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Fatigue life prediction in the damaged and un-damaged compressor blades

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A R T I C L E I N F O

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A B S T R A C T

A land-based gas turbine may operate at the different environment including corrosive and dusty environments. These conditions can cause early damages in the “cold” parts as well as in the “hot” parts of the gas turbine. In this research, fatigue life of a compressor blade is predicted with and without damages. Damage is considered as a notch at the blade and is classified as three types of short, medium and long notches. These are numerically simulated in the compressor blade and fatigue crack initiation life is calculated. Stress analysis is carried out using finite element analysis followed by fatigue life calculations. Furthermore, the influence of blade tip displacement on life of the undamaged and damaged blade is investigated. Moreover, the effect of damage's size and location on blade’s life is reported. This procedure can be tuned with corrosion damages observed during the minor inspection where the gas turbines are operating in the unusual environments.

1. Introduction

During the operation of a gas turbine, both turbine and compressor blades are subjected to the various loadings. These loadings can induce thermal, static and dynamic stresses in the blades. Therefore, in order to avoid unexpected failures, it is necessary to predict the blade’s life, accurately. Fatigue failure is the most common failure mechanism for compressor and turbine blades. Blade vibration can cause high cycle fatigue (HCF) and centrifugal stress due to the high rotational speed of the blade can cause low cycle fatigue (LCF). Furthermore, the damage induced by corrosion or small hard objects that remove a part of the blade material may combined with the typical load spectra experienced by blade during engine operation leading to early unexpected fatigue failures. In fact, when a part of material is removed from a component the damage is so called Foreign Object Damage (FOD). FOD can be as small as debris or as big as a 3 inch bolt. FOD generally reduce the fatigue strength of the compressor blade because of the stress concentration due to notch-like geometry defect. There are literatures available on the fatigue failure of compressor blades. For

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example, Witek et al (2009) conducted a failure analysis of compressor blade belonging to a helicopter turbo shaft engine. Similar study has been demonstrated by Tsai (2004), Kermanpour et al. (2008), Poursaeid et al.(2008), Infante et al.(2009) and Masoumi khalil abadi et al.(2011). Poursaeid et al.(2013) investigated failure of the compressor first row rotary blades, they concluded corrosion and pitting in high stress location on blade was the source of the crack initiation which was propagated via high cycle fatigue process and lead to compressor blade failure. Effect of FOD in fatigue strength has also been subject of some researches. Ruschau et al. (2001) investigated damage induced by FOD in a loading edge of the blade. Nowell et al. (2003) introduced an elastic short-crack arrest approach for estimating the fatigue life of the compressor blade under combined high- and low-cycle fatigue following FOD. Residual stress created by FOD impact is investigated by Frankel et al.(2012), Their studies focused on impact of spherical object onto flat surface and generalization of results into the blades.

In present research, we study the effect of size and location of FOD defects in high cycle fatigue life of a compressor blade. The main objective is to find a range of notches (simulation of damages) that has the minimum effect on the blade's fatigue life. Observation of such a defect during the minor inspection of the gas turbine does not force the owner to replace the blade. We considered the defect by v-notches illustrated in Fig.1.b. Three different notch sizes in three different locations are investigated.

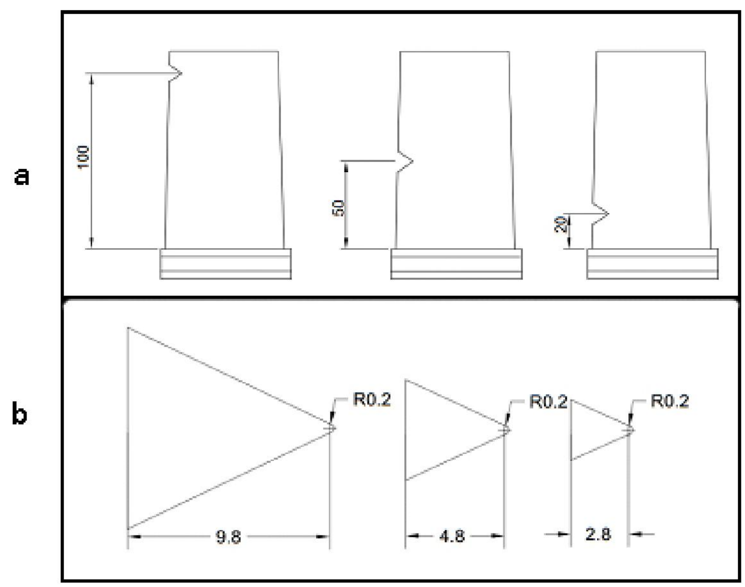

Fig. 1. a) Notch locations on blade b) notch geometries

2. Stress analysis

Stress analysis is performed on two blade types, non-defected and defected blades; the former is used for calculation of the referenced life. In both models, the blades are fixed from the bottom surface of the dovetail; the blade is then bent toward its right and left sides as shown in Fig.2. Different bending force is generated with various blade tip displacements imposed as a boundary condition. This is the simulation of the first transverse resonant condition. Material property of the blade is listed in Table1.

