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DESIGN OPTIMIZATION OF T-ROOT GEOMETRY OF A GAS ENGINE HP COMPRESSOR ROTOR BLADE FOR LIFING THE BLADE AGAINST FRETTING FAILURE

Kumar.K.Gowda
Vivekananda Institute of Technology
Bangalore- 560074, Karnataka, India
kumartbg@gmail.com

S.L. Ajit Prasad
P.E.S. College of Engineering
Mandya - 571401, Karnataka, India

Vinayaka. N
National Institute of Technology
Durgapur - 713209, West Bengal, India

ABSTRACT

Stress Concentration Factor (SCF) is significant in machine elements as it gives rise to localised stresses which lead to peak stresses introducing cracks which propagate further and hence the component fails before the desired design life. Turbine blades are subjected to high centrifugal stresses and vibratory stresses in a Gas Engine HP Rotor. The vibratory stresses arise due to air wake flow excitations called Nozzle Passing Frequency (NPF). Hence, Turbomachinery industry calls for an optimum structurally rigid blade root geometry. An optimum blade root was defined, as a root with practical geometry, which when loaded returns the minimum fillet SCF. In the present work an approach has been done for design optimization of fillet stresses at sharp edges of T-root blade, optimization of platform dimensions, shank dimensions , root land dimensions and to ensure that stress distribution is uniformly spread along the filleted width of the root land on both sides of the blade, which otherwise will lead to crack initiation, propagation and hence, fretting failure at blade root lands. This may further lead to blade lift and effect on stage and overall gas engine failure over a period of cycles. Hence, a special attention is made on SCF of the T root -blade which fails and to guarantee for safe and reliable operation under all possible service conditions. Finite Element Analysis (FEA) is used to determine the fillet stresses and Peterson's SCF chart is effectively utilized to modify the blade root. The root is modified due to the difficulty in manufacturing the butting surface of the tang which grips the blade to the disk crowns having small contact area. The blade height is suitably designed using Campbell diagram by ensuring the working frequency is well within 6e excitations for the specified operating speeds. Hence, increasing the life of the HP compressor blade.

Keywords: Stress Concentration Factor,Gas Engine HP Rotor, Nozzle Passing Frequency, T-root blade, Fretting failure, Fillet Stresses, Blade root, Peterson's SCF chart, Blade height, Campbell diagram.

INTRODUCTION

Compressor blades face high vibrations, thermal stresses and excitations along its flow path. As a result, blades are subjected to high centrifugal stresses and vibratory stresses in a gas engine HP rotor. The vibratory stresses arise due to air wake flow excitation called Nozzle Passing Frequency (NPF). Nozzle wake excitation are caused by the aerodynamic force-fluctuations seen by the rotating blade as it passes each blade. This is seen by the rotating blades as excitation at the nozzle passing frequency.

The blade-root attachment takes the entire centrifugal stresses induced at high speed rotations[1,12]. As the blade root attachment area to the disk is highly stressed, it needs to be properly designed, which if not addressed can lead to fretting crack initiation due to stress peaks in the root, This is followed by crack propagation over a period of time and the blade may lift with portion of disc cut and hit the compressor inner casing and cause lot of noise, hit other blades and initiate crack in them. Which may lead to rotor stage failures in that stage and adjacent stages and the Gas Engine wont perform to its efficiency and may lead to shut down. This will cause a huge economic loss and entire rotor has to replaced.

Failure can occur with crack initiation at the stress raiser location and propagation. Geometrically the ratio of maximum stress to the average or nominal stress is called as Stress Concentration Factor (SCF) and is denoted by 'K_t' [2]. At critical areas the fillet stresses are modified using the Peterson's stress concentration factor chart [1,12]. Geometric discontinuities cause a large variation of stress locally and often produce a significant increase in stress called as 'Stress Concentration'. This can also appear when loads are applied over a small area or at a point. Geometric discontinuities are often called as 'Stress Risers'. Examples of stress risers include holes, notches, fillets and treads in a structural member.

