LIFE ESTIMATION OF TURBINE BLISK FOR A GAS TURBINE ENGINE

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ABSTRACT

The present paper deals with fatigue analysis of a turbine blisk (integrated blade and disc) for a small gas turbine engine for a short life application. Due to high rotational speeds, the blisk is subjected to high centrifugal loads. This blisk experiences varying stress amplitudes due to mission profile of the vehicle, which includes cruise and maximum thrust. Studies show that the low cycle fatigue is the critical failure mechanism of turbine blisk, hence strain-life approach is followed. A nonlinear elastic-plastic finite element analysis is carried out in ANSYS Mechanical and the strain amplitudes are obtained. The results from static structural analysis are fed to ANSYS nCode Design Life tool for further fatigue life estimation. The damage accumulation is obtained using Palmgren-Miner’s rule and Smith-Watson-Topper’s mean stress correction method is used to obtain the strain-life curve.

Keywords: Turbine blisk, Low cycle fatigue, Strain-life approach, Mean stress correction, Palmgren-Miner’s rule.


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1. INTRODUCTION

Fatigue is a critical failure phenomenon as it is the reason to more than 90% of all service failures of machine components. Therefore, fatigue life estimation is extremely important in extremely loaded components’ design [1]. The turbo machines are the most critical parts of
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gas turbine engines. These components possess enormous kinetic energies that are at the peak during maximum thrust conditions. Such operations induce severe cyclic stresses in the turbo machines. Hence, the absence of life prediction will lead to low cycle fatigue (LCF) failures [2, 3]. The expected life of small gas engine is much less than large engines. Hence certain degree of plastic deformation is permissible in this class of engines. Strain-life approach (E-N) is the best suited life estimation method for the case under study. For such application the E-N approach is better than the S-N approach which basically ignores plasticity.

The blisk is allowed to function in elastic-plastic condition for a short life application. Therefore, a nonlinear static structural analysis is carried out to capture the stresses which in turn needs nonlinear stress-strain curve for the chosen material. The turbine blades are working at extremely high angular velocities. The mission profile of the vehicle causes varying angular velocities of the turbine. This in turn leads to varying centrifugal loads on the blisk, which causes fatigue. the life has to be predicted for the blisk using strain-life approach.

2. METHODOLOGY

A design flow chart of turbine blisk presented in figure.1 gives an outlook of the methodology followed in estimating fatigue life of the turbine blisk.

![Design Flow Chart of Turbine Blisk](image)

**Figure 1** Design Flow Chart of Turbine Blisk

2.1. Geometry and Loads

The geometry considered in this paper is a 43 bladed turbine blisk. As there are 43 blades, a CAD model of 1/43rd sector of the blisk containing one blade is created and is shown in figure.2.

The figure.3 shows load profile of this blisk during its operation. The load profile of turbine blisk considered in this study consists of start, cruise (43250rpm), maximum thrust (52765rpm) and shutdown. This turbine is assumed to be working at 800°C.
2.2. Non-Linear, Cyclic and Fatigue Properties

The turbine blisk is made of nickel based super alloy [5] Nimonic 105 material. The non-linear stress-strain curve (obtained using material properties provided by material supplier catalogue [10]) of the material for turbine operating temperature is fed into the FEM software. The non-linear stress strain curve fed into ANSYS for Nimonic 105 is shown in figure 4. Where, the abscissa represents the stress in Pascal and the ordinate represents the Plastic strain.
To carry out the life estimation using strain-life approach, four fatigue and two cyclic properties, are required. These properties are derived from monotonic properties of the chosen material [8, 9]. These properties are used to arrive at the E-N curve (Coffin-Manson curve) shown in figure 5 and cyclic stress-strain curve for the material. Where, the abscissa represents the strain amplitude in Log\(_{10}\) scale and the ordinate represents the reversals to failure (2N) in Log\(_{10}\) scale. The properties required to obtain this curve are listed in table 1.

**Table 1 Material Properties of Nimonic 105 at 800°C**

<table>
<thead>
<tr>
<th>Property</th>
<th>Magnitude</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s Modulus, E (MPa)</td>
<td>168 X 10(^3)</td>
</tr>
<tr>
<td>Yield Strength (MPa)</td>
<td>646</td>
</tr>
<tr>
<td>Ultimate Tensile Strength (MPa)</td>
<td>836</td>
</tr>
<tr>
<td>Fatigue Strength Coefficient, (\sigma'_f) (MPa)</td>
<td>1254</td>
</tr>
<tr>
<td>Fatigue Strength Exponent, b</td>
<td>-0.079</td>
</tr>
<tr>
<td>Fatigue Ductility Coefficient, (\varepsilon'_f)</td>
<td>0.478</td>
</tr>
<tr>
<td>Fatigue Ductility Exponent, c</td>
<td>-0.6</td>
</tr>
<tr>
<td>Cyclic Strain Hardening Coefficient, K' (MPa)</td>
<td>1382.865</td>
</tr>
<tr>
<td>Cyclic Strain Hardening Exponent, n'</td>
<td>0.132</td>
</tr>
</tbody>
</table>

Since, the material undergoes cyclic loading, a suitable material hardening phenomenon is chosen to analyze the turbine blisk. In this paper, Kinematic strain hardening model is chosen to account Bauschinger effect [4].

