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Life Estimation of a Steam Turbine Blade Using Low Cycle Fatigue Analysis

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Abstract

This work is primarily focused in utilising a technique to assess low cycle fatigue life of steam turbine blade. Two approaches are employed here, first is based on the linear elastic finite element analysis. Fictive elastic results are recalculated using Neuber's rule. Second approach is based on elastic-plastic Finite Element analysis (FEA). Strain amplitude approach is followed through Universal slope method and Coffin-Manson equation to determine the number of start-up and shut down cycles

Proper blade design with conservative stress levels is important for reliability in rotor blade design. Finite Element Analysis based fatigue tools enable reliable fatigue life calculations to be done at the design stage of a development process and a proper design methodology is important to predict the catastrophic failure of turbine blade due to fatigue.

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Key words: Low cycle fatigue; Steam turbine blade; Fatigue life estimation; Universal slope method (USM)

1. Introduction

One of the main factors concerning mechanical integrity of steam turbines is the interface region between the blade and the rotor disc. Stresses generated in this region are mainly produced by the centrifugal force resulting from the blade. Thermal loads, gas bending and torsion resulted due to steam pressure in moving blades.

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The geometric properties of blade are designed to meet the specific operating. The turbine stages often use blades with internal cooling, while the very long blades in the LP stages of large steam turbines are pre twisted. The centrifugal force untwists the blades to the correct angular alignment with the steam flow. Often the high pressure turbine stage and the low pressure stage use double ‘T’-root, the short and medium impulse blades at low pressure stage use single ‘T’-root blades is commonly seen.

Steam turbine blade and disc have few critical regions for which life time cortication is necessary such as T-root blade disc attachment region. The associated loads with these regions are due to the centrifugal force in blade, steam bending in hoop direction. The rotor is subjected to hoop and radial stress.

The objective of the present work is to:

- Evaluate the mechanical integrity of bladed disc assembly using simulation technique
- Geometric and material nonlinearities considered for the simulation to build customized methodology for fatigue life estimation
- To bring in correlation between bilinear kinematic hardening and neuber's rule to estimate the total strain
- Low Cycle Fatigue evaluation using USM and Coffin Manson equation
- To conduct fatigue life evaluation using commercial package Ansys.

Nomenclature

LCF	Low cycle fatigue
HCF	High cycle fatigue
$\Delta\varepsilon_t$	Total Strain
$\Delta\varepsilon_e$	Elastic Strain
$\Delta\varepsilon_p$	Plastic Strain
σ	Maximum Stress
E	Modulus of elasticity
M	Coefficient for ductility
N_f	Number of Life Cycles
G	Coefficient for strength
σ_u	Ultimate tensile strength
D	Logarithmic ductility
σ_m	Mean stress
RA	Reduction Area
σ_{Von}	Von mises stress
API	American Petroleum Institute
LE	Leading edge
TE	Trailing edge

2. Low Cycle Fatigue

When the cyclic stresses are high, typically in the neighbourhood of the yield strength of the material, the applied load cycles to failure are small. The life is controlled by inelastic component of strain. The total strain has elastic and inelastic components, and the plots of these components versus cycles to failure on a log-log plot are straight lines.

The most widely quoted fatigue life model is often expressed by the ‘ Mason-Coffin’ equation relating life to plastic strain range through a power law. Manson and Halford proposed that the cyclic life depended on total strain range. This in turn consisted of elastic and plastic components, each of which was linear with cyclic life on log-log scales. Total strain range was thus asymptotic to plastic line in the LCF region. The plastic strain was much larger than elastic strain and asymptotic to the elastic line in the HCF range where, the elastic strain greatly exceeds plastic strain. P. Mestanek., et al (2008)

The total strain range is then made up of the elastic and plastic components given by,

$$\Delta\varepsilon_t = \Delta\varepsilon_e + \Delta\varepsilon_p \tag{1}$$

This may be written as,

$$\Delta\varepsilon_t = \frac{\Delta\sigma}{\mathrm{E}} + \Delta\varepsilon_p \tag{2}$$

$$\Delta\varepsilon_t = \mathrm{MN}_f + \frac{G}{\mathrm{E}}\left(\mathrm{N}_f\right) \tag{3}$$

The coefficient 'M' is primarily governed by ductility and coefficient 'G' by strength. Murari, P. Singh., et.al (2005) (2000).