Stress Concentration Factors are significant in machine elements as it gives rise to localized stresses for any change in the design of surface or abrupt change in the cross section. Almost all machine components and structural members act as 'Stress risers'. These discontinuities are very dangerous, as it leads to failure due to the peak stresses introducing cracks. These cracks propagate to catastrophic failure before the desired design life. Hence, it is very much essential to analyze the stress concentration factors for critical applications like turbine and compressor rotor blades. Finite Element Analysis (FEA) with appropriate mesh is used to determine the fillet stresses and Peterson's SCF chart is effectively utilized to modify the blade root [1].

The most comprehensive source of stress concentration factors for commonly encountered geometries has been compiled by Peterson (1953, 1974). However in these references, the stress concentration factors for only filleted shafts are available and are only approximations based on photo elastic results for two-dimensional strips. The relation between two and three dimensional stress concentration factors is made by assuming an analogy which exists between a circumferential fillet and a circumferential groove. This is the limitation of the Peterson Graphs for estimation of the stress concentration factors [2].

Now the design of experiments techniques (DOE) and optimization methods are used to optimize shape and minimize peak stress values, so as to improve structural integrity of bladed disk

Though we are well aware of various blade root geometries used for gas turbine engine like fir tree, dove tail, straddled, pinned, welded etc. here, an approach has been made to optimize a T-root geometry to make it suitable for HP Gas turbine engines for lifing the blade against fretting failure and a concept of blade root evaluation technique for MI is customized using simple T-root geometry.

FRETTING FATIGUE AND FRETTING WEAR

Fretting is a special wear process that occurs at the contact area between two surfaces which has oscillatory motion of very small amplitudes[7]. The wear occurs due to relative slippage between the contact surfaces. During fretting the fatigue strength decreases to less than one-third of that without fretting. The strength is reduced because of concentrations of contact stresses such as contact pressure and tangential stress at the contact edge, where fretting fatigue cracks initiate and propagate[6]. In this paper fretting-wear process was estimated using contact pressure and relative slippage.

Fretting is a structural damage mechanism arising from a combination of wear, corrosion and fatigue of two contact surfaces subjected to an oscillatory loading. It also acts as severe stress concentrator at the edge of contact and a flaw generator leading to premature crack nucleation, when compared with fatigue process without fretting contact.

The failure induced by fretting fatigue has gradually become a prominent issue during long-time service and in some case micro slip at the edge of a contact zone can reduce life as much as 40% to 60% [8,9]. There are two key issues in the analysis of fretting fatigue tests. One is solving the contact stress accurately; the other carrying out fretting fatigue tests and trying to find suitable parameters for fretting fatigue life prediction.

Small displacement amplitudes, including partial slip, favor cracking, whereas large dissipative sliding gross slip amplitudes favor wear[10]. Fretting wear analysis was first addressed using Archard's wear law, which expresses the increase of wear volume as a function of accumulated Archard's work (ΣW), defined as the product of normal force and total sliding distance multiplied by the so-called Archard's wear coefficient(i.e $V=K_V \times \Sigma W$) [11]. In this paper an approach has been done to determine the wear using the above approach along with linear sliding velocity.

NOMENCLATURE

SCF - Stress Concentration Factor
DOE - Design of Experiments
HCF - High Cycle Fatigue
R - Disk Fillet
R2 - Blade Fillet
W - Butting Width
FEA – Finite Element Analysis
UFCS- Upstream Fillet Contact Surface

DFCS- Downstream Fillet Contact Surface
MI- Mechanical Integrity
CD –Campbell Diagram

OBJECTIVES

1. Reducing the SCF using Peterson's chart.
2. Parametric optimization of blade T- root geometry.
3. To shift the peak stresses acting at sharp edges to the center of filleted T- root land
4. Determination of Fretting wear depth and wear rate and its comparison for UFCS (High Pressue side) and DFCS (Low Pressure side).
5. Fatigue life evaluation for lifing the blade against the fretting failure.
6. To provide a customized methodology which can be extended even to dove-tail and fir-tree roots of an aircraft gas engine blades

