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Design and Finite Element Analysis (FEA) of Gas Turbine Rotor Disc

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Abstract: A Turbine wheel (rotor) plays a significant role in gas turbine. The rotor on which the blades are mounted transmitting this motion holds a key point to better efficiency of gas turbine.
Thus, more focus should be given to the design of the turbine rotor. Gas-turbine discs are normally operated at high temperatures.
The hot gases contact the blades and the rim of the turbine rotor and thus maintain the rim at high temperature. The temperature gradient at rim and central portion of rotor causes the sources for thermal stresses. The disc is expected to perform well despite all the stringent operating conditions. The Advancement in gas turbine materials has been always a major concern—higher their capability to with stand elevated temperature service, produce lower stresses, light weight and more the engine efficiency.
In this thesis, an attempt is made to determine the stresses like Thermal, Structural, Radial, and other stresses with different materials.
The analytical analysis is carried out by using ANSYS to determine the intensity of stresses. Later an attempt is also made to determine the theoretical fatigue life and creep of the rotor disc.

I. INTRODUCTION

Gas turbines have an important role in power generation and propulsion unit. Gas turbine technology is used in a variety of configurations for electric power generation.
Gas turbine is divided into three modules namely compressor, combustion & Turbine. In the gas turbine (GT), rotor consist of many components like rotor disc, blades, Intermediate shaft, front hollow shaft, rare hollow shaft, tie bolt, tie nut, support cone etc. A Turbine wheel/disc (rotor) plays a significant role in gas turbine.
The rotor on which the blades are mounted transmitting this motion holds a key point to better efficiency of gas turbine. Thus, more focus should be given to the design of the turbine rotor.
Gas-turbine discs are normally operated at high temperatures. The hot gases contact the blades and the rim of the turbine rotor and thus maintain the rim at high temperature.
The temperature gradient at rim and central portion of rotor causes the sources for thermal stresses. The disc is expected to perform well despite all the stringent operating conditions.
The Advancement in gas turbine materials has been always a major concern higher their capability to with stand elevated temperature service, produce lower stresses, light weight and more the engine efficiency.
In this thesis, the study has been carried out for compressor rotor disc subjected to higher temperature. Attempt are made to study the structural and thermal behaviours of the GT rotor disc.

A. Objectives

1) Evaluation of thermal behavior of the rotor disc rotating at high-speed operating condition and subjected to varying temperature.
2) Evaluation of structural behavior of the rotor disc when subjected to various load like blade pull, bolt pretension etc.
3) To perform the strength assessment.
4) To predict the fatigue life of the rotor disc
5) To perform the creep assessment of rotor disc.