2.1. *Universal Slope Method*

Low cycle fatigue tests are time consuming and expensive as compared to monotonic tensile tests. Halford and Manson proposed the method called Universal Slope after examining the properties of a large number of materials, Manson, S. S (1967). By utilizing monotonic tensile proper- ties, a strain versus cycles to failure curve can be estimated. This method does not give the most accurate results for all materials, but it provides a very good first estimate of the minimum number of cycles for crack initiation. The total strain versus number of cycles is obtained using mathematical equations and can attain approximate number of cycles without the experimental fatigue curves. This method utilises the approximate fatigue factors like surface, load, size, service factors and so on. To aid in the material testing for fatigue and initial comparison among materials, this method is been found to be very useful.

To obtain fatigue properties of material, it is time consuming and expensive. Monotonic test properties are used to estimate fatigue properties. A simple equation universalizing the life model exponents with an approximate relation of M and G to the material properties is proposed. Julie. A. Bannantine et.al (1990)

$$\Delta\varepsilon = \frac{3.5\sigma_u}{\mathrm{E}}\left(2\mathrm{N}_f\right)^{-0.12} + D^{0.6}\left(2\mathrm{N}_f\right)^{-0.6} \tag{4}$$

Marrow equation is given by,

$$\Delta\varepsilon = \frac{3.5\left(\sigma_u - \sigma_m\right)}{\mathrm{E}}\left(\mathrm{N}_f\right)^{-0.12} + D^{0.6}\left(\mathrm{N}_f\right)^{-0.6} \tag{5}$$

Where $D = \ln\dfrac{100}{(100 - \%RA)}$ (6)

Mean stress, $\sigma_m = \dfrac{\sigma_{von}}{2}$ (7)

2.2. *Theories of Failures for static loading*

Many theories have been propounded to estimate the maximum static load that may be applied to a component without causing failure. These theories use data obtained from uniaxial tests when in the real situation the stress system will be multiaxial. This way one can avoid experimental determination of an infinite number of stress combinations of stresses that may arise in the real situation. Some of them are listed below. Joseph Edward Shigley (1986).

- Maximum Normal Stress Theory (by Rankine)
- Maximum Normal Strain Theory (by Saint Venant)
- Maximum Shearing Stress Theory (by Guest)
- Mohr Theory and Internal-Friction Theory (by Coulomb and Mohr)
- Maximum Strain Energy Theory (by Beltrami, by Huber, by Haigh)
- Von Mises-Hencky Theory (by Hencky and by Von Mises)

In a uniaxial tension test, when the specimen starts to yield, the following quantities reach their limits simultaneously:

- The principal stress ($\sigma = P/A$) reaches the tensile elastic strength (elastic limit or yield point) of the material.
- The tensile strain ε reaches the value of strain ε_ε.
- The maximum shearing stress ($\tau = 0.5\ P/A$) reaches the shearing elastic limit or shearing yield stress τ_{yp} of the material, $\tau_{yp} = 0.5\ \sigma_{yp}$
- The total strain energy W absorbed by the material per unit volume reaches the value $W_e = 0.5\ \sigma^2\varepsilon/E$
- The strain energy of distortion W_d (energy accompanying change in shape) absorbed by the material per unit volume reaches a value $W_d = ((1+\mu)/3E)\ \sigma_e^2$.