**2.3. Stress-Strain Analysis**

Static structural analysis for the sector model of turbine blisk is carried out in ANSYS Mechanical. The stress-strain response is captured for varying speeds according to the load profile of the turbine. The triaxial state of stresses is reduced using von Mises theory. These stresses and strains obtained are used further in life estimation. The critical stresses in the turbine blisk are found at the root of the blade due to stress concentration at this region.

**2.4. Mean Stress Correction**

The number of cycles that the material can withstand is obtained from the E-N curve. Mean stress correction using Smith-Watson-Topper’s approach is carried out as stress ratio \(\neq -1\) and Manson-Coffin curve cannot be used [6, 7]. The Smith-Watson-Topper’s fatigue curve expression is given by,
\[ \sigma_{\text{max}} \varepsilon_a = \frac{a r^2}{E} (2N_f)^{2b} + \sigma' f \varepsilon' f (2N_f)^{b+c} \]  \hspace{1cm} (1)

Where, \( \sigma_{\text{max}} \) is the maximum stress experienced by the structure, \( \varepsilon_a \) is the strain amplitude, \( 2N_f \) number of cycles.

2.5 Cumulative Damage
The relation between damage and cycles to failure is given by Palmgren-Miner. The Palmgren-Miner’s rule assumes that the total damage is given by the summation of the percentage of life consumed by each loading case [7, 8]. The Palmgren-Miner’s rule is given by the expression,

\[ D = \sum_{j=1}^{K} \frac{n_j}{N_j} \]  \hspace{1cm} (2)

As damage at the failure of the structure is 100%, i.e. \( D = 1 \). The expression becomes,

\[ 1 = \sum_{j=1}^{K} \frac{n_j}{N_j} \]  \hspace{1cm} (3)

Where, “D” is total damage, “n” is the number of cycles at specific stress level, “j” is the load case, “K” maximum number of load cases, “N” is the fatigue life in terms of number of load cycles.

2.6 Life Estimation
The stress and strain amplitudes obtained from ANSYS static structural analysis are used in ANSYS nCode DesignLife tool. Palmgren-Miner’s rule and Smith-Watson-Topper’s mean stress correction approach are chosen in nCode DesignLife to estimate the fatigue life of the turbine blisk. Simultaneously analytical approach is followed to verify the results obtained.

3. RESULTS
At maximum thrust condition the stresses are found to be maximum at the blade root (715.9MPa) and the total strain at the same location is 0.0164. The figure 6 shows the distribution of stresses in the turbine blisk, the stress concentration at the blade root. The numerically estimated fatigue life of the blade using these stress-strain values is 582 cycles. Figure 7 shows the life estimation carried out in nCode DesignLife. The number of cycles the blisk can withstand for stress developed is obtained analytically from the Smith-Watson-Topper’s fatigue curve. The analytically estimated fatigue life is 543 cycles. The damage accumulation for one load cycle is estimated to be 0.001842, using Palmgren-Miner’s rule.

![Figure 6 Distribution of Von Mises Stresses in The Turbine Blisk](http://www.iaeme.com/IJMET/index.asp)
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The table 2 shows comparison of obtained numerical and analytical results.

Table 2 Comparative Results of Turbine Blisk

<table>
<thead>
<tr>
<th>Approach</th>
<th>Fatigue Life</th>
</tr>
</thead>
<tbody>
<tr>
<td>Analytical</td>
<td>543</td>
</tr>
<tr>
<td>Numerical</td>
<td>582</td>
</tr>
</tbody>
</table>

4. CONCLUSION

In this study attempt is made to estimate the fatigue life of a turbine blisk of a small gas turbine engine. A nonlinear static structural analysis is carried out obtain true state of stress and strain in the blisk. The stresses are found to me maximum at the blade root. Hence, the estimation of fatigue life is done at this critical region as the crack initiates at this region. The maximum stress obtained is 715.9MPa which is above the yield strength of the material (646MPa), therefore Strain-life approach is used to estimate the life. Palmgren-Miner’s rule on cumulative damage and Smith-Watson-Topper’s mean stress correction approach are used to estimate the fatigue life. Both numerical and analytical methods are followed and the estimated lives are compared. The life obtained indicates that the blisk is safe to operate for its test cycles and flights. There is a scope for design optimization to vary the fatigue life according to and the weight of the blade according to the application to obtain still more efficient design.

REFERENCES