BLADE TERMINOLOGY AND NEED FOR FRETTING FAILURE STUDY

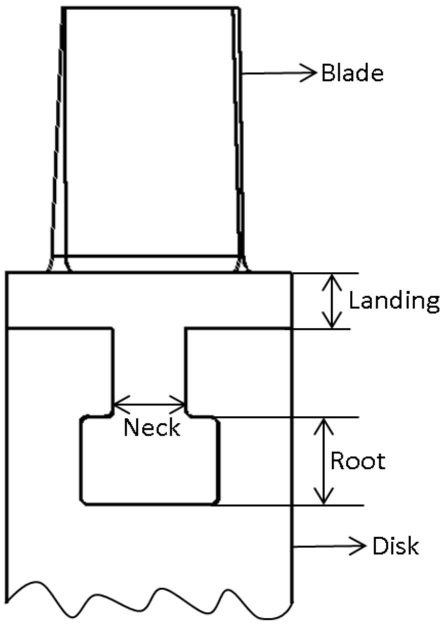

Figure .1 Blade Terminology

The above figure.1 shows the different parts of the blade butted to the disk portion used in aircraft engines. At the place where tight butting happens, it results in causing peaks stresses, which may induce a crack and the crack may propogate deep into the root causing a total stage failure of the rotor as the blade with root will get lifted. Hence, there is a neccesity to identify where the stress peaks happen in the contact region between the blade root and disk hub to solve the problem of vibratory stresses leading to a failure called fretting failure[1,12]. Also there is need for parametric optimization of necking area to avoid failure due to fretting.

Fretting happens in contact region and fatigue happens in the blade root fillets; which is clearly highlighted in the figure.2 below. Fretting usually happens on over a period of time at low pressure end of the fillet land area, as there is less slippage; hence lesser wear rate when compared to high pressure end, wherein, the slippage is less due to tight fit and hence lesser wear rate.

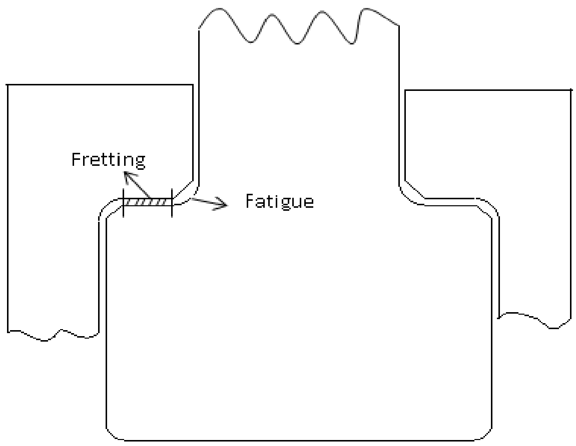

Figure 2. Fretting and Fatigue location in blade root and disc interface

DESIGN OPTIMIZATION OF ROTOR BLADE T-ROOT

In engineering that involves designing, simulation and fabrication, optimization helps to achieve ultimate perfection. Hence, there is a need for optimization to get the most out of a given system

Design of experiments techniques (DOE) and optimization methods are used to optimize and minimize peak stress as design goal[5]. Since, peak stress gradient decides the minimum number of startup and shut down cycles required for crack initiation. The contact butting surfaces talk to each other and can lead to fretting over a period of time. The above two factors can reduce the stiffness and due to thermal gradients softening can result in damping frequency. Variation in the frequency level at contact butting surface can affect High Cycle Fatigue and wear coefficient.

To reduce the SCF using Peterson Chart, which is used usually for radial blade-lift analysis, we now perform parametric optimization of the blade T-root as shown in figure.3; which shows the selected parameters for the T-root optimization and is as tabulated in table.1 with considered optimization upper and lower bounds to its right side column.

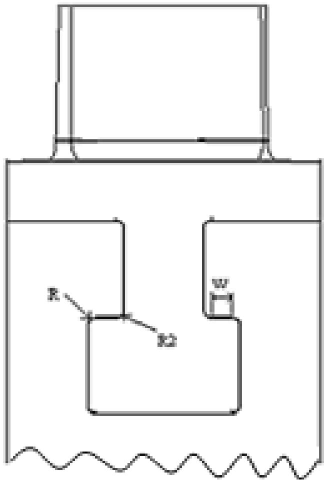

Figure 3. Parametric optimization of T-root.

