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# T-root blades in a steam turbine rotor: A case study

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ABSTRACT

The present work illustrates, 3D finite element analysis (FEA) of low-pressure (LP) steam turbine bladed disk assembly are carried out at a constant speed loading condition. The prime objective is to study structural integrity of bladed disk root with aid of design considerations at design stage. Secondly, design rules are developed for structural integrity of blades and disk considering a factor of safety for material, manufacturing and temperature uncertainties. These design rules are in turn used as design checks with aid of finite element analysis results.

Investigations are performed based on Neuber formulae for solving a highly non-linear problem employing linear analysis tool ANSYS 12.0. Local peak stresses at blade and disk root fillet of linear analysis is used to identify the equivalent non-linear stress value by strain energy distribution method for estimating the minimum number of cycles required for crack initiation for low cycle fatigue (LCF) calculations.

Design methodology is developed to address the structural integrity of blades at design point and for off-design conditions.



## 1. Introduction

The importance of power generation through thermal and nuclear plants is very much felt due to the uneven fluctuations in monsoon, which has made power generation and supply inconsistent. With ever-increasing demand for power, due to increased industrial activities, it has become imperative to generate power economically and consistently. One such attempt has resulted in increased use of steam turbine for power generation. In a steam turbine, numbers of blades are used for energy transfer [1]. The blades are subjected to centrifugal force hence, become critical parts, which affect the satisfactory function of a turbine [2]. Bladed disk are most flexible elements in high speed rotating machinery. Due to rotation, blade root gets tightened in the disk slot to transmit the centrifugal load. The mating contact may be just two or may increase to six or more for low-pressure blades. While the average stress in the mating surface is fully elastic and well below yield stress, the peak stress at singularities in the groove shape can reach yield stress values and into local plastic region [3]. Last stage LP turbine blades are the most severely stressed blades in the system. Usually these are the limiting cases of blade design allowing the peak stress to reach yield or just below yield conditions. Failure can occur with crack initiation at the stress raiser location and propagation. Now design of experiments techniques (DOE) and optimization methods have become available to optimize shape and minimize peak stress values so as to improve structural integrity of bladed disk. Because the problem is highly non-linear due to centrifugal stiffening and spin softening, considerable time is taken to achieve the optimized root [4]. There is a vast literature available on linear-elastic stress concentration factors $K_t$, depend solely on the specimen/notch geometry and loading. However, in the presence of plasticity at the notch root, the actual stress concentration factor $K_\sigma$, is

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found to be smaller than the tabulated $K_t$, mainly due to stress re-distribution at the yielding zone. In turn, strain concentration factor $K\varepsilon$ at notch root, which strongly affects the fatigue life predicted by strain based approach ($\varepsilon N$) method [5–7] can be much larger than $K_t$. In many cases, these models may provide reasonable estimates of the maximum stresses and strains at the notch root; however, the differences among fatigue life predictions by each rule can be unacceptably large. In addition, these methods do not account for the geometrical changes at the notch root under large displacements, leading to further errors.

In the light of the above, in the present work, knowledge based engineering is applied to develop the design rule as a check for structural integrity of blades at design stage. Neuber's technique of using linear analysis to predict non-linear stress/strain values in the component is effectively utilized to solve highly non-linear problem based on strain energy re-distribution method. Sensitivity study on shape variables of the root geometry is done to finalize the root dimensions for manufacturing feasibility and structural integrity.

## 2. Objectives of present work

The main objective of the study is to carry out structural analyses of LP stage blade disk sector for its strength evaluation. It also includes: (1) Development of a design rule for turbo machinery blades to ensure structural requirement. (2) Effective utilization of design rule to meet structural requirements at par with finite element results as a design check. (3) To modify the blade root to reduce the stress concentration at root fillet and correspondingly at disk hook fillet achieve desired LCF life.