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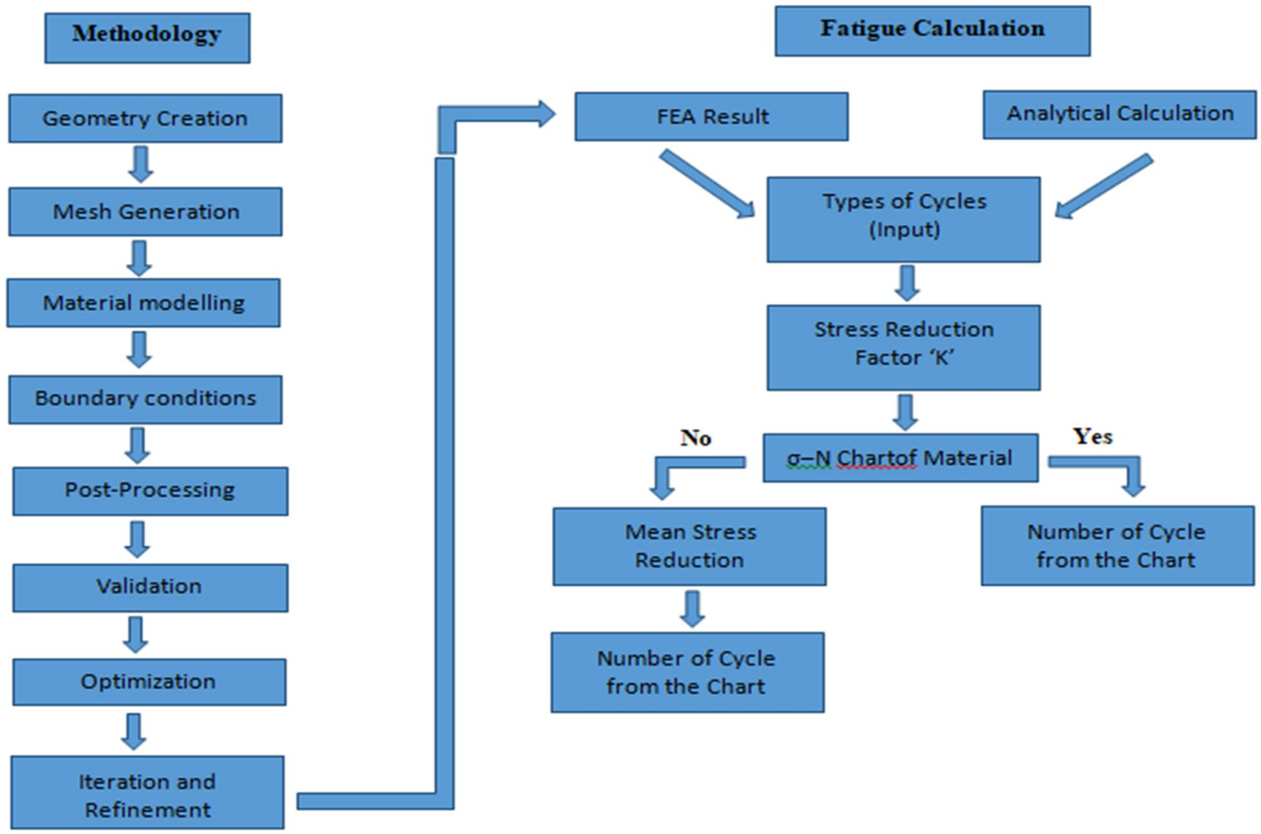

II. EXPERIMENTAL DESIGN AND PROCEDURE

Follow given methodology for this experiment.

1) *Geometry Creation:* Start by creating the 3D geometry of the rotor disc in NX-12.Generated 3 different rotor disc with different material. Disc rotors are equally divided into 4 part to assign 4 different zone temperature.Ensure that the geometry accurately represents the rotor disc, including features such as blade attachments, cooling channels, and any other relevant details.With the help of Ansys Design Modular convert our 3D model in to 2D axis symmetric section. To find the load and stress.

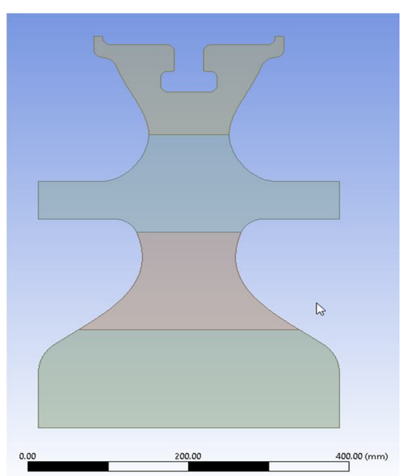

Figure 1 Rotor Disc 2D Cross Section

2) *Material Assignment:* Define the material properties for Alloy 718 and X12CrMoWVNbN10-1-1 in NX/ANSYS. This includes specifying the elastic properties, such as Young's modulus and Poisson's ratio, as well as other mechanical properties like yield strength, ultimate tensile strength, and thermal expansion coefficients.All mechanical properties field as per material selection table.
3) *Mesh Generation:* Generate a suitable mesh for the rotor disc geometry. The mesh should have enough resolution to accurately capture the stress and deformation behaviour of the component.In 2D cross section analysis we are taking two types of meshing. Face sizing mesh- in this meshing I select all the body cross section with 4 mm element size.Edge Sizing mesh- in this meshing, I choose the blade route position where the blade draw load will occur, and the element size in this meshing site is 0.5mm.Number of total nodes = 30057, Number of contact elements = 38, Number of solid elements = 9742, Number of total elements = 9780