In the case of a uniaxial tensile test each of the six quantities described above are reached simultaneously. When the state of stress is multiaxial the listed six quantities will not occur simultaneously. It becomes important to consider which one of the quantities should be chosen to limit the loads that can be applied to a member without causing inelastic strain.

2.3. *Neuber's Rule*

Typically light weight structures are dimensioned in a way that limited local yielding is allowed. Traditional non-linear FEA analysis simulating the local material plasticity are still very resource intensive. Yet, fatigue and life endurance simulations commonly need stress and strain results for various different load levels making analysis expensive. In order to reduce the number of non-linear simulation results, approximation techniques based on the Neuber's formula is used to estimate the plastic stress-strain state from linear analysis runs are utilized. Neuber H., et al (1961)

Stresses are much higher than yield stress (585 MPa) of the material under linear analysis at design speed. Hence, use of linear analysis is alarming. However, the traditional non-linear finite element analysis simulating the local material plasticity are still very resource intensive, yet fatigue and life endurance simulations commonly need stress and strain results for various load levels, making such an analysis expensive. In order to reduce the number of non-linear simulation results, approximation techniques based on the Neuber's formula will estimate the plastic stress–strain state from linear analysis runs were utilized. The general Neuber's procedure of extrapolating linear stresses into the plastic material region can thus is applied to arbitrary geometries. The following Fig 1. shows neuber's hyperbola and its use. The definition of stress and strain concentrations the final equation is written as:

$$\sigma\varepsilon = \text{Constant} \tag{8}$$

The above equation represents a hyperbola called Neuber's hyperbola. The intersection of this hyperbola and the stress-strain curve should provide the actual state of stress condition in the material.

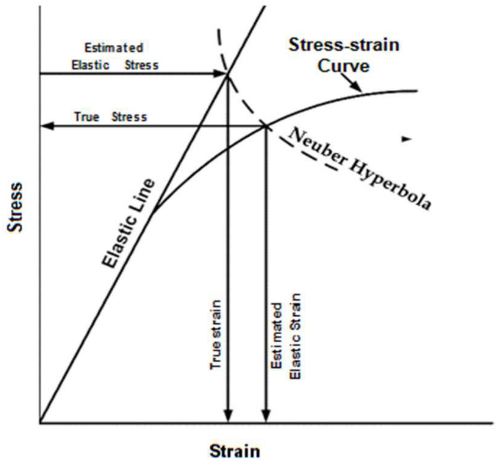


Fig. 1. Neuber's Hyperbola. Neuber H., et al (1961)

2.4 *Design specifications (operating conditions)*

Based on the explanations provided in above section, the present work design specifications are as follows:

- Inlet temperature: 450°C
- Mechanical loads: Static, Fatigue and Speed loading
- Thermal loads with temperature gradient of 450°C
- Rated Speed at 8650 rpm
- 121% over speed as per API
- Static pressure at 0.1 bar

2.5 *Design considerations*

Based on the level of uncertainty in material property i.e. grain boundary distribution, manufacturing uncertainties like variable transition fillet radius at base of airfoil both at LE,TE and at chord cover. Over- speed of 11% and the control system to break the separation margins and design speed. Temperature variation and number of over-speed in one start up and shut down cycle are the three important parameters with a common factor, which goes in to the margin limits at minimum yield of the material as 'Factor of Safety'.

$$\sigma_{allowable} = \frac{\sigma_{minimum}}{(Overspeed)^2 * (AdditionalFOS)} \quad (9)$$

The design rule margins are necessary for uncertainties present in material, manufacturing, assembly and on site operating conditions. Since speed plays a vital role in rotary machinery and centrifugal force $F_c = m\omega^2 r$ special care to be provided for speed load which varies as a square function. Factors for temperature, manufacturing and material uncertainty is taken 15% over minimum yield at designed speed. In the present work as per API standards 121% margin is considered for 11% over-speed due to full fuel throw–off condition. Additional margin of 15% is considered for manufacturing uncertainties. Mahesh Shankar, et al., (2010)

3. Problem Description

To estimate the peak stress and the average stress of the various cross sections of the blade and disc when running at 121 % over-speed of the machine then Neuber's rule is applied on linear static run to achieve nonlinear total strain. The method of assessing LCF life of a component with the effective use of cycle versus strain amplitude relation to determine the life of an intake assembly for a hot gas expander based on the Finite element analysis results comparing with graphical and analytical.