## 3. Blade type

With steam flow path through a turbine, the environment for the energy converging blades varies strongly and, as a consequence, so do the mechanical requirements. These requirements have a strong influence on the choice of material and the design with respect to temperature, wetness and cleanliness of medium, acting forces and other factors as i.e. harden ability and oxidation. Therefore, different blade families exist which can be categorized according to their use in the primary three turbine modules as high, intermediate and low-pressure blades (HP, IP and LP). Generally the LP stage blades are shrouded or laced, because they have to resist huge centrifugal forces and bending produced by the steam loads. Fig. 1 shows the geometry of the LP blade.

## 4. Design rule development and use of factor of safety

Design rule margins are necessary for uncertainties present in material, manufacturing, assembly and on site operating conditions. As per API standards the blades should prove its integrity at design speed and also at off-design conditions. Based on knowledge base engineering in the present work 121% margin is considered for 11% over-speed due to full fuel throw-off

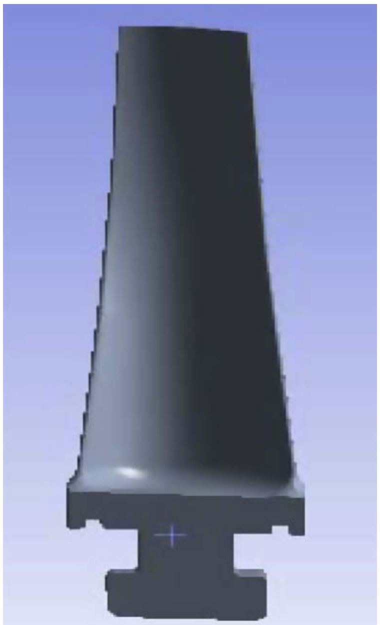

**Fig. 1.** Simple 'T'-root LP blade.

condition. Additional margin of 15% is considered for manufacturing and thermal uncertainties. Therefore allowable stress in component at design speed may be written as

$$\sigma_{\text{allowable}} = \sigma_{\min}/(\text{over speed})^2 * (\text{additional FOS}) \tag{1}$$

Speed load contribution is high in rotating machinery and centrifugal force varies as $Fc = m\omega^2 r$. Care should be exercised for speed load which varies as a square function. With this assumption at design speed, Factor of Safety = (1.212) × (1.15) = 1.68 allowable FOS at different speeds is as shown in Table 1.

This argument shows that the material has 15% additional margin at 121% over-speed condition. The allowable average stress limit at average stage temperature of 750 °C for the blade material with a minimum yield of 585 MPa is 349 MPa.

## 5. Average section stress

It is seen that low-pressure blades are highly subjected to centrifugal and thermo mechanical loads. Since the centrifugal load dominates thermal loads for every mission cycle, emphasis is paid to address mechanical loads for structural integrity of blades. In bladed disc assembly, the blade happens to be the weaker section compared to disc. Replacing blades under failure is much cheaper than that of disc. Average section stresses attained at minimum cross-section at operating and at over-speed condition should be with in the allowable design limits following the design rule both in blade and disc. The design rule is the average section stress in the disc groove should have greater margin compared to blades, based on weak link analogy.

The following are the critical zones identified in a bladed disc assembly. Average section stress in airfoil should be less than allowable stress limits at design speed.

(a) Average section at blade root neck and at minimum cross-section of disc should be within the allowable design limits [8].
(b) Average section stress at minimum cross-section of disc should be 10% less compared to blade root neck average stress, based on weak link analogy.
(c) Average section stress in the disc groove considering loading and unloading characteristics at over-speed conditions should have a margin of 1.18 at 0.2% proof stress of the material.
(d) The behavior constraints to evaluate the over-speed and burst margin may be as follows:
  - Allowable hoop and von-mises stress at the bore is 95% of 0.2% proof stress.
  - Allowable hoop stress at the web is 85% of 0.2% proof stress.
  - Allowable radial and von-mises stress at the web is 80% of 0.2% proof stress.
  - Allowable hoop and von-mises stress at the rim is 67% of 0.2% proof stress.
  - Allowable radial growth < Specified tolerance of 3 mm.
  - Allowable axial growth < Specified tolerance of 1 mm.
  - Allowable AWMHS <72% of 0.2% proof stress at peak temperature in the disc.
  - Allowable AWMHS <64% of UTS at peak temperature in the disk.
  - Disc burst speed $\geqslant$125% of maximum allowable steady state speed of 6000 rpm.
  - Disc over-speed $\geqslant$118% of maximum allowable steady state speed.

