Stress Distribution on Turbine Blade along with Root at Several Operating Speeds

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Abstract: Steam turbine blades are a very important element in the thermal power generation station and these blades are exposed to higher temperatures and pressure in the thermodynamic power cycle. The designing of such turbine blades and roots is one of the crucial components; most of the failures occurring in the power plant are at blade sections/roots of the sections. The stress distribution is a very important parameter for designing such blades. In this research paper, the turbine blade along with an inverted fire tree root is considered for stress analysis. Blade and roots are divided into no of section based up the load acting on it. The stress distribution analysis can be done by different rpm for the X20Cr13 material. The original operating speed as 6728 rpm, the analysis is extended up to 30\% of increased speed along with original conditions. The failure of the turbine blade occurs at 15 \% of increased speed from the original speed condition, in section 1 for QT 650 (material 1), and for QT 750 (material 2) it is increased up to 25\% for the safest operating conditions.

1. Introduction:

The main feature of steam turbines is to convert pressure energy of steam form into work form, the areas where mainly to concentrate in steam turbines are pressure drop and centrifugal stresses developed due to rotation of the rotor. The various types of turbines are reaction and impulse turbines. The moving and stationary blades are present in reaction turbines and in the case of impulse turbines it has no pressure drop or small across its moving blades. In a reaction turbines, moving blades are designed in such a way that they use the steam energy passed through the stationary blades which act as nozzles, a reaction force produced by the pressure drop across them supplements the steam jet force of the stationary blades this causes rotation. Pressure drop is a very significant term in steam turbines which leads to the conversion of kinetic energy into various forces like impulse and reaction forces due to which rotation of the rotor or shaft takes place. The various types of blades used in steam turbines are HP and LP blades, Previously HP blades used are straight in structure now a day’s latest
one’s are leaned, bowed and Shrouds blades are used to reduce losses and leakages. There are some problems in the case of HP and LP blades which needs to solve immediately in the case of HP blades design it is all about stresses developed due to rotations (tensile, compressive, and shear stress) and to overcome this problem proper grouping of blades is to be done and necessary yield strength is essential then static and Dynamic stresses can be reduced. In the case of IP blades, greater radial variation in the flow are points of consideration. Centrifugal stresses are important source of stresses on blades and they can show most effect during root attachment. Subramanyam Pavuluri et.al [1] gave the information related to an issue of efficiency of a steam turbine and discussed the latest overhaul design of high-pressure steam turbine blades. The main focus was on blade profile, materials to be used and the important factors related to turbine blade failure and turbine failure itself. This paper also has information about the latest technologies that improve the overall efficiency of the turbine, the prevention of turbine failure mainly due to erosion and cracking of blades, and the effectiveness of turbine blades. Dr. A. Siva Kumar et.al [2] as discussed about the Experimental investigation on design of high-pressure steam turbine blade addresses the issue of steam turbine efficiency. The main focus was on aerofoil profile for high-pressure turbine blades, Effectiveness in turbine blades and information about thermal conditions of blades. Tulsidas. D et.al [3] Concentration was mainly on the geometry of root blades and design modification of the T-root blade. Finite Element Analysis is used to calculate stresses and Stress Concentration charts are used in blade root. Dr. Murari P.[4] PE in their paper given information about the design, specification to be used, and an overview of blade design and techniques. The paper also gives information about modal, fatigue, creep analysis, commonly used materials, and manufacturing processes. Hydro-Dynamic Performance Of Francis Runner [5]. This Analysis describes a methodology that predicts the performance and behaviour of the turbine runner in terms of the absolute pressure variation from inlet to outlet of the runner.

2. Nomenclatures:

| Symbol | Description |
|--------|-------------|
| QT     | Quenching and Tempering |
| m      | Mass        |
| ŵ       | Specific weight |
| V       | Volume      |
| g       | Acceleration Due to Gravity |
| FOS    | Factor of Safety |
| σall   | Allowable Stress |
| σact   | Actual Stress |
| C.F    | Centrifugal Force |
| ω      | Angular Velocity |
| N      | Speed R,P,M. |
| σcf    | Centrifugal (or Tensile) Stress |
| C.F    | Centrifugal (or Tensile) Force. |
| σc     | Compressive Stress |
| σs     | Shear Stress |

3. Design Methodology

The turbine blade and roots are divided into different sections. The blade is divided into 2 sections and the root is in six sections. In this paper, the stress distribution is a plot at each section for different speeds. The material 1 composition yield stress and material 2 yield stresses are considered to plot the FOS at each section of the blade/root. The blade modelled in AutoCAD and measures the dimensions as well as areas for every section, shown in below table- 3.1.
Figure 4.1. Geometrical model of the rotor blade along with an inverted fire tree.

| Sections | Area (A) mm² |
|----------|--------------|
| 1        | 166.0069     |
| 2        | 958.60       |
| 3        | 207.41       |
| 4        | 395.33       |
| 5        | 69.61        |

Table 3.1. Geometrical values

The allowable stress is considered for the material composition X20Cr13 as per the given table. For sample material-1, the heat-treated condition +QT650, the yield point, Rp0.2 is taken as 556 MPa and for sample material – 2 heat-treated condition +QT700, the yield point, Rp0.2 is taken as 667 MPa. The allowable tensile stress, allowable compressive stress and shear stress are calculated at each and every section.

4. Design Calculations

The centrifugal force is generated due to rotation of the rotor, it is very important to identify the different loads acting at different sections. Steam turbine blade section 1, section 2 and section 3(X-X) are subjected to tensile load and these sections tensile stresses are calculated at different operating speeds for material-1 and material-2. Section 3 (B-B) and section 5 (C-C) are subjected to compressed load and the sections 4 (E-E) & 5 (F-F) of inverted fire tree are experienced with the shear stress. The design calculations are used for developing a stress distribution throughout the given blade/root for different materials at different operating speeds.