4. Methodology

Turbo machinery blade and disk is analysed by exploiting the cyclically symmetric nature of the LP stage assembly, thereby reducing the computational time. One cyclic sector of the system is considered for the analysis. The geometric model is generated using CAD package. Surface to Surface contact pairs were created between the blade and disk mating surfaces. Frictionless contact is assumed at the butting faces. The blade is allowed to talk only to disk butting face under centrifugal pull thereby simulating the highest possible pull. A matching node pattern is maintained at the blade root and disk hook pressure faces (in order to avoid any ambiguities on stress values due to contact elements), where the loads transfer between blade and disk takes place. Cyclic boundary conditions are applied. The blade and disk both are assumed to be made up of chrome steel with yield stress of 585 MPa, Young's modulus 210 GPa, density 7900 kg/m^3 and Poison's ratio 0.3. Mahesh Shankar, et al., (2010)

5. Results and Discussions

The Finite element analysis is performed with the loadings described in section 4. The analysis considered elastic deformation only. The equivalent peak von mises stress is about 2978MPa at blade root fillet shown in Fig 2 and 2968MPa at disc root fillet shown in Fig 3, which is higher than the yield strength of the material of construction (585MPa) .The peak stresses, is resulting at few nodal locations at fillet region. These peak stresses are linear which may not be feasible to consider. However, the correction factors for these peak stresses are done through bilinear runs and are neuberised. These elastic peak stresses are used on to the stress strain curve of the material to obtain the nonlinear peak stress. In the similar fashion, the total strain 'S-N' curve can be computed through neuber's technique as well through bilinear analysis. The nonlinear stresses and the neuber stresses developed at the blade root fillet and disc root fillet are shown graphically in Fig.4 and Fig.5. It is evident that the linear static stress results are far away from bilinear and neuberised stress values. It is also significant that bilinear curve is closely associated with neuberised stress values at 50% speed, 100% speed and 121%speed as per API standards. The average stresses from gross yielding point of view in blade root fillet, airfoil base and disc root fillet at minimum cross section is tabulated in the Table.1.It is observed that at critical location shown in Fig.6, the average stress is 446MPa which is well below the yield at 121% over- speed. As per weak link analogy, the disk is stiffer than the blade. The weaker sections in the assembly can be replaced if in case of failures but modifications or repairs are difficult, sometimes impossible, as the rotor is a single forged unit.