In addition to the standard behavior constraints on stresses, deflections, Eigen values and buckling load factors, there are constraints related to "area weighted mean hoop stress" (AWMHS) and "area weighted mean radial stress" (AWMRS). [9] These constraints are specific to rotor design in the aero-engine industry and therefore not readily available from a general-purpose program like ANSYS and hence many non-standard constraints have been incorporated into design rule for effective evaluation of rotor components.

## 6. Low cycle fatigue life approach

Steam turbine blades are on constant speed machines which rarely under go one over-speed for every start up and shut down cycle. The blade loading pattern is as shown in Fig. 2.

In the present work, it is desired to achieve minimum of 2500 LCF cycles in both blade and disc. Often the blade root fillets and disk groove fillets undergo highest peak stress/strain for every cyclic loading (start up cycle). Therefore some amount of

**Table 1**
Factor of safety limits.

| Speed (%) | Allowable FOS limits |
|---|---|
| 100 | 1.68 |
| 110 | 1.38 |
| 121 | 1.15 |

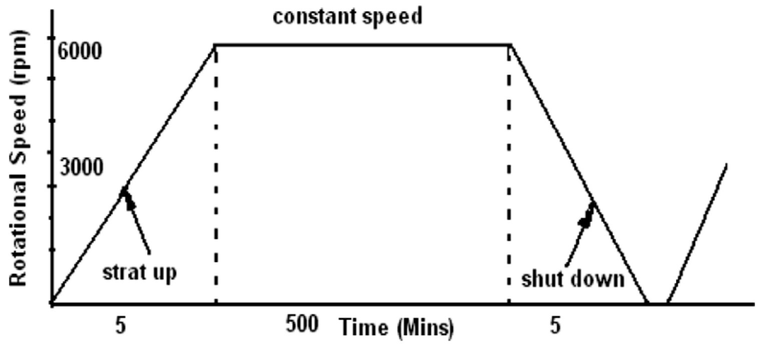


**Fig. 2.** Blade loading pattern.

plastic strain gets accumulated for every cycle [10]. The total strain accumulated = Sum of elastic and plastic strains. Knock down factor of two is considered for fatigue modifying factors like size factors, surface factors reliability factor and temperature factor. For better design with these uncertainties the actual life at design stage is 5000 cycles. Lifing is generally done with classical approach considering all the fatigue strength modifying factors in stress based and strain based approach The $\varepsilon N$ material curve data for a given material at operating temperature is utilized for lifing blade and disc based on local strain approach. For stress based approach the total stress/strain is estimated using Neuber's rule. The non-linear stress and S–N material curve for notched specimens at operating temperature is used to achieve the minimum start up shut down cycles [11]. In the present work, one such effort is made to reduce the peak stresses at blade root fillet to improve LCF life of blade disc assembly.

The stress base theory, which has been recognized as much slower in modeling the crack initiation phase, the most promising approach seems to be based on strain approach. Although the crack being not initiated in the model, the life of the component before the crack initiates is our concern. The basic premises is at the local fatigue response of the material at the critical point i.e., at the site of crack initiation. It is analogous to the fatigue response of a component subjected to cyclic stresses, to properly perform the local cyclic stress-strain history at the critical point. The structure must be determined either by analytical or experimental means. In the light of the above cyclic symmetry pre-stress static analysis with different load step procedure as per the blade loading pattern is conducted using finite element analysis. The available stress/strain in the component from the analysis and material fatigue curve LCF calculations are conducted.

