cycle of blade is obtained from fatigue analysis. **Application:** The total damping energy of the blade is evaluated, which further used to calculate equivalent damping co-efficient of turbine blade. The failure mode and total available life of the turbine blade is predicted through computational analysis of turbine blade. This method is used as a necessary tool for the structural health monitoring of turbine blade in thermal power plants.

Keywords: Fatigue Analysis, First Stage Gas Turbine Blade, Lazan's Law, Resonant Frequency, Total Damping Energy

1. Introduction

The most critical and important component of a turbine is turbine blades. Turbine blades undergo reversed loading and hence mostly failed in fatigue mode[1]. When the natural frequency of structure coincides with any of the harmonic frequency then resonance occurs. In service, the turbines face high temperature, high stresses, and a potentially high vibration environment. These vibrational stresses lead to fatigue failures. The percentage of failures gets reduced by dissipating vibratory energy. The method employed for reducing the level of resonant stresses within permissible limits is damping[2,3].

In this research, the blade under consideration is a first stage 30 MW gas turbine blade as shown in Figure 1. The blade is made of nickel based super alloy IN738LC,

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having operating temperature 1100°C, rotates at 5135 rpm and pressure of gas stream on leading edge of turbine blade is 10 Kgf/cm^2.

Modal and fatigue analysis of CAD model of turbine blade are carried out using *ANSYS*® software[4]. The mass, volume, yield stress, resonant frequencies and total deformation at resonant frequencies are obtained through modal analysis. Equivalent alternating stresses is obtained from fatigue analysis. Equivalent alternating stress is used to calculate fatigue stress, which further used with yield stress and volume to evaluate total damping energy of turbine blade through Lazan's law[5]. The total damping energy of the blade is evaluated, which further used to calculate equivalent damping co-efficient of turbine blade[6]. The failure mode of the turbine blade is predicted through modal analysis of turbine blade. The available life cycle of blade is obtained from fatigue analysis [8,10].

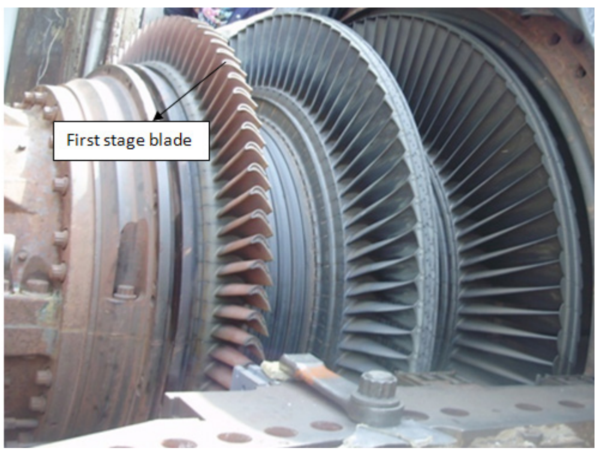

Figure 1. General view of Gas turbine blade.

2. Methodology

2.1 Fatigue Analysis of Turbine Blade

In this paper, modal and fatigue analysis of turbine blade are carried out using *ANSYS*® software[10]. The computational design model of turbine blade is generated by using 3D scanner. The 3D data set of the scanned turbine blade is converted to a solid model. The finite element model has 136468 nodes and 84361 elements as shown in Figure 2(a) and 2(b). Triangular surface mesh is made followed by volumetric mesh and it is used by auto meshing feature. The material properties and dimensions of turbine blade are presented in Table1 and Table 2 respectively.

The turbine blade fir-tree region is constrained by fixed support is given to fixed boundary conditions on its outer surfaces. The blade rotates at 5135rpm and pressure of gas stream impinge on leading edge of turbine blade is 10 Kgf/cm^2 as shown in Figure. 3.

For fatigue analysis, the turbine blade is subjected to constant amplitude loading as shown in Figure 4. The stress life is calculated based on Goodman mean stress theory as shown in Figure 5.

The equivalent alternating stress is obtained from fatigue analysis, which further used for evaluating fatigue stress of turbine blade using Eq. 1. The collected value of equivalent alternating stress, (σ_a) is *1.17211e^7 N/m^2* and yield stress, (σ) is *551 Mpa*.