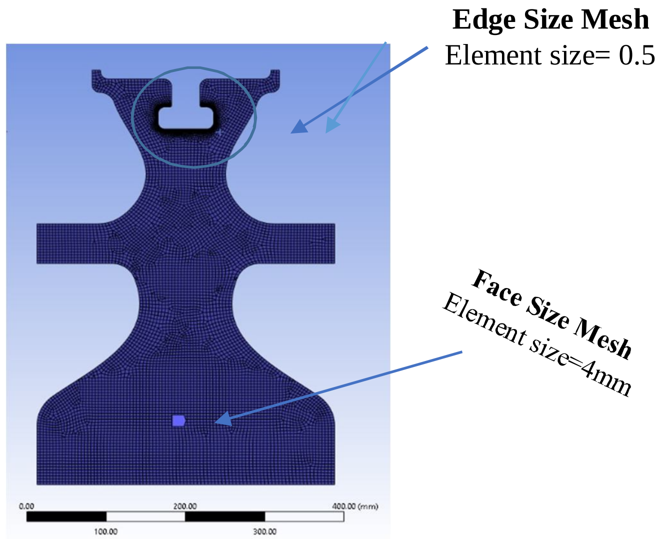

Figure 2 Meshing and Element Size

4) *Boundary Condition*
1) Step -1Apply a fixed constraint at one end where two Hirth Teeth are matched.Apply Pretension value - 177 MPa given to the opposite side of the Hirth Teeth.
2) Step -2 Apply a Rotation velocity- 3000 RPM. This will provide centrifugal force on the disc body. Applying Blade pull load- 506 MPa at the blade route location.Apply thermal load to the disc since these conditions have an influence on the disc during operation, which is why we divided our disc body into four halves to establish various temperature zones. At the top, where the blade is mounted, the temperature will be high; therefore, I calculated that it will be 300°C there and 200°C in the centre.

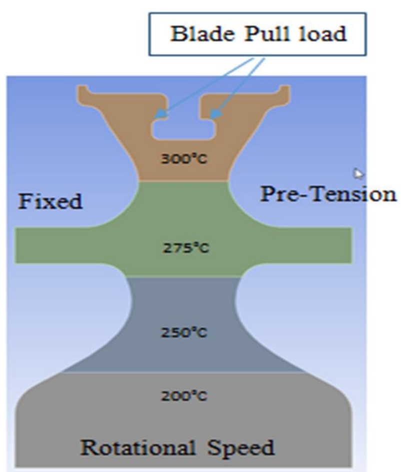

Figure 3 Boundary Condition

5) *Analysis Setup:* Specify the type of analysis to be performed, such as static analysis, transient analysis, or fatigue analysis, depending on the objective of the study.Define the analysis settings, such as time or load steps, convergence criteria, and solution controls. Consider any specific analysis requirements or considerations for Alloy 718, such as its high-temperature behaviour or time-dependent properties.

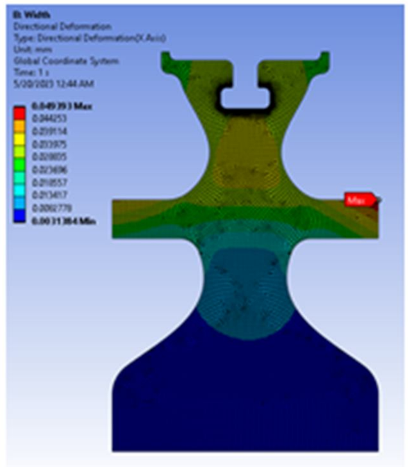

Figure 4First Stage Radial Deformation

6) *Solution and Post-Processing:* Solve the FEA model using ANSYS to obtain the analysis results, including stress distribution, deformation, and any other output variables of interest.Verify the convergence of the solution and ensure that the analysis results are within acceptable limits.Post-process the results to visualize and interpret the data, such as generating stress contour plots, displacement plots, or extracting specific values at critical locations.

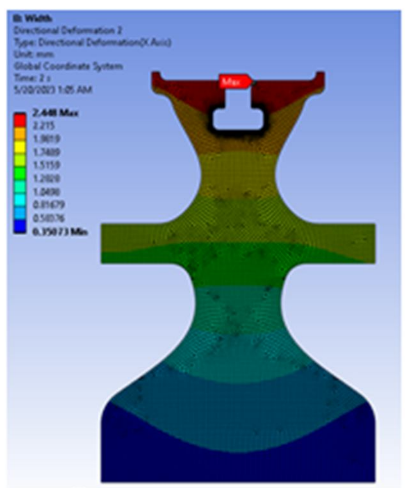

Figure 5Second Stage Radial Deformation

7) *Total Deformation with Step 1 and Step 2 loading condition.*

After Applying Boundary condition like fixed and Pre-tension Direction Deformation shown in Fig-2

Calculate the total deformation: The total deformation is calculated by measuring the magnitude of the nodal displacements. The total deformation at a specific node is the vector sum of the displacements in the x, y, and z directions. The Euclidean norm (magnitude) of the displacement vector can be calculated as:

Total Deformation= $\sqrt{\Delta \mathbf{x}^2 + \Delta \mathbf{y}^2 + \Delta \mathbf{z}^2}$

Where Δx, Δy, and Δz are the displacements in the x, y, and z directions, respectively.