Table 3.2. Allowable stresses

| Material -1 | Material -2 |
|-------------|-------------|
| Allowable Tensile stress kg/mm² | Allowable Compressive stress kg/mm² | Allowable Shear stress kg/mm² |
| Allowable Tensile stress kg/mm² | Allowable Compressive stress kg/mm² | Allowable Shear stress kg/mm² |
| 21.007 | 31.5108 | 15.7554 | 27.52 | 41.28 | 20.64 |

Mass (m) = Specific weight (Ŵ) x Volume (V)/Acceleration Due to Gravity (g)
Factor of Safety (FOS) = Allowable Stress/ Actual Stress
Volume (V) = Area x Length (mm³)
Centrifugal Force (C.F) = mro² (Kg)
Angular Velocity (ω) =2πN/60 (rad/sec)
Centrifugal (or Tensile) Stress (σcf) = mro²/A (Kg/mm²)
Compressive Stress (σc) = Compressive force/2 x Ac (Kg/mm²)
Shear Stress (\(\sigma_s\)) = Shear force / 2 x As (Kg/mm\(^2\))
FOS = Allowable Stress / Actual Stress

5. Results and Discussion

Figure 5.1. Original operating condition

Figure 5.1 shows the variation of stress distribution throughout the steam blade/root for the material -1, at an original operating condition (6728 rpm) the maximum tensile stress acting at section -1 (throughout the blade height) is 16.3766 kg/mm\(^2\), the maximum compressive stress is 20.6272 kg/mm\(^2\) at section 5 (C-C) and the maximum shear stress acting 9.46065 kg/mm\(^2\) at section 4 (E-E). Fig 5.2 shows the stress distribution throughout the steam blade/root for the material -1, at 10 % increase of the original operating speed (7400.8 rpm) The maximum tensile stress is acting at section -1 (throughout the blade height) is 19.8157 kg/mm\(^2\), The maximum compressive stress is 24.9589 kg/mm\(^2\) acting at section 5 (C-C) and the maximum shear stress is acting 11.4473 kg/mm\(^2\) at section 4 (E-E).

Figure 5.2. 10 % increase of the original operating condition

Figure 5.3. 12 % increase in the original operating condition

Figure 5.3 shows the stress distribution throughout the steam blade/root for the material -1, at 12 % increase of the original operating speed (7535.36 rpm) the maximum tensile stress is acting at section -1 (throughout the blade height) is 20.5428 kg/mm\(^2\), the maximum compressive stress is 25.8748 kg/mm\(^2\) acting at section 5 (C-C) and the maximum shear stress is acting 12.5117 kg/mm\(^2\) at section 4 (E-E). And at 15 % increase of the original operating speed (7737.2 rpm) the maximum tensile stress is acting at section -1 is 20.5428 kg/mm\(^2\), the maximum compressive stress is 25.8748 kg/mm\(^2\) acting...
at section 5 (C-C) and the maximum shear stress is acting 12.5117 kg/mm² at section 4 (E-E) shown in figure 5.4.

![Figure 5.9. FOS at Each Section for Material-1](image)

Figure 5.9 Shows the Factor of safety throughout the turbine blade and root for the given material 1, as the speed increases the forces acting on the blade increases. The turbine blade operated safely at original operating speed as well as 10% and 12% of increased in the speed of operating conditions, but if the speed reaches the 15% increased of original speed the tensile stress developed due to the rotation of the blades is more than the allowable tensile stress at that section and it will be failed at section -1. It is observed that for the material 1, the maximum safest operating speed is 7535.36 therefore the material 2 is consider for the analysis.

![Figure 5.5. 15% increase in the original operating condition](image)

![Figure 5.6. 21% increase in the original operating condition](image)

![Figure 5.6. 21% increase in the original operating condition](image)

![Figure 5.7. 25% increase in the original operating condition](image)

![Figure 5.8. 30% increase of the original operating condition](image)
Figure 5.1-5.4 shows the variation of stress distribution throughout the steam blade for the material -2. The maximum stresses are shown in table- 5.1

| S.No | Nature of Stress (kg/mm²) | 7737.2 R.P.M. | 8140.88 R.P.M. | 8410 R.P.M. | 8746.4 R.P.M. | Section of the Blade |
|------|---------------------------|---------------|----------------|-------------|---------------|---------------------|
| 1    | Maximum Tensile Stress    | 21.6580       | 23.9770        | 25.5884     | 27.6765       | In section - 1.     |
| 2    | Maximum Compressive Stress| 27.2795       | 30.2003        | 32.2300     | 34.8600       | At section 5 (C-C). |
| 3    | Maximum Shear Stress      | 12.5117       | 13.8513        | 14.7822     | 15.9885       | At section 4 (E-E). |

Figure 5.10 Shows the Factor of safety throughout the turbine blade sections and root sections for the given material 2, the turbine blade operated safely at original operating speed as well as up to 25% of the increase in the speed of operating conditions, but if the speed reaches the 30% increased of original speed the tensile stress developed due to the rotation of the blades is more than the allowable tensile stress at that section and it will be failed at section -1. From the above figures, it is observed that for the material2 the maximum safest operating speed is 8410 R.P.M.

6. Conclusion:

The speed of the rotor and the stress are directly proportional. Based on the structural analysis of the blade for the material -1, the FOS at 15% of the increased speed of blade at section 1 is less than one. Therefore for material -1, the safe working speed is up to 12% of the original operating speed. The structural analysis of the blade for the material -2 gives the FOS at 30% of increased speed, at the blade section 1 is less than one. Therefore for material -2 the safe working speed is up to 25% of the originally operating speed.
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