The equivalent alternating stress is given by the relation[10,11].

$$\sigma_a = \sigma_e\left(1 - \frac{\sigma_m}{\sigma_u}\right) \quad (1)$$

Where σ_a equivalent alternating is stress, σ_e is the fatigue stress, σ_m is the mean stress and σ_u is the ultimate stress. The mean stress in terms of ultimate stress is given as

$$\sigma_m = 0.9\sigma_u \quad \text{For bending[11]} \quad (2)$$

By using Eq. 1 and 2, the evaluated value of fatigue stress, (σ_e) is *1.17211e^8 N/m^2*.

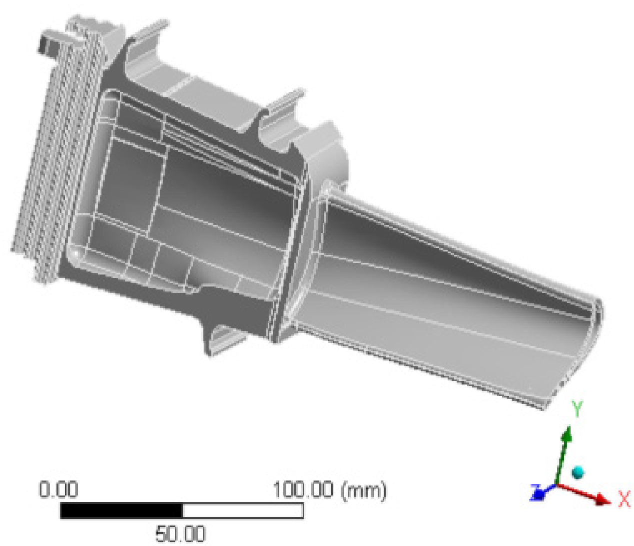

Figure 2(a). 3-D Model of turbine blade.

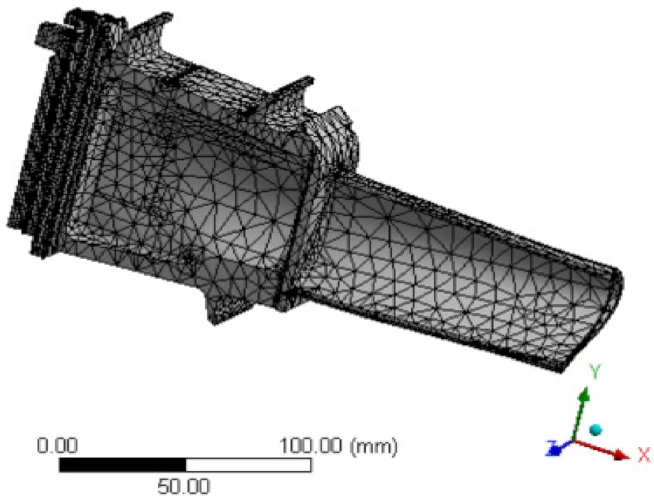

Figure 2(b). Mesh model of turbine blade.

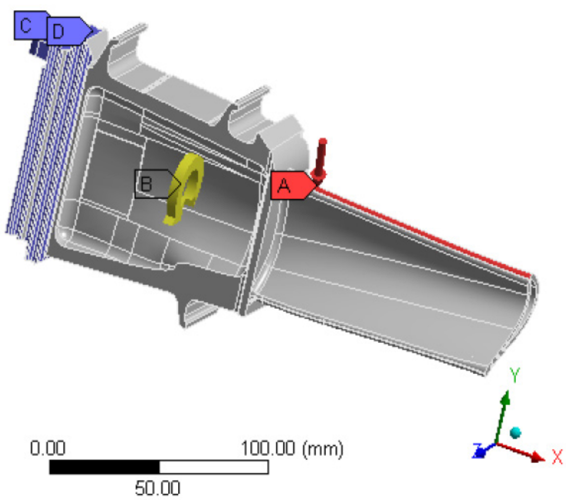

Figure 3. Loading condition of turbine blade for FEM analysis.

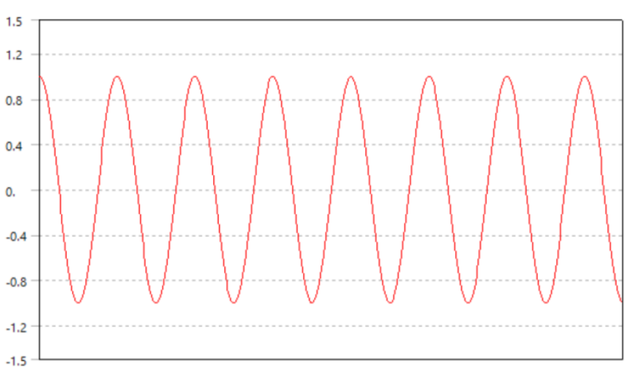

Figure 4. Constant amplitude loading fully reversed.