Visualize the total deformation: Display the total deformation results as contour plots or vector plots. Contour plots show the deformation magnitudes as color-coded contours, allowing you to visualize areas of high and low deformations. Vector plots represent the deformation as arrows, indicating both the direction and magnitude of displacement at each node.

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Solve the FEA model: Solve the FEA model with the applied pre-tension and fixed constraints. The FEA software will calculate the displacements and deformations of the structure under these conditions.

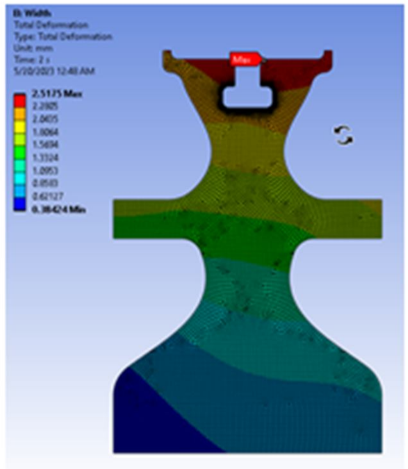

Figure 6Total Deformation

To calculate the von Mises stress, you need to consider the three principal stresses (σ1, σ2, and σ3) acting on a material point. These principal stresses are obtained through finite element analysis (FEA) or other stress analysis methods. The von Mises stress (σv) is then determined using the following equation:

$$\sigma v = \sqrt{[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]}$$

This equation essentially considers the differences between the principal stresses, resulting in a measure of the combined stress state that leads to failure. The von Mises stress provides a simplified representation of the stress state and is particularly useful when dealing with materials that exhibit similar strengths in tension, compression, and shear.

By comparing the calculated von Mises stress to the yield strength or ultimate tensile strength of the material, engineers can assess whether the material is at risk of failure. If the von Mises stress exceeds the material's strength, it indicates that the material is likely to undergo plastic deformation or failure.

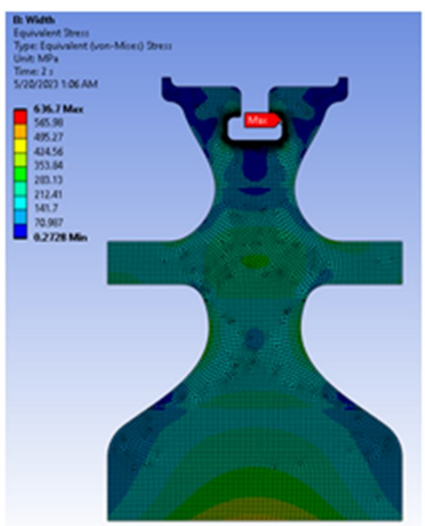

Figure 7Equivalent Stress (Von-Mises)

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III. FATIGUE AND FRACTURE CALCULATION & LIFE ASSESSMENT

A. For High-Cycle Fatigue Condition

$S'_f = \sigma'_f(2N_f)^b$ (Basquin Relation)

Where:

σ_a = Stress amplitude

σ'_f = Fatigue strenght coefficient

b = fatigue strength exponent

$2N_f$ = Reversals to failure

B. For Low-Cycle Fatigue Condition

$\varepsilon_p = \varepsilon'_f(2N_f)^c$ (Coffin-Manson Law)

Where:

ε_p = Independent Plastic Strain amplitude

ε'_f = fatigue ductility coefficient

c = fatigue ductility exponent

The parameters $\sigma'_f, \varepsilon'_f, b, and\ c$ are experimentally obtained material qualities using experimental data of $\sigma_a, \varepsilon_p, and\ N_f$

C. For Intermediate fatigue Conditions

Where:

