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Fracture Analysis of a First Stage Turbine Blade by FRANC3D

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Abstract. The premature failure of a blade occurred after a service life of 8127 EOH operation houre. This paper presents the fracture analysis of the first stage blade through fractographic and mechanical analysis. Crack growth mechanisms were evaluated based on the microscopic observations of the fracture surfaces by SEM. The analysis of the different region of the fracture surface shows that crack propagation is mainly related with fatigue mechanism. The crack propagation occurred in the pressure-suction side direction. The dynamic characteristics of the blade were evaluated by FEM in order to identify the cause of blade failure. The result depicts that the second mode of vibration might be excited and the vibratory stresses cause to HCF damage of the blade. Eventully fracture analysis of the blade under the presence of a fatigue crack was analyzed by FRACNC3D software.

Introduction

Turbine blades in a typical gas turbine engine are identified as critical locations for stress fields and hence fatigue damage. Blade fatigue failures are often related to anomalies in mechanical behavior and manufacturing defects [1]. It this paper, a mechanical analysis has been performed with the metallurgical examinations to identify the possible causes of fatigue. Finite element method was utilized to determine the steady-state stresses and dynamic characteristics of the turbine blade under both laboratory and service conditions. Finally, the steady and dynamic stresses of blade have been checked by simulating the final crack through FRANC3D, state-of-the-art crack propagation software developed at Cornell University, which employs boundary elements and linear elastic fracture mechanics.

Background

A failure is addressed to a heavy duty land base gas turbine in hot section side. As a short history of failure can be mentioned that the unit was suffering from extraordinary turbine bearing vibration for period of the time which caused the unit forced outage. After casing removal; it was observed that among whole set of first stage rotary blades; four parts were fully broken in airfoil near meanline section. According to Fig. 1, out of four blades, one of them exhibited fatigue marks on the fracture surface (which is shown by two arrows), which has been proven by fractographic investigations [2]. The fracture surface of the failed blade was examined by a scanning electron microscope (SEM). Presence of fatigue striation lines reveals that HCF was the active mechanism of failure. However, the different analysis showed that the geometry and material characteristic complied with manufacturing requirements. In order to find out the root cause of fatigue cracking, the mechanical analysis has been carried out by using full featured model geometry including complex cooling passages of the blade which the results are presented hereinafter.

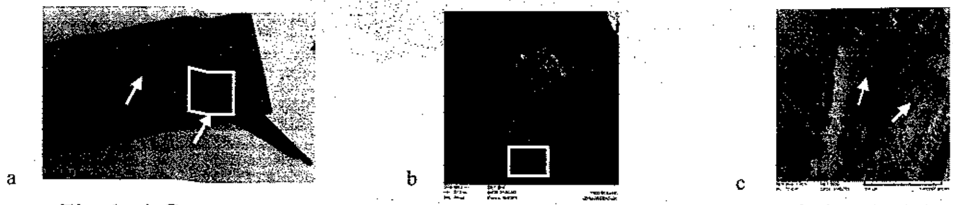


Fig. 1. a) General view b) SEM micrograph c) Higher magnification of Fig. 1b region

Finite Element Analysis

As above mentioned the metallurgical investigation showed that the blade failure was due to high cycle fatigue this means that the blade was exposed to cyclic loading. Fracture due to cyclic loading suggests that the blade can be subjected to resonance. So; modal analysis and dynamic FEM stress analysis have been conducted to extract blade natural frequencies, mode shapes and dynamic blade behavior. The FEM model is first validated by experimental results of the blade frequency test bench. The boundary condition which has been applied is clamped condition. Table 1 summarizes the results of experimental blade vibration measurements compared with the FE model predictions. The numerical analysis was performed by using commercial FEM software.

Table 1.Validation results for the FE model

Blade constraints	Mode no.	Tests [mean, Hz]	Model predication[Hz]	Mode type	Difference [%)]
clamped	1	562.2	569.4	Bending	1.2
	2	1303.3	1327.4	Torsion	1.8

A comparison of the natural frequencies is given in Table 1 depicting that there are good agreement between the test results and model prediction. In order to examine the dynamic behavior of the blade under service conditions, a pre-stressed modal analysis was performed. A FE analysis was conducted by exerting the centrifugal and gas pressure loads computed by CFD technique. The temperature dependency of relevant mechanical properties has been also considered in this analysis. The FE results were used as the initial conditions for a modal analysis. The stress distributions and mode shapes corresponding to the first and second vibration modal analysis are shown in Fig. 2. a & b.