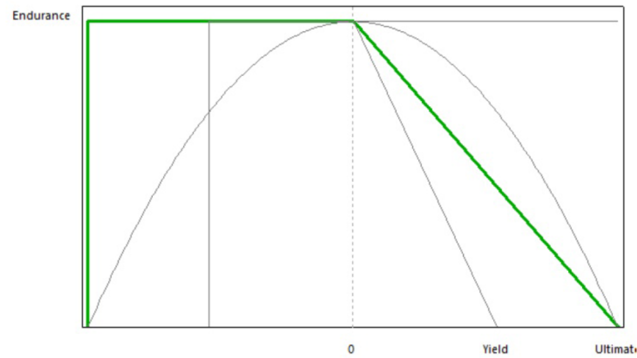

Figure 5. Goodman mean stress theory.

Table 1. Material properties of turbine blade 9

Material properties	Turbine blade
Young's Modulus, (N/m2)	1.3996x1011
Poisson's Ratio	0.3
Density (Kg/m3)	8110
Bulk Modulus, (N/m2)	1.1664x1011
Shear Modulus, (N/m2)	5.3832x1010
Tensile Yield Strength, (N/m2)	3.4474x108
Tensile Ultimate Strength, (N/m2)	4.5505x108

Table 2. Dimensions of turbine blade

Dimensions	Turbine blade (size of bounding box)
Length, L,(m)	0.259
Breadth, b,(m)	0.11
Depth, h,(m)	0.054

3. Evaluation of Damping Energy through Lazan's Law

The total damping energy of turbine blade is calculated through Lazan's law[5]. Harris and Crede[6] have expressed the material damping property in relation with yield stress and fatigue stress as

$$D = J\left(\frac{\sigma}{\sigma_e}\right)^n \tag{3}$$

Where D is specific damping energy $kNm/m^3/cycle$, J is a constant of proportionality, n is a damping exponent, σ is yield stress of material, σ_e is fatigue strength of material. The value of damping exponent (n) varies from *2, 2.1,* *2.2, 2.3* and *2.4*. The value of constant of proportionality (*J*) taken as *16*.

The total damping energy dissipated within volume (*v*) is described by Lazan's law as

$$D^0 = \int_0^V D dv \tag{5}$$

The specific damping energy at damping exponent (*n*) varies from *2, 2.1, 2.2, 2.3* and *2.4* are evaluated as 353.578, 412.766, 481.862, 562.524 and 656.688 *kNm/m³/cycle* respectively by using Eq. 3. The total damping energy of turbine blade is evaluated by multiply specific damping energy with volume of blade as given in Eq. 4.

4. Modal Analysis of Turbine Blade

The resonant frequencies and total deformation at these frequencies are obtained through modal analysis of CAD model of turbine blade[12-15]. First six resonant frequencies collected through modal analysis are presented in Table 3. The total deformations at first six resonant frequencies are shown in Figure 6.

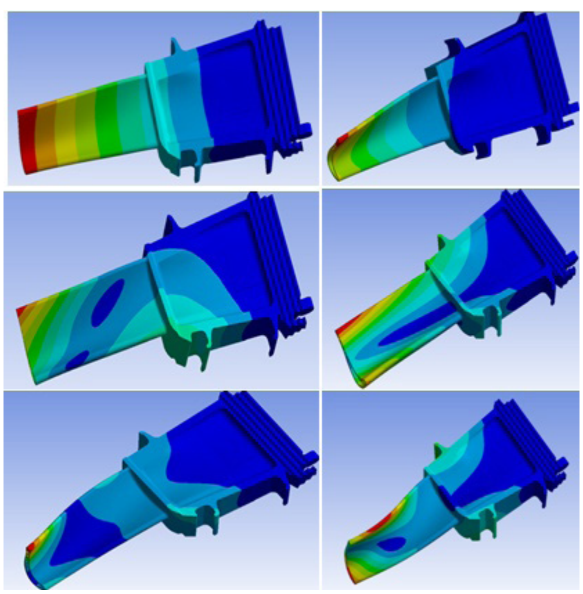

Figure 6. Total deformations at six resonant frequencies.

5. Results

The collected value obtained from computational analysis for volume of blade, (*v*) is 3.1224e-004 m^3.