## 7. Finite element model

Cyclic sector of a disk of LP stage is considered. The CAD geometry of one blade with disk sector is modeled using commercially available modeling software Pro-E. The finite element model of LP sector stage bladed disk is generated using ANSYS 12.0 [12], as shown in Fig. 3. SOLID45 element is considered to generate the finite element model. De-generated tet-

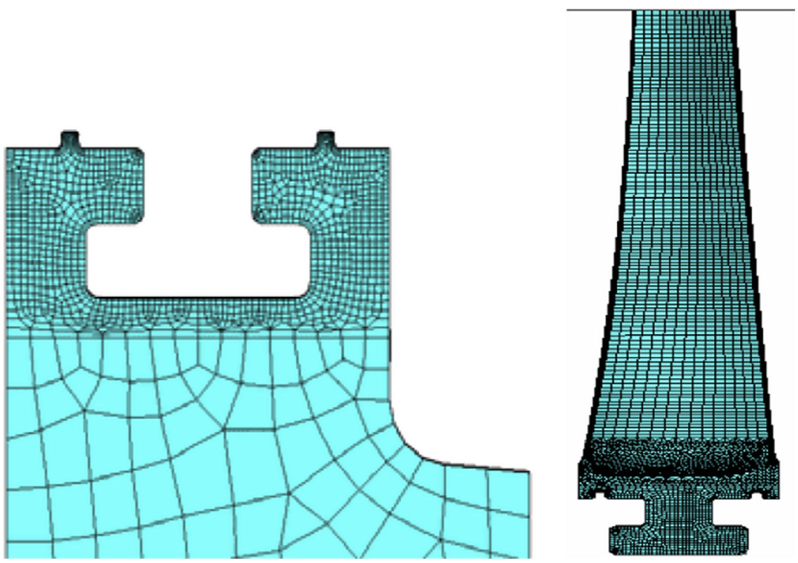

**Fig. 3.** Finite element model of blade disk sector.

rahedral elements do not exist in the mesh, as the de-generated Tet elements are not good for accurate stress calculation. 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.

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 [6] which estimate the plastic stress–strain state from linear analysis runs were utilized. With the application of finite element analysis, notch concentration factors are being inherently considered. The general Neuber's procedure of extrapolating linear stresses into the plastic material region can thus is applied to arbitrary geometries.

The following discussions for comparison and design decision are based on stresses after Neuber's procedure [6]. The geometry shows a peak stress of 1187 MPa in blade root fillet as shown in Fig. 4, it lead to very poor LCF cycles of 5100 cycles, calculated based on the strain approach from Neuber's formula. Also the knowledge based design criteria (established based on experience and experimental tests) limits the peak stress to 1200 MPa from the linear finite element stress analysis, for the component life of 5000 start up/shut down cycles.

The local peak stress at disk hook fillet is 1102 MPa as shown in Fig. 5. The linear peak stress of 1187 MPa after neuberisation yields a total strain which can initiate a crack after the desired LCF life of 5000 cycles. Since both the blade and disk are made of chrome steel the limiting peak stress in linear analysis remains same for both blade as well as disk.

From gross yield point of view, airfoil base, blade root neck and minimum cross-section of the disk are considered to be the critical locations in the assembly. These locations may have local peak stress crossing the yield limit but as per design rule at design speed as well as at over-speed condition these zones should have factor of safety as explained earlier in Section 5.

The average section stress at airfoil base is as shown in Fig. 6. Maximum stress is concentrated uniformly at the pressure face of the airfoil base with average section stress of 232 MPa which is within the designed limit of 349 MPa at design speed. At over-speed condition the maximum allowable average section stress is 508 MPa. The observed average stress at airfoil base is 326 MPa which is within the design limits.