ε_e = Elastic Strain

ε_p = Plastic Strain

$(\varepsilon_a - N)$ = Total Stain – Life

$$\varepsilon_a = (\varepsilon_e + \varepsilon_p) = \left(\frac{\sigma_a}{E} + \varepsilon_p\right) = \left(\left(\frac{\sigma'_f}{E}\right)(2N_f)^b + \varepsilon'_f(2N_f)^C\right)$$

D. Fatigue Calculation Formula

- $\boldsymbol{S_f = a * N * b}$
- $\boldsymbol{a} = \frac{(fS_{ut})^2}{S_e}$
- $\boldsymbol{b} = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right)$
- $\boldsymbol{N} = \left(\frac{\sigma_{rev}}{a}\right)^{\frac{1}{b}}$

$S_e = k_a * k_b * k_c * k_d * k_e * k_f * S'_e$

Where:

1. Surface Factor = k_a

   $k_a = a(S_{ut})^b$

   Where

   $a = 4.51$ (*Machine Surface*)

   $b = -0.265$

   $S_{ut} = 1100$ MPa (Inconel – 718)

   $k_a = 4.51(1100)^{-0.265} = 0.70977$
2. Size Factor = k_b

   $k_b = 1.24(d)^{-0.107}$ Where $2.79 \le d \le 51$mm

   $k_b = 1.51(d)^{-0.157}$ Where $51 \le d \le 254$mm

   $k_b = 1$ for axial loading

   $k_b = 1.51(180)^{-0.157} = 0.6682$
3. Load Factor = k_c

   $k_c = 1$ Bending

$k_c = 0.85$ Axial

$k_c = 0.59$ Torsion

4. Temprature Factor $= k_d$

$$k_d = \frac{S_T}{S_{RT}}$$

$$k_d = \left(\frac{S_T}{S_{RT}}\right)_{300°C} = 0.975$$

5. Reliability Factor $= k_e$

$$k_e = 1 - 0.08Z_a$$

$$k_e = 1 - 0.08 * 2.326 = 0.814$$

6. Miscellaneous – effect Factor $= k_f$

$$k_f = \frac{Max\ Stress\ in\ notched\ Specimen}{Stress\ in\ notch - free\ specimen} = 1.6460$$

$$q = \frac{k_f - 1}{k_t - 1}(Notch\ Sensitivity) = 0.2923$$

7. $S_e = k_a * k_b * k_c * k_d * k_e * k_f * S'_e$

$$S'_e = 0.5S_{ut}$$

$$S'_e = 0.5 * 1100 = 550\ MPa$$

$$S_e = k_a * k_b * k_c * k_d * k_e * k_f * S'_e$$

$$0.70977 * 0.6682 * 0.59 * 0.975 * 0.814 * 1.6460 * 1100 = 402.09\ MPa$$

8. $a = \frac{(fS_{ut})^2}{S_e}$

$$a = \frac{(0.79 * 1100)^2}{402.09} = 1878.09$$

9. $b = -\frac{1}{3}\log\left(\frac{fS_{ut}}{S_e}\right)$

$$\boldsymbol{b} = -\frac{1}{3}\log\left(\frac{0.79 * 1100}{402.09}\right) = -0.112$$

10. $\boldsymbol{N} = \left(\frac{\sigma_{rev}}{a}\right)^{\frac{1}{b}}$

$\boldsymbol{N} = \left(\frac{\sigma_{rev}}{a}\right)^{\frac{1}{b}}$ Here $\sigma_{rev} = Sf = amplitude\ Stress = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{595 - 141}{2} = 227$

$$\boldsymbol{N} = \left(\frac{227}{1878}\right)^{\frac{1}{-0.112}} = 156x10^6\ Cycle$$

$\boldsymbol{N'} = \left(\frac{321}{1878}\right)^{\frac{1}{-0.112}} = 7x10^6\ Cycle$ With corrective amplitude Stress

E. *Fracture Mechanics Analysis (Crack Propagation Life)*

$$\boldsymbol{N_p} = \frac{\boldsymbol{a_o}^{\left(-\frac{m}{2}+1\right)} - \boldsymbol{a_f}^{\left(-\frac{m}{2}+1\right)}}{\left(\frac{m}{2} - \mathbf{1}\right)\boldsymbol{cf^m}\left(\frac{a_o}{w}\right)(\Delta\boldsymbol{\sigma})^{\boldsymbol{m}}\boldsymbol{n}^{\frac{m}{2}}} \qquad \textbf{– Paris Law}$$