Table1: Upper and lower bounds values for selected parameters

Parameter	Description	Lower bound in mm	Upper bound in mm
P2 - R	Disk fillet	1.5	1.8
P4 - R2	Blade fillet	1.2	1.5
P7 - W	Butting width	4	4.4

RESPONSE CHART FOR EQUIVALENT STRESS MAXIMUM

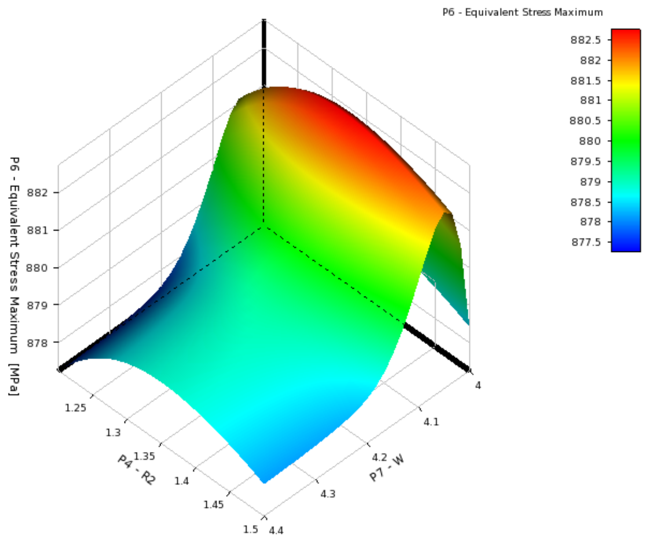

Figure 4. Response chart from Design Optimization.

The above figure.4 shows the response surface which gives the response of parameters R and W for equivalent stress. Based on this response the candidate points is chosen.

Later the original model of the blade root is modified using DOE to optimize blade fillet radius and blade butting surface to obtain the final model as shown in the figure.5 below.

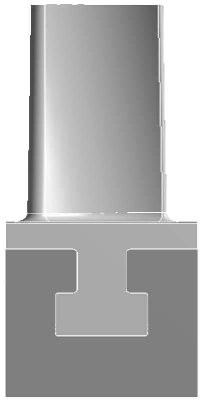

Figure 5. Optimized Blade Model

THE BLADE AND DISK PROPERTIES

Material chosen for both blade and disk is Ti-6Al-4V (Grade5) having material properties Young's modulus, density, yield strength and Poisson's ratio as 1.138x 10^5 MPa, 4430 kg/mm^3, 880 MPa and 0.42 respectively, Blade height arrived upon resonance check using CD is 100 mm.

RESONANCE CHECK USING CAMPBELL DIAGRAM

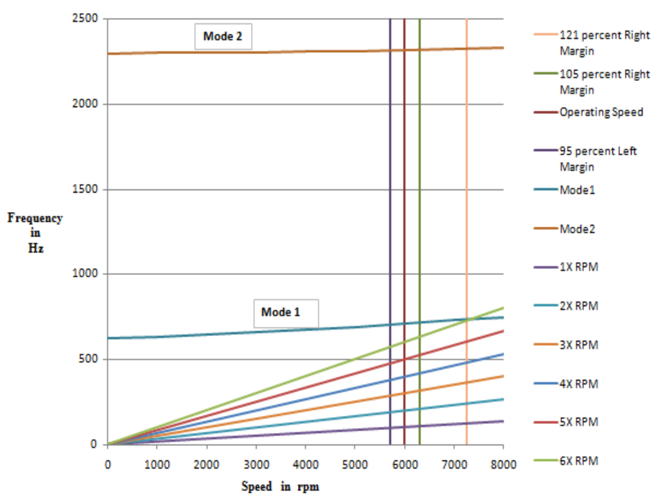

Figure 6. Campbell Diagram of optimized final blade

Figure.6 shows the Campbell Diagram which is an Industrial best practice used to check for blade resonance. Here, the separation margins considered is left margin 5% and right margin 5% for an operating speed of 6000 rpm and over-speed margin is considered at 121% as per API standards. The results shows that all excitations from 1X to 6X are safely passing outside the separation margins considered. In practice usually upto 6X or 6e powerful excitations are checked for resonance which has been very well achieved in this case and hence the blade is safe from resonance and provides better structural performance.