Blade neck is the second most critical region in the assembly since the centrifugal load of the complete airfoil is held at the neck region, the average section stress distribution at the blade neck region is as shown in Fig. 7.

Maximum allowable stress at neck region is 349 MPa at design speed and 508 MPa at over-speed condition. Finite element analysis shows an average stress of 242 MPa and 354 MPa at over-speed. As per design rule, the average section stress at blade neck is within the desired design considerations.

Disk hook fillet is the most critical component in turbine rotor assembly. As per weak link analogy, the disk happens to be stronger 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. The maximum allowable stress limits is similar to that of blade at operating conditions. Fig. 8 shows the average section stress distribution at minimum cross-section of the disk with a magnitude of 231 MPa at design speed and 338 MPa at 121% speed which is within the design limits.

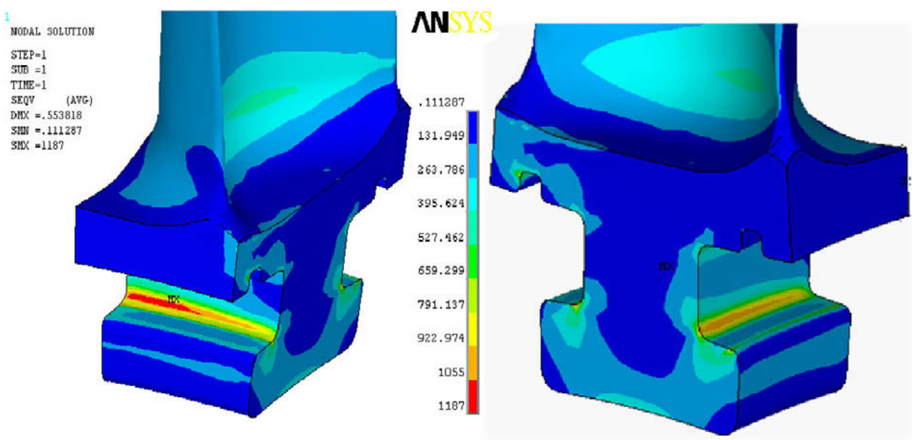


**Fig. 4.** Von-mises stress in elastic domain at 6000 rpm peak stress of 1187 MPa is seen at blade root fillet pressure face and 1055 MPa at suction side.