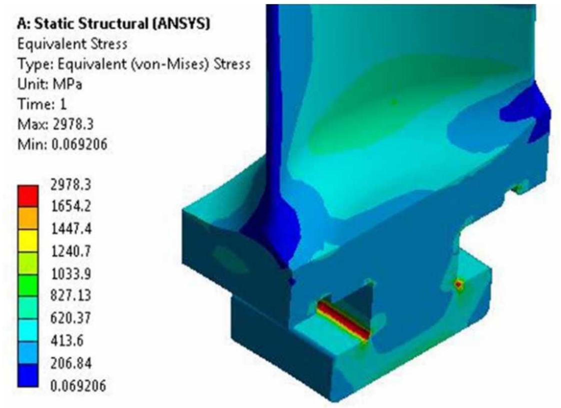


Fig. 2. Blade root fillet equivalent stress distribution

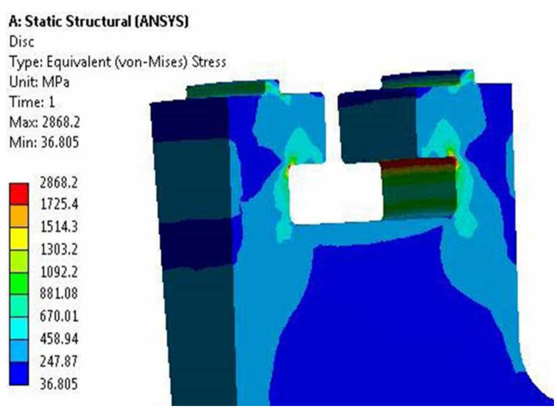


Fig. 3. Disc root fillet equivalent stress distribution

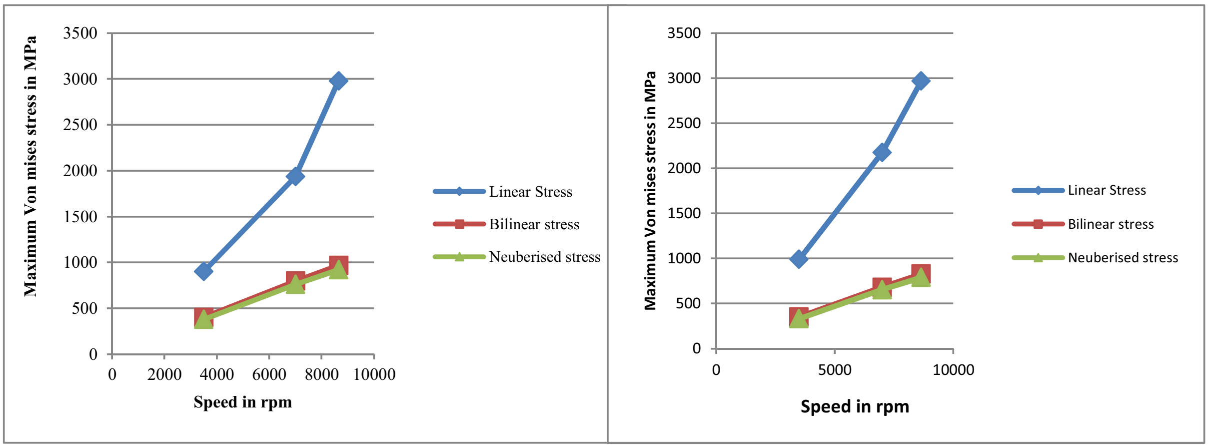


Fig. 4. and Fig.5. Maximum Von mises Stress distribution at blade root fillet and disc root fillet for various speeds

Table.1.Average stress distribution at 121% speed

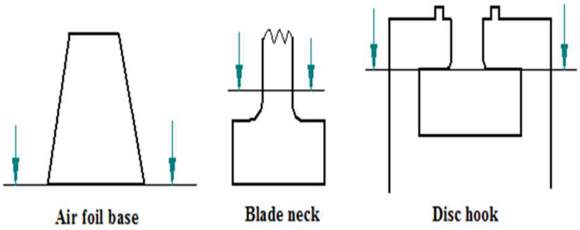
Critical Region	Average stress in MPa
Airfoil Base	446
Blade Neck	275
Disc Hook	333

6. Fatigue Loading On Steam Turbine Blades

In the present work, it is desired to achieve minimum of 5000 LCF cycles (start up and shut down) in both blade and disc. Often the blade root fillets and disk fillets undergo highest peak stress/strain for every cyclic loading.

Therefore some amount of plastic strain gets accumulated for every startup and shut down cycle. The blade loading pattern is as shown in Fig.7.

Therefore the total strain accumulated = Sum of (total elastic + total plastic) based on the desired amount of startup and shut down cycles required for a given component, a common factor should be considered like factor of safety for manufacturing and thermal uncertainties to decide upon the minimum number of startup and shut down cycles required to initiate the crack in blade. As the minimum number of startup and shut down cycles required to initiate the crack in blade can be found out using LCF analysis, strain-life approach can be used for the analysis.