CYCLIC SYMMETRY RESULTS AND DISCUSSIONS

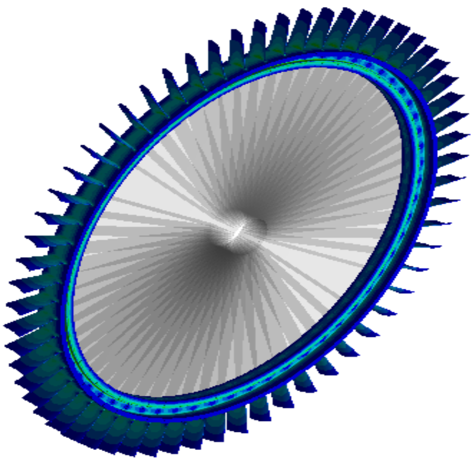

Figure.7 Cyclic Symmetry Analysis of Blade

Figure.7 shows that analysis is carried out using cyclic symmetry option in the FEA commercial software, with step input for design rotational velocity of 6000 rpm.

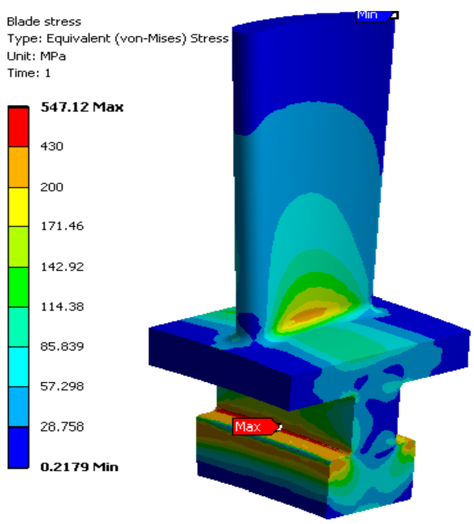

Figure 8. Von-Mises Stress Distribution

Figure.8 shows the maximum and minimum peak stresses in the blade model after parametric optimization. In the baseline model before optimization the peak stress was observed at the blade fillet edges. Now upon parametric optimization the peak stress is shifted to the center of fillet T-root land as shown in figure.8; which will take much more time for crack initiation and propagation, as now the crack has to propagate deep inside the root for a greater distance against the material mass; which provides more stiffness due to material volume and young's modulus ; hence, lifing the blade against the possibility of blade lift which else would have caused fretting failure at blade T-root.

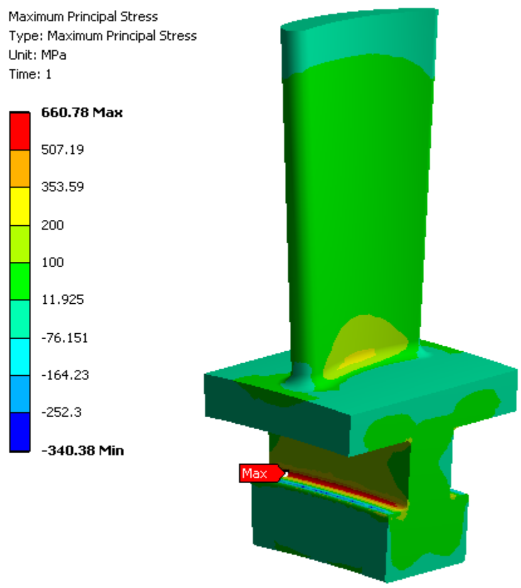

Figure 9. Maximum Principal Stress Distribution

In figure.9 the maximum stress is clearly reflected and shows the uniform distribution in the filleted region along the fillet root land and no peak stresses at fillet edges. This shows that the best blade design condition is achieved as stress distribution is not localized and causes no peak stresses.

FRETTING WEAR CALCULATIONS

Fretting causes the material to wear and induces cracks at contact edges. In early stages, these cracks tend to close due to high contact pressure and propagate slowly [6]. Here, we are only concentrating on the wear depth evaluation at contact edge. With the obtained contact pressure and slippage amount from FEA commercial software, wear is calculated using **Classical Archard's Equation:**

$$W = K \times P \times S \qquad - (1)$$

Where 'W' is the wear depth, 'P' is the contact pressure, 'S' is the slippage and 'K' is the wear coefficient. 'K' can be defined as the wear per unit load per sliding distance and it changes with sliding speed between the blade root and disc hub interface.