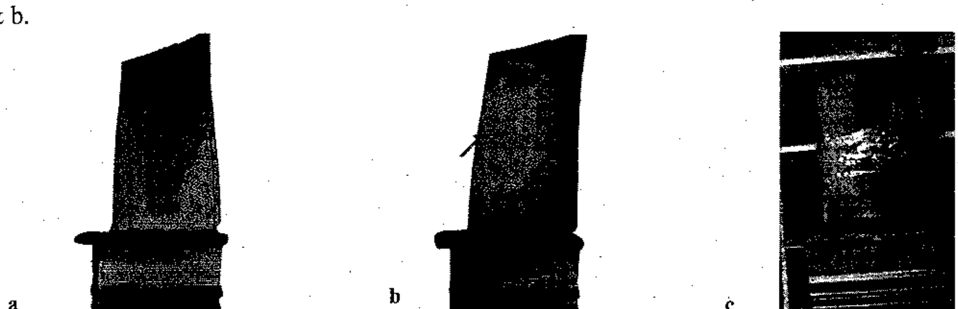


Fig. 2. Stress distribution for the blade in different vibration modes: (a) First mode-bending; (b) second mode- torsion; (c) arrows imply to initiation site of fatigue cracks in the damaged blade

In respect with the position of fatigue cracking in broken part (Fig. 2c) and comparing with stress plot of the second vibration modal shape; it can be easily recognized that the peak stress of the blade comply with where cracks initiated. Therefore; the second mode of blade vibration has probably been excited in this case of fracture. Generally; there are two possible vibration excitation sources for the blade, (a) gas dynamic fluctuations and (b) blade tip rubbing. No significant evidence of gas dynamic pulsation abnormalities were found due to nozzle passing. As a consequence; normal gas dynamic fluctuations are most unlikely to be the excitation source. On the other hand; the rubbing between blades tip and vane upper shroud was evident from the vestige of wear at the tip of the blades which would occur in the form of an impulse load (Fig. 3a). It should be mentioned that in this turbine design; blades rotate under the vane upper-shroud roof functioning as shroud segment feature. This loading has been analyzed to determine whether it could contribute to the failure. Rubbing is essentially a "stick-slip" phenomenon [3]. In the case of a resonant or near resonant operation under these harmonic forces the blade can suffer a significant reduction in the design life. In a turbine stage, tip-rubbing can create a dynamic source of excitation that is most likely to vibrate blades at their second vibration mode. In among; the blade with fatigue marks was probably experienced most extensive rubbing and then has been fractured from the section with maximum vibratory stress near mean line section. The magnitude of stress was created by such a rubbing is impossible to predict preciously unless direct measurements of the tip displacement have been made. Therefore; in order to calculate the dynamic stress of second mode excitation, a harmonic analysis was performed in the condition of different harmonic force 100, 300, 500 and 700 N when applied as a tip rubbing force. Fig. 3c illustrates the stress distribution that resulted when rubbing was simulated.

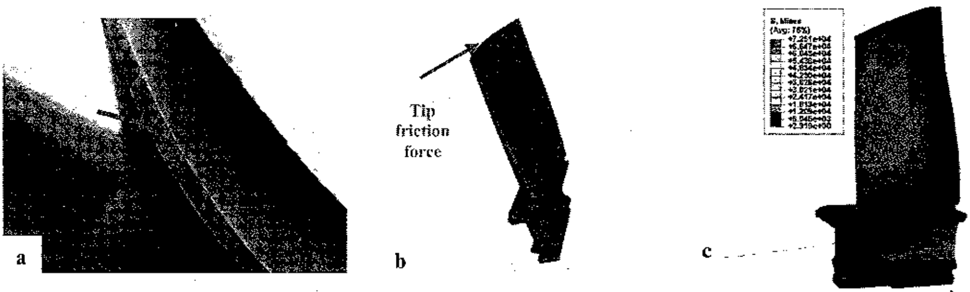


Fig 3.a) tip rubbing vestige b)Representation of tip rubbing force c) Resonant stress in 2nd mode-300 N rubbing force