Calculated Data:

Max. Equivalent Stress, $\sigma_{max} = 595$ MPa

Min. Equivalent Stress, $\sigma_{min} = 141$ MPa

$\Delta_\sigma = 454$ MPa

Consideration the below data

$$f\left(\frac{a}{w}\right) = 1.12$$

$$a_o = 0.0031m$$

$$a_c = a_f = \frac{k_{IC}}{(\pi(1.12\sigma_{max})^2)}$$

$$a_c = a_f = \frac{165^2}{(\pi(1.12 * 595)^2)} = 0.1925m$$

For Ferrite-Pearlite,

$$\frac{da}{dN} = 6.8 * 10^{-12} * (\Delta k)^3$$

From the above equcation, $C = 6.8 * 10^{-12}$

$M = 3$

Now, substituting all the above value in the Paris Law Equation

$$\mathbf{N_p} = \frac{\mathbf{0.0031^{\left(-\frac{3}{2}+1\right)} - 0.1925^{\left(-\frac{3}{2}+1\right)}}}{\mathbf{\left(\frac{3}{2}-1\right)6.8 * 10^{-12}(1.12) * 3(454)^3\pi^{\frac{3}{2}}}}$$

$$\mathbf{N_p} = \frac{15.68}{2481.6 * 10^{-6}}$$

$$\mathbf{N_p} = \boldsymbol{6318\ Cycles}$$

IV. RESULT DISCUSSION

The results of our examination clearly indicate that for a critical component subjected to elevated temperatures, it is vital to undertake fatigue and fracture analyses to establish its safety under off-design conditions.

Table 11

Sr. No.	Description	Inconel 718	X19CrMoCoVB9-2-1	X12CrMoWVNbN10-1-1
1	Overview of the materials studied:	High Fatigue	High Temp	High Temp
2	Comparison of structural behaviour	Good At 22°C	Good At 425°C	Good up to 650°C
3	Strength and safety considerations	Creep +	Thermal Stability	High Temp Apply
4	Thermal behaviour	Good	Best	Better
5	Weight considerations	Near Same	Near Same	Near Same
6	Performance implications	Strength	High Temperature	
7	Availability	Easy	Moderate	Difficult
8	Cost	Cheaper	Costly	Moderate
9	Manufacturing	Moderate	Easy	Good
10	Machining	Moderate	Easy	Good
11	Deformation (mm)	0.144	0.2518	0.214
12	Stress (n/mm^2)	252.219	206.141	258.539
13	Temperature Dissipation °C/m	61.66	121.46	61.66
14	Strength (N/mm^2)	1375	1050	1100
15	Fatigue Life	$9x10^6\ Cycle$	$7x10^6\ Cycle$	$7.1x10^6\ Cycle$
16	Crack Propagation life	$8.5x10^3\ Cycle$	$6x10^3\ Cycle$	$6.2x10^3\ Cycle$

V. CONCLUSION

In conclusion, the finite element analysis (FEA) conducted on the gas turbine rotor disc has provided valuable insights into its structural integrity and performance. The FEA results, in conjunction with analytical studies, have demonstrated that the rotor disc is deemed safe under the specified boundary conditions and environmental factors. This analysis has helped ensure that the rotor disc can withstand the anticipated mechanical loads and thermal stresses during normal operation.

By incorporating comprehensive fatigue and fracture analyses into the assessment process, engineers can gain a deeper understanding of potential failure modes, identify areas of concern, and make informed design decisions to enhance the component's durability and longevity. These further analyses will provide a more comprehensive evaluation of the rotor disc's performance, ensuring its ability to withstand unexpected loading scenarios and maintain safe operation throughout its service life.

VI. FUTURE SCOPE

The future scope for FEA rotor disc thesis can encompass several areas of research and development. Here are a few potential directions:

1) Advanced material modeling.
2) Nonlinear analysis.
3) Fracture mechanics.
4) Multi-physics coupling.
5) Uncertainty quantification.
6) Optimization and design improvement.
7) Experimental validation.
8) Condition monitoring and predictive maintenance.
9) Computational efficiency and parallel computing.

These are just a few potential areas for future research in FEA rotor disc analysis. Depending on your specific interests and goals, you can further refine and explore these areas or identify new directions to contribute to the field.

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