For Ti-6Al-4V, wear coefficient K= 27.32 X 10^{-3} mm^3/N-mm for linear velocity of 25 mm/s [4]

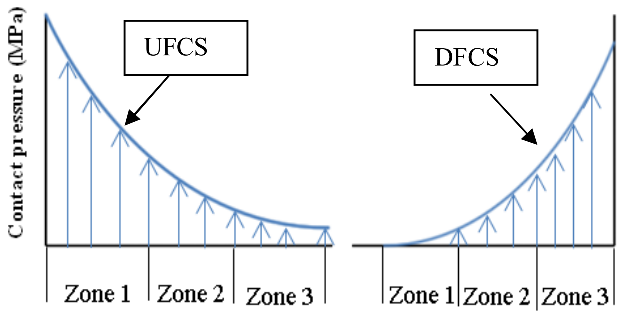

Figure 10. Contact pressure distribution over upstream and low stream contact surfaces

Figure.10 shows the physics of the contact pressure distribution in the 3 zones considered in blade T-root and disc hub interface. The up-stream and down-stream contact surfaces of blade root are divided into three zones for the fretting wear depth analysis and the estimated results using Classical Archard's Equation are as shown in the figure.11. Figure.11 shows that DFCS wear is greater than UFCS.

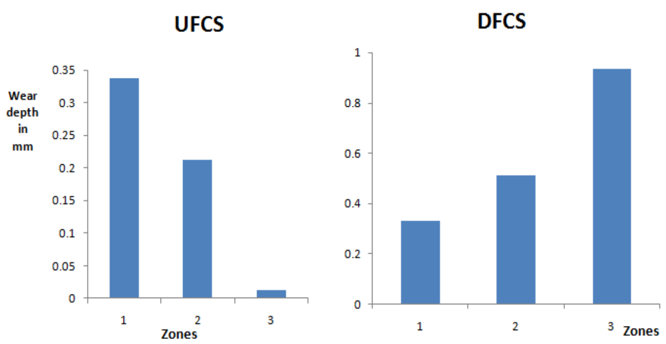

Figure 11. Wear Depth Comparison for UFCS (High Pressure side) and DFCS (Low Pressure side)

The upstream side of the blade fillet at T-root i.e at the left side portion of the blade has high contact pressure and lesser clearance and is tight. This causes low slippage and hence lesser wear rate in the UFCS. Whereas in downstream side of the blade fillet at root i.e at the right side portion of the blade, there exists low contact pressure and is loose. Hence, there exists high slippage at DFCS causing greater wear rate. The simulated values of wear rate at the 3 zones considered along each side of UFCS and DFCS respectively are plotted as shown in the figure.12 which tells that DFCS exponentially increases whereas UFCS exponentially decreases respectively.

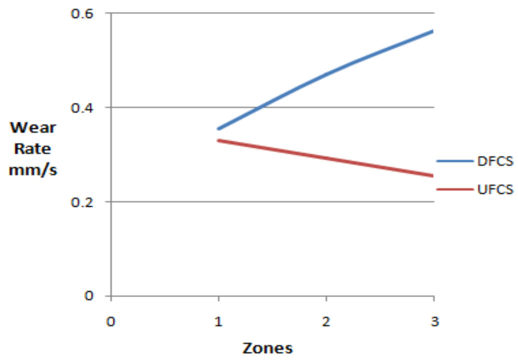

Figure 12. Wear rate comparison for UFCS and DFCS

FATIGUE LIFE EVALUATION

Fatigue life estimation is a very important criteria to know the reliability of the component for the subjected varying loads. The components subjected to repeated cyclic loads should be checked for its life to know when it may fail for its replacement by another of the same. In this work fretting fatigue evaluation has been performed to check the lifing of the blade against fretting failure. The most widely used fatigue life model is often expressed by the 'Coffin-Manson' equation, relating life to plastic strain range through a power law [3]

Coffin-Manson relationship between Fatigue life and total strain is as follows :

$$\frac{\Delta\varepsilon}{2} = \frac{\sigma'_F}{E}(2N)^b + \varepsilon'_F(2N)^c \quad - (2)$$

Where:
ΔE- denotes the maximum principal strain at the tight end side whose value is obtained from FEA software as shown in figure.13
N- denotes the fatigue life cycles.