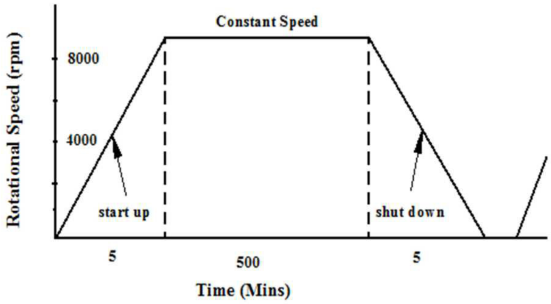


Fig. 6. Critical region in blade and disc

Fig. 7. Blade loading pattern for fatigue life

7. Results of Fatigue Analysis

Section 2 describes the input parameters required for fatigue analysis. Based on assumptions the fatigue life of blade and disc is estimated using Ansys fatigue tool. Fig.8. indicates the margin of possible life available for the present work in blade and disc at various locations. The blade and disc root fillet is the most critical possible location which undergoes stress reversals. The fatigue tool indicates life of 10834 cycles at blade root fillet which lies in close proximity of analytical method using USM and graphical which is indicated in Table.2. Using USM the strain versus Number of cycles of failure plot is shown in Fig.9. with the life estimation of 10500 cycles.

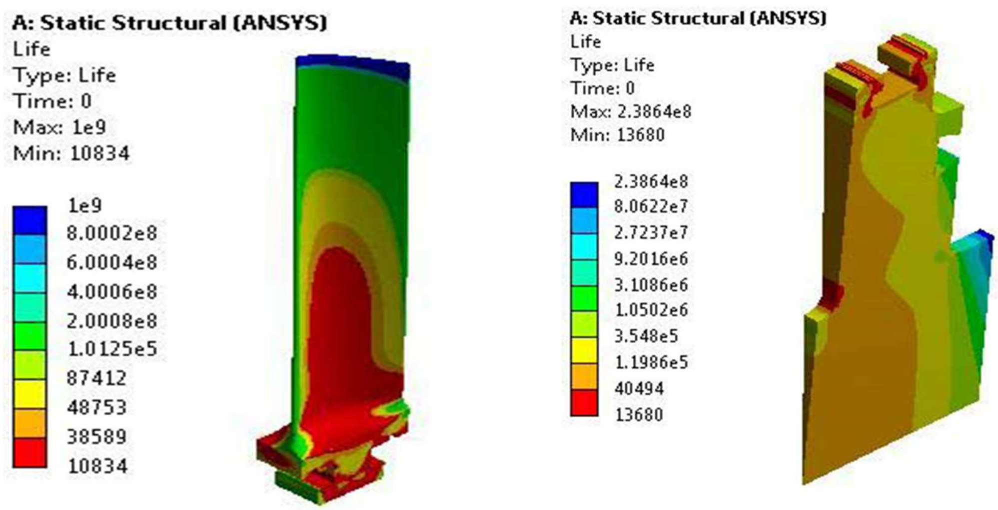


Fig. 8. Fatigue life of blade and disc

7.1. *Calculation Of Life Estimation Of Steam Turbine blade*

Analytically the life estimation of blade is calculated as, N_f = 10486 cycles

Graphically the life estimation is shown below with the Table 2.