Table 1. Material Properties of the blade (Dowling, 1999)

Young's Modulus	Poisson Ratio	Density
113800 MPa	0.342	$4.43 \times 10^{-9}\ ton/mm^3$

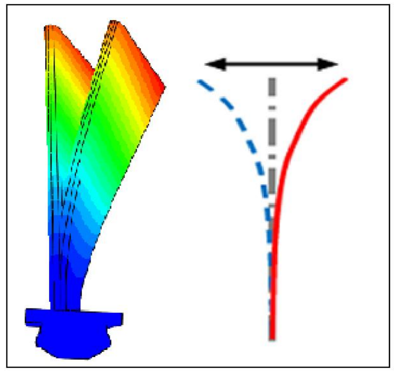

Fig. 2. Blade vibration in 1st mode idealization

The effect of FOD in the blade was considered as V-notched defects which are located in up, middle and bottom section of the blade on leading edge. The stress analysis was performed for various tip displacement for non-defected and defected blades. As shown in Fig.3 maximum Von-Misses stress in non-defected blade is located on the convex surface of the blade where the blade profile is connected to the blade dovetail. For notched blade, the value and location of the maximum stress are changed and has found to be dependent to the notch sizes and position. With all notches and notch locations considered in this research, stress concentration resulted from consideration of a notch change the maximum Von-Mises stress location, from the bottom profile to the notch location. As instant for blade with 5 mm v-notch on profile bottom section the maximum Von-Mises stress is raised 4.5 times larger in same tip displacement amplitude and also is accrued in notch tip region.

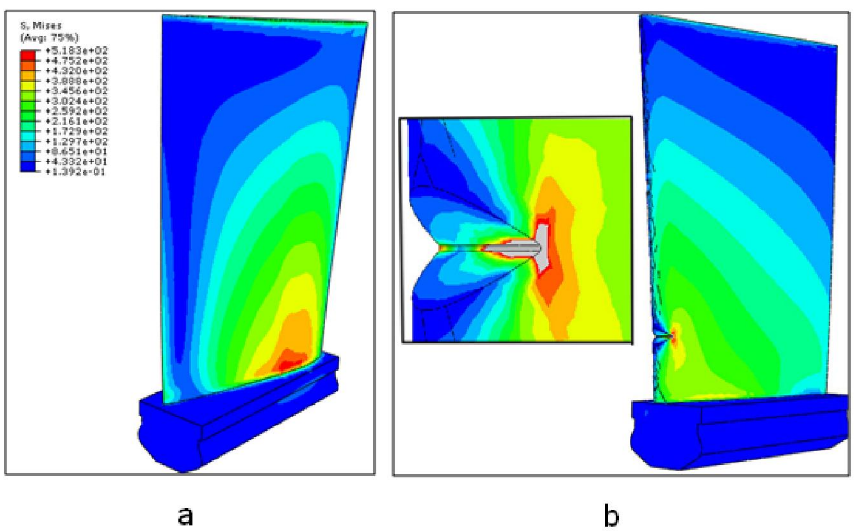

Fig. 3. Maximum stress location on blade, a) Non-defected blade, b) notched blade

Figs. 4 to 6 illustrate the variation of the blade tip displacement with maximum Von-Mises stress. Fig.4 compare maximum Von-Mises stress in non-defected blade with the blades with 3 mm, 5mm and 10 mm notches, where the notch is considered in the bottom of the profile. Figs.5 and 6 show the similar results having the notch of the middle and the tip of the blade profile, respectively. Obviously, results clearly illustrate that stress field is related to notch size and location on blades and notch in bottom section is most dangerous than other ones.