The Strain life parameters considered for Ti-6Al-4V are:
Strength coefficient, σ'_F = 1737 MPa
Strength exponent, b = (-0.085)
Ductility coefficient, ε'_F = 0.396
Ductility exponent, c = (-0.684)

$$\frac{0.00478}{2} = \frac{1737}{1.138 \times 10^5} \times (2N)^{-0.085} + 0.396 \times (2N)^{-0.684}$$

N = 1.489 X 10^9 cycles

Thus, we have achieved approximately **19%** more life in the optimized blade model when compared to base line model (without optimization) life which was about **1.250x 10^9 cycles** initially.

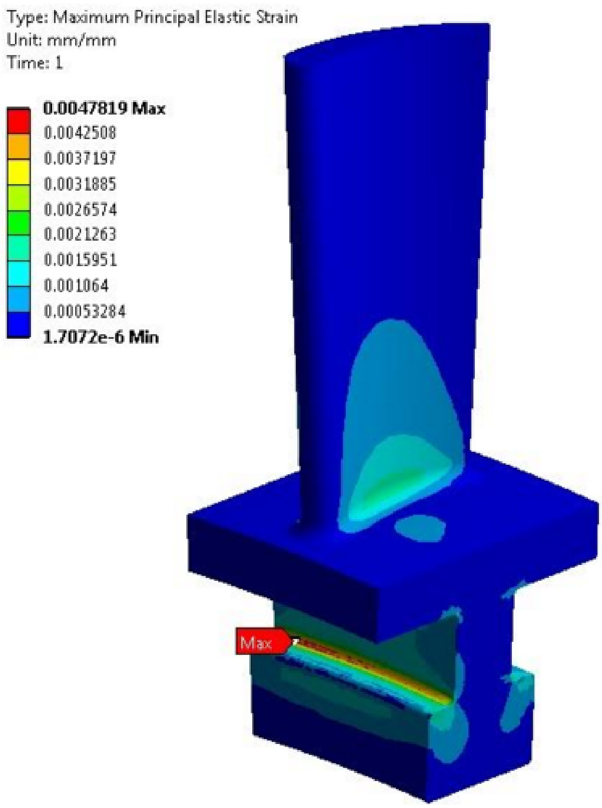

Figure.13 Maximum Principal Strain Distribution

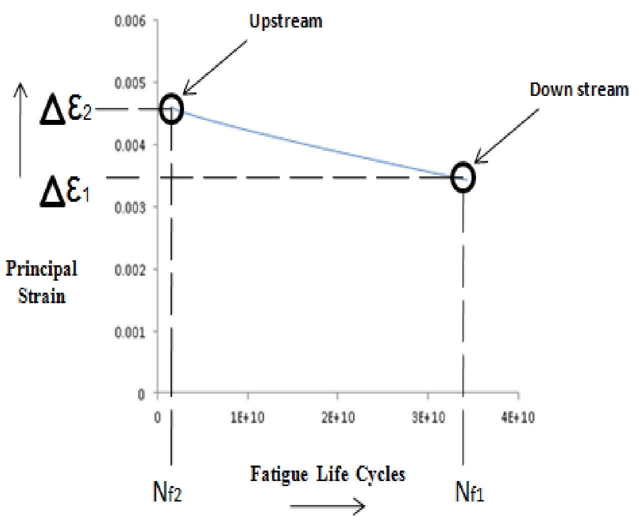

Figure.14 Optimization of Peterson chart values used in DOE

Figure.14 shows the plot of principal strains v/s fatigue life for UFCS (High Pressure side) and DFCS (Low Pressure side). Results show that DFCS provides better fatigue life cycles and corresponds to lesser principal strain.

CONCLUSIONS

1. SCF has been reduced to a great extent using Peterson's chart.
2. Parametric optimization of blade T- root geometry has been achieved and it clearly showed that we get peak equivalent stress of 882.5 MPa at butting width (W) = 4.05mm and at Blade fillet (R2) =1.3 mm and hence we have to chose other acceptable dimensions in the range chosen to get better structural performance .
3. Peak stress is shifted to the center of filleted T- root land
4. Upstream side has high contact pressure and lesser clearance causing low slippage and hence lesser wear rate; whereas in downstream side low contact pressure hence high slippage causing greater wear rate
5. Fatigue life evaluated result after lifing the blade against the fretting failure showed an increase in **19%** more life when compared to base line
6. DFCS provides better fatigue life cycles and corresponds to lesser principal strain
7. The results show that this customized methodology can be extended even to dove-tail and fir-tree roots of an aircraft gas engine blades

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