Fracture Mechanics Analysis

In this step by using of FRANC3D software, the fracture mechanic behavior of a crack on the airfoil of the blade is simulated to predict whether crack initiate and propagate under already mentioned operation condition. According to the theory of facture mechanics, a certain level of vibratory energy is required to initiate a crack. The threshold stress intensity factor (ΔK_{th}) for IN738 material in gas turbine environment is approximately 8MPa$\sqrt{m}$ [4]. The values lower than threshold stress intensity factor is not providing sufficient energy to initiate and propagate a crack. For a complicated geometry and stress field produced by a complex three-dimensional structure like the airfoil of the gas turbine blade, the computer model is used to explicitly represent the damage and directly compute the ΔK along the crack front. So a boundary element model of the blade was constructed by OSM/FRANC3D [5]. In this approach; the complex geometry of the blade including sophisticated geometry of internal cooling system has been considered. Centrifugal and gas pressure loads were applied to the model as the boundary condition of the problem. The dynamic stress induced by alternative tip friction force has been calculated by FEM harmonic analysis and introduced into FRANC3D software as a cyclic stress. A semi-elliptical edge crack which is oriented approximately normal to the surface is introduced into the airfoil (Fig. 4a). The dimension is 0.5 mm length by 0.25 mm depth. The size and location of the initial flaw are taken from the fractographic analysis. Fig.5 shows the normalized K_I along the crack front under steady state operation condition and cyclic stress when different tip rubbing forces of 100, 300, 500 and 700 N are applied in the blade.

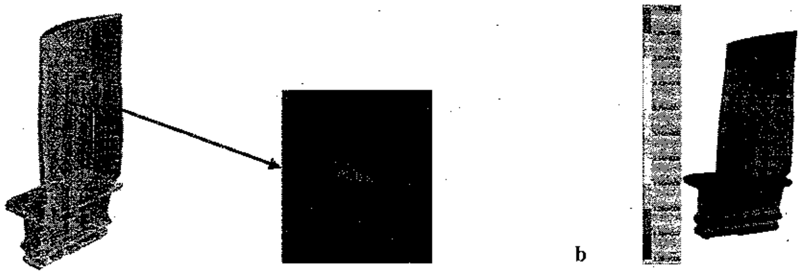
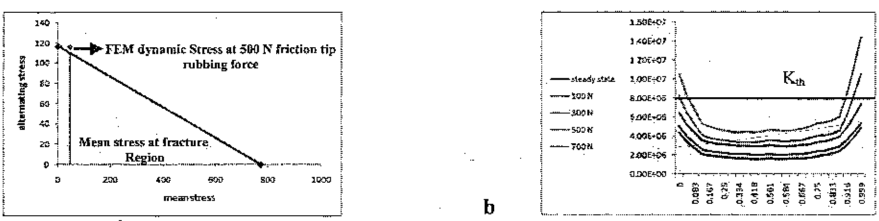
a  b

Fig 4. a) Typical model of blade in OSM. b) Superposition of centrifuge, Gas bending and 300 N harmonic forces in FRANC3D

According to Fig. 5b the calculated stress intensity factors indicate that only with the tip rubbing force more than 500 N, initiation and propagation of a fatigue crack is susceptible. This is in agreement with the result of Goodman diagram and FEM harmonic analysis (Fig. 5a). According to Goodman diagram an alternating stress of 110MPa is needed to occurrence of fatigue at the airfoil. Harmonic stress simulation by commercial FEM software shows this alternating stress is allocated in nearby the fracture region when 500 N tip rubbing friction forces is applied.

a 

b

Fig 5. a) Goodman diagram. b)K_I stress intensity factor for different condition

Summary

In order to realize the root cause of happened failure; a fracture mechanics study has been carried out. The results indicate that sever rubbing between vane outer-shroud and blade squealer probably made blade excited which led to failure by HCF mechanism. The complementary FEM analysis also showing the 2nd vibration mode of the blade has been resonated. The calculated vibratory stress depicts that excitation of the 2nd vibration mode leading to fatigue crack initiation through the meanline section where the maximum and critical stress is experienced. In this case; the critical vibratory stress is introduced when blade tip friction forces would be higher than 500 N.

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