5.1 Total Damping Energy of Turbine Blade

The total damping energy at damping exponent (*n*) varies from *2, 2.1, 2.2, 2.3* and *2.4* are evaluated by multiply specific damping energy with volume of blade as presented in Table 4.

Table 3. First six resonant frequencies of blade

Modal Parameters	Natural Frequency	Nature
Mode	495.8 Hz	Bending
Mode	1306.9 Hz	Bending
Mode	1961.6 Hz	Twisting
Mode	2162 Hz	Twisting
Mode	3089.6 Hz	Bending+ Twisting
Mode	3764.9 Hz	Bending + Twisting

5.2 Failure of Blade

The areas of stress concentration are clearly shown on leading and trailing edge of the turbine blade in Figure 7. These stress concentration areas found at sixth resonant frequency. This shows that the turbine blade would be most susceptible to fatigue fracture at sixth resonant frequency.

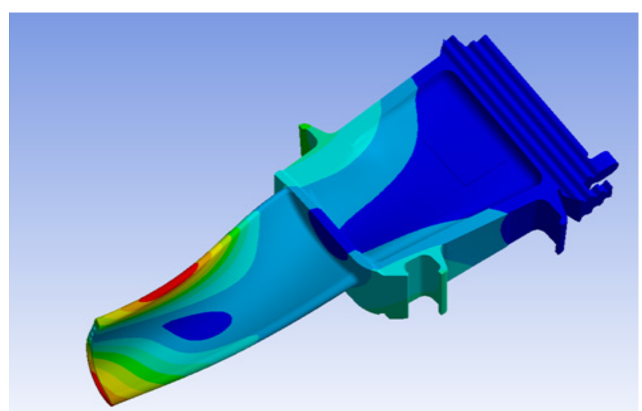

Figure 7. Total deformation at sixth resonant frequency.

Table 4. Total damping energy of turbine blade

Damping Exponent, n	2	2.1	2.2	2.3	2.4
Specific Damping Energy, kNm/m3/cycle , D	353.578	412.766	481.862	565.524	656.688
Total Damping Energy, kNm/cycle, D0	0.11040	0.12888	0.15045	0.17564	0.20504

6. Life of Blade

The available life (cycles) of turbine blade is obtained from fatigue analysis. The available life of turbine blade is 31694 cycles as shown in Figure 8.

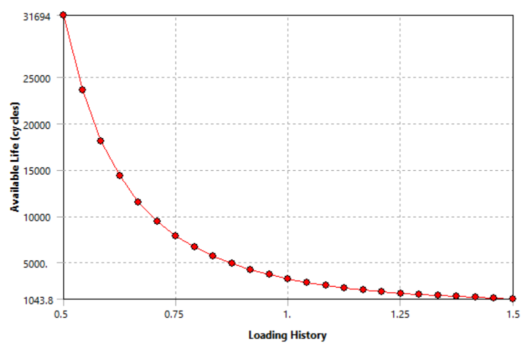

Figure 8. Available life (cycles) of turbine blade.

7. Discussions

The total damping energy of the blade is evaluated, which further used to calculate equivalent damping co-efficient of turbine blade. From modal analysis results, it is found that there are stress concentration areas on leading and trailing edge of turbine blade at sixth resonant frequency. This shows that the turbine blade would be most susceptible to fatigue fracture at sixth resonant frequency. The available life of turbine blade is 31694 cycles obtained through fatigue analysis of turbine blade. This method is used as a necessary tool for the structural health monitoring of turbine blade in thermal power plants.

8. Conclusions

There are areas of stress concentration found on leading and trailing edge of the turbine blade at sixth resonant frequency. This shows that the turbine blade would be most susceptible to fatigue fracture at sixth resonant frequency. The total available life of the turbine blade is 31694 cycles obtained through fatigue analysis. The method is used as a necessary tool for the structural health monitoring of turbine blade in thermal power plants.

9. Nomenclature

σ_a = equivalent alternating stress N/m^2
σ = Yield stress N/m^2
σ_e = Fatigue stress N/m^2
σ_m =Mean stress N/m^2
σ_u = Ultimate stress N/m^2
n= Damping exponent
D= Specific damping energy kNm/m^3/cycle
D^0= Total damping energy kNm/cycle
J= constant of proportionality
m = Mass of turbine blade, Kg
K_{eq} = spring stiffness for blade
ω_{n_1} = First resonant frequency

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