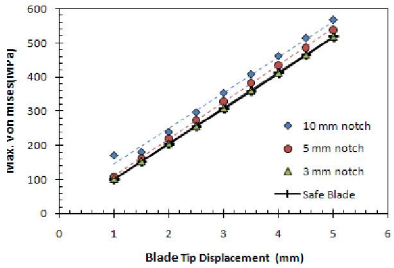

Fig. 4. Maximum Von-Mises stress in blade with notch in top section

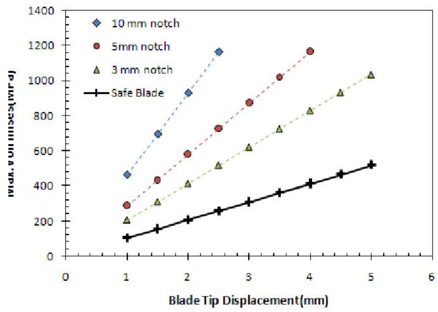

Fig. 5. Maximum Von-Mises stress in blade with notch in middle section

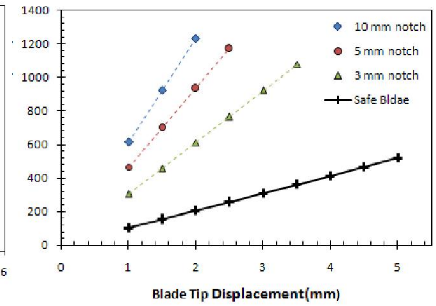

Fig. 6. Maximum Von-Mises stress in blade with notch in bottom section

3. Fatigue analysis

During the operation for the bending resonant mode, the blade is bent in the left and the right, resulting fatigue within the blade. Existence of the notches in blade raises the stresses beyond the elastic limits at the notch tip region. In order to predict the fatigue life of the blade, strain-life method is used. Neuber's rule is adopted where the local plasticity is existed. The relationship between the applied strain and fatigue life under multi-axial loading is given by Morrow equation:

$$\frac{\Delta\varepsilon_{eq}}{2} = \frac{\sigma_f' - \sigma_m}{E}(2N_f)^b + \varepsilon_f'(2N_f)^c, \tag{1}$$

where $\Delta\varepsilon_{eq}$ is equivalent strain rang, σ_f' is Basquin's fatigue strength coefficient, σ_m is local mean stress, b is Basqin's coefficient, ε_f' is fatigue ductility coefficient and c is fatigue ductility exponent.

To compute local strain and stress, Neuber's rule was used. The Neuber's rule is written as:

$$K_t = \sqrt{K_\sigma K_\varepsilon} \tag{2}$$

K_t is stress concentration factor and K_σ , K_ε is the local stress and stress concentration factor relatively. Eq. (2) can be transformed to:

$$\frac{K_t^2 \Delta S^2}{E} = \Delta\sigma . \Delta\varepsilon \tag{3}$$

Since for the given loading and geometry, the left side of Eq. (3) is known, other relationship between $\Delta\sigma$, $\Delta\varepsilon$ is considered by cyclic stress-strain relationship:

$$\Delta\varepsilon = \frac{\Delta\sigma}{E} + 2\left(\frac{\Delta\sigma}{2K'}\right)^{\frac{1}{n'}} \tag{4}$$

where $\Delta\sigma$, $\Delta\varepsilon$ can extract from Eqs. (3-4), the schematic of Neuber's rule was illustrated in Fig .7.

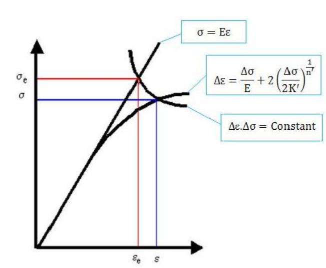

Fig. 7. Neuber Plastic Correction Diagram

The ABAQUS results for linear-elastic material was imported to fatigue analysis. In the next stage local plastic strain is extracted by Neuber's rule and finally for each discussed defected and non-defected blades crack initiation life is calculated with morrow relationship. Fatigue properties of blade used in the analysis is listed in Table 2. Fatigue analysis results have been reported in diagrams of Figs. (8-10). It is clearly obvious that 3 mm notch in top section on blade has negligible effect in the blade life and it may be chosen as a criterion for industrial applications.

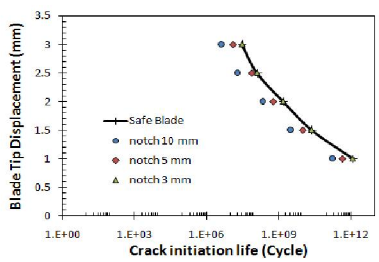

Fig. 8. Crack initiation life for top section notched blade

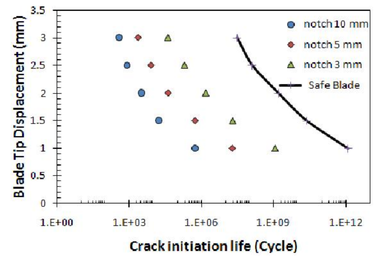

Fig. 9. Crack initiation life for middle section notched blade

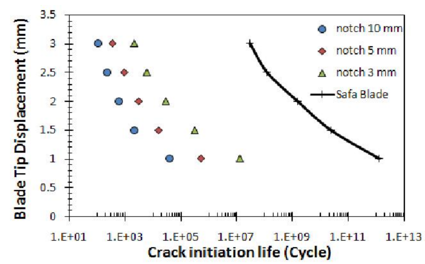

Fig. 10. Crack initiation life for bottom section notched blade

As shown in Fig.11, first crack initiation region is located in convex surface of the blade where the profile is connected to dovetail for non-defected blade whereas for notched blade first cracks initiate in notch tip region on suction side of profile