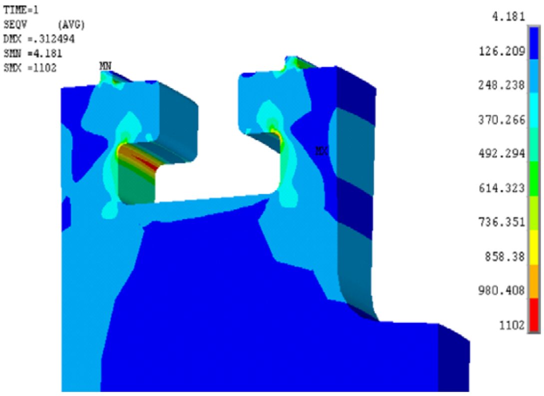


**Fig. 5.** Von-mises stress in elastic domain in disk at 6000 rpm peak stress of 1102 MPa is seen at blade root.

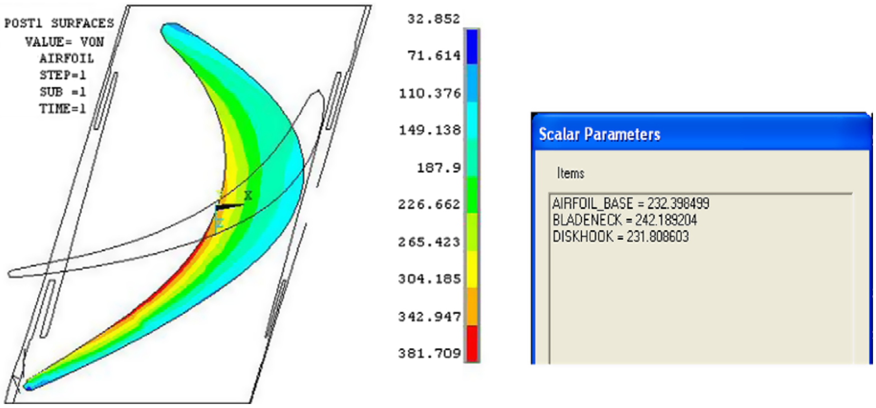


**Fig. 6.** Average stress distribution at base airfoil average stress at 110% speed.

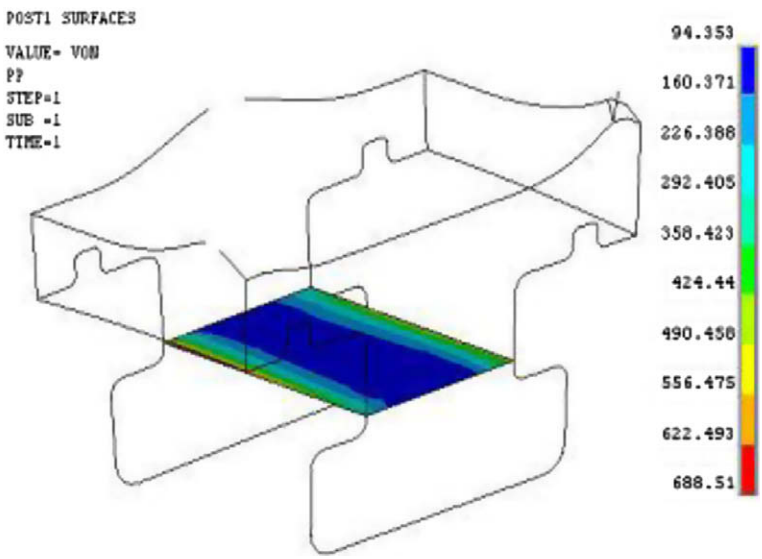


**Fig. 7.** Average stress distribution at blade neck = 242 Mpa.

However, the design rule for average section stress as discussed earlier in Section 5, the following checks (a–d) in airfoil, blade root and disc groove minimum cross-section are well within the allowable design limits of 349 MPa at design speed.