Table2. No. of Reversals Vs Total Strain

No. of Reversals	Elastic strain	Plastic strain	Total strain
1	2.83E-03	0.873705	8.77E-01
1000000000	2.35E-04	3.48E-06	2.39E-04

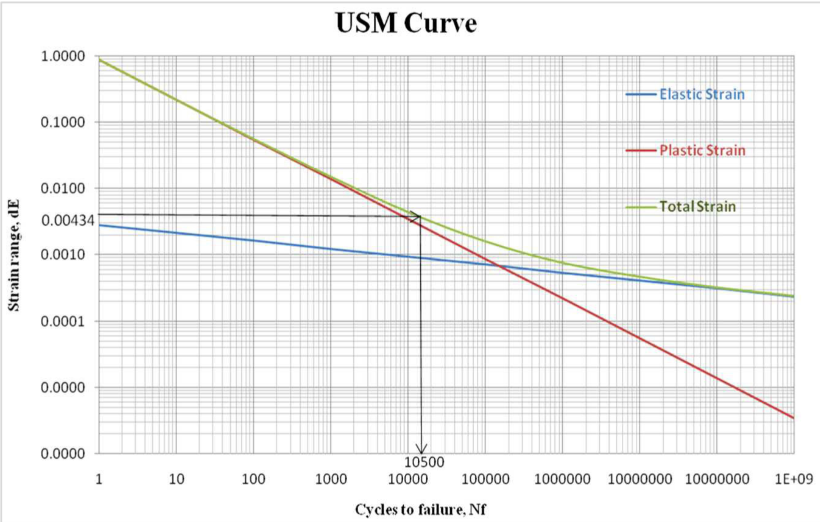


Fig.9. Strain range Vs Cycles of failure. Murari, P. Singh., et.al (2005)

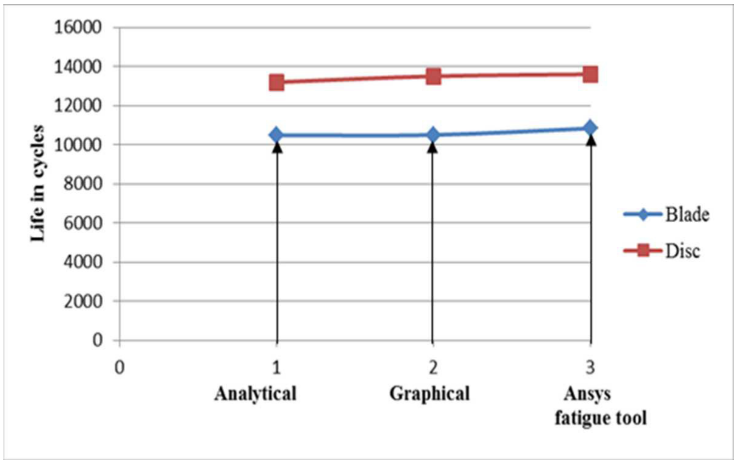


Fig.10. Results of life evaluation for different approaches

From Coffin-Manson equation we obtained number of cycles to initiate a crack at critical regions of the blade and disc. These life cycles are greater than 5000 cycles as per the standards under Zero-Max-Zero load conditions. From the above it concludes that the blade and disc is safe under these load conditions. Proposed method of LCF life evaluation based on critical strain approach, using Universal slope method results in achieving desired start up and shut down cycles for crack initiation. The life evaluation for analytical, graphical and Ansys fatigue tool approaches for blade and disc is as shown in Fig.10.

8. Conclusions

Preliminary design considerations, classical fatigue theories and commercially available tool revels the possible LCF life of turbo machinery blade with following observations

- The mechanical integrity of bladed disc assembly is ensured using simulation technique to build customized methodology
- Geometric and material nonlinearities are taken care for bilinear and neuberised methods carried out using strain life approach
- The correlation between bilinear kinematic hardening and neuber's rule are successfully established to estimate the total strain and to reduce the computational time which can be effectively made use in determination of fatigue life in industrial blades
- Coffin Manson equation and USM are effectively utilized to estimate the minimum number of cycles required for the crack to initiate in blade root fillet and disk root fillet.
- Fatigue life evaluation is carried out for blade and disc assembly using commercial package Ansys.

The study involves a blend of classical and simulation approach to evaluate the fatigue life of a component purely non-linear in nature with linear static results. The customized methodology can be successfully adopted for blade life estimation.

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