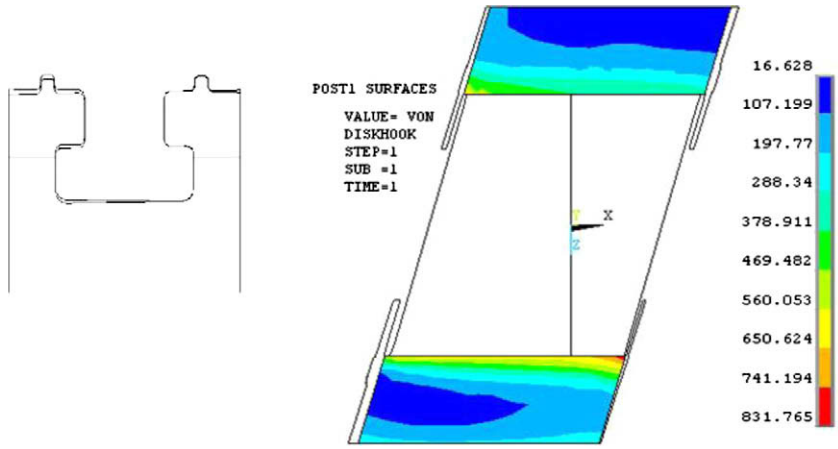


**Fig. 8.** Average stress distribution at disc hook = 191 MPa.

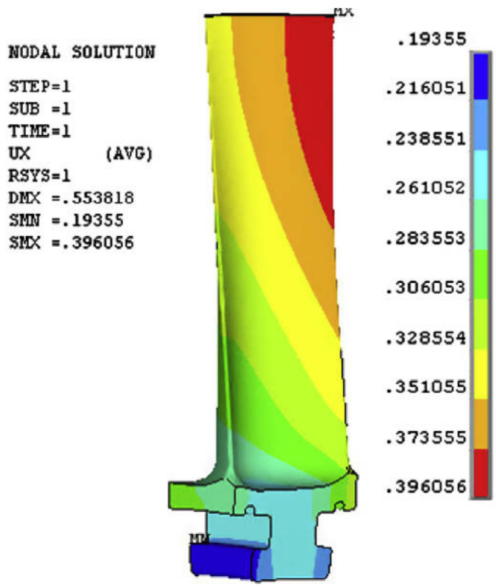


**Fig. 9.** Radial growth at 100% speed = 0.4 mm.

The necessary design margin at over-speed conditions with radial growth at average stage temperature of 70 °C as per design rule is satisfied. The local peak stress/strain at the blade root fillet was not sufficient to achieve the minimum desired 5000 LCF cycle. The radial growth of blade at operating conditions is 0.4 mm as shown in Fig. 9. The allowable limit between blade tip and casing is 3 mm. This indicates that there is no rubbing of blade with the casing even at over-speed conditions.

## 8. Conclusions

The following observations were made pertaining to the LP bladed disk assembly of steam turbine at design speed of 6000 rpm with appropriate boundary conditions as explained earlier.

For the base line model linear stress analysis approach has shown a peak stress of 1187 MPa in blade root fillet and 1102 MPa at disk hook fillet, when Neuberised lead to very poor LCF life of 5100 cycles. From gross yielding point of view, average stress in blade neck is with in the desired allowable limit.

The design rule developed is successfully utilized to provide the structural integrity in turbo machinery blades, disk. These rules act as design checks for verifying the finite element analysis results at various design points like average section stress, over-speed margin and radial growth.

The following conclusions could be drawn from the present work:

($\alpha$) Custom made methodological procedure is used for strength evaluation of a LP steam turbine bladed disk assembly using FEA results and classical approach.

($\beta$) The developed design rule was successfully used in safety and strength evaluation of turbo machinery components.

($\chi$) Proposed method of LCF life evaluation based on local strain approach, using Neuber's Rule results in achieving desired startup and shut down cycles for crack initiation.

## References

[1] Kearton WJ. Steam turbine theory and practice. 7th ed. New Delhi: CBS Publishers and Distributors; 1988.
[2] Richter Christopher-Hermann. Structural design of modern steam turbine blades using ADINA. J Comput Struct 2003;81:919–27.
[3] Heinz PBloch. A practical guide to steam turbine technology. Mcgraw Hill; 1982.
[4] Pilkey, Walter D. Peterson's stress concentration factors. John Wiley & Sons; 1997.
[5] Rao JS. Turbine blade life estimation. Narosa Publishing House; 2000.
[6] Neuber H. Theory of stress concentration for shear-strained prismatical bodies with arbitrary nonlinear stress-strain law. ASME J Appl Mech 1961;28.
[7] Kim Hyo-Jin. Fatigue failure analysis of last stage blade in a low pressure steam turbine. Eng Fail Anal 1998;5:93–100.
[8] Taylor RM, et al. Detail part optimization on the F-35 joint strike fighter. AIAA 2006-1868, 47th AIAA/ASME/ASCE/AHS/ASC structures, structural dynamics, and material conference, May 2006, Newport, Rhode Island.
[9] Jean Lemaitre, Rodrigue Desmorat. Engineering damage mechanics: ductile, creep, fatigue and brittle failures. Springer; 2005.
[10] Pereira JC, et al. A low cycle fatigue analysis on a steam turbine bladed disk. 12th IFToMM World Congress, Besançon, June 18–21; 2007.
[11] Rao JS. Transient stress analysis and fatigue life estimation of turbine blades. J Vib Acoust, ASME 2004;126(October):485–95.
[12] Ansys training manual 12.0.