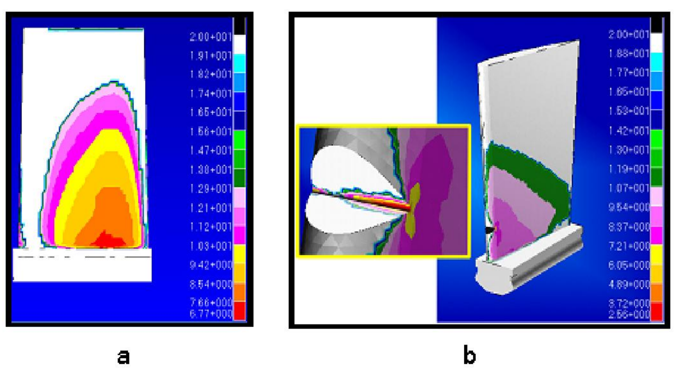

Fig. 11. First crack initiation region a) Non-defected blade b) Notched blade

Table 2. Fatigue properties of Ti-6Al-4V

Fatigue properties	Symbol	amount
Cyclic strength coefficient (MPa)	K'	1772
Cyclic strain-hardening exponent	n'	0.106
Fatigue strength coefficient (MPa)	σ_f'	2030
Fatigue ductility coefficient	ε_f'	0.841
Fatigue strength exponent	b	-0.104
Fatigue ductility exponent	c	-0.688

Effect of the notch location in blade fatigue life was reported in Figs. (12-14). As showed in these diagrams, even 3 mm notch in bottom section of the blade strongly reduced the blade life. If 5 mm notch locates in bottom section, the blade operate 16468 cycles under vibration condition whereas 5.44×10^5 cycles in middle section and 9.142×10^9 cycle in top section notch, when blade tip displacement being 1.5 mm. Fig.15 show twin notch size and location effect on blade fatigue life

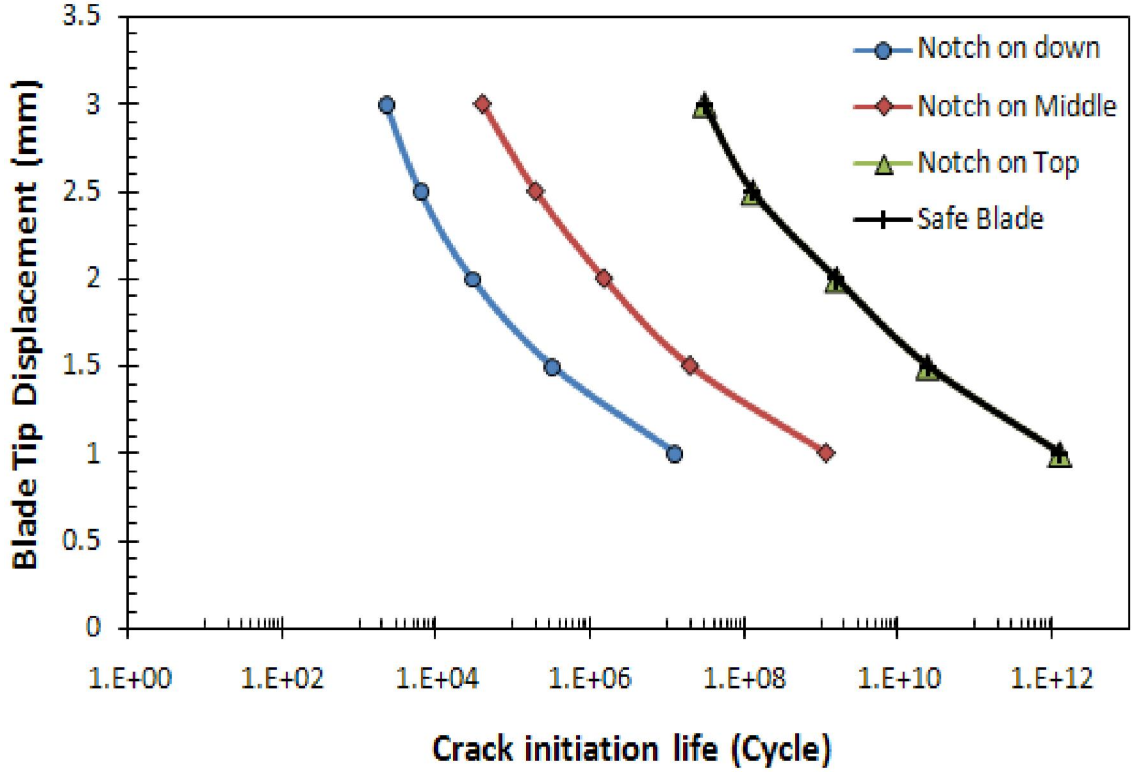

Fig. 12. Crack initiation life in 3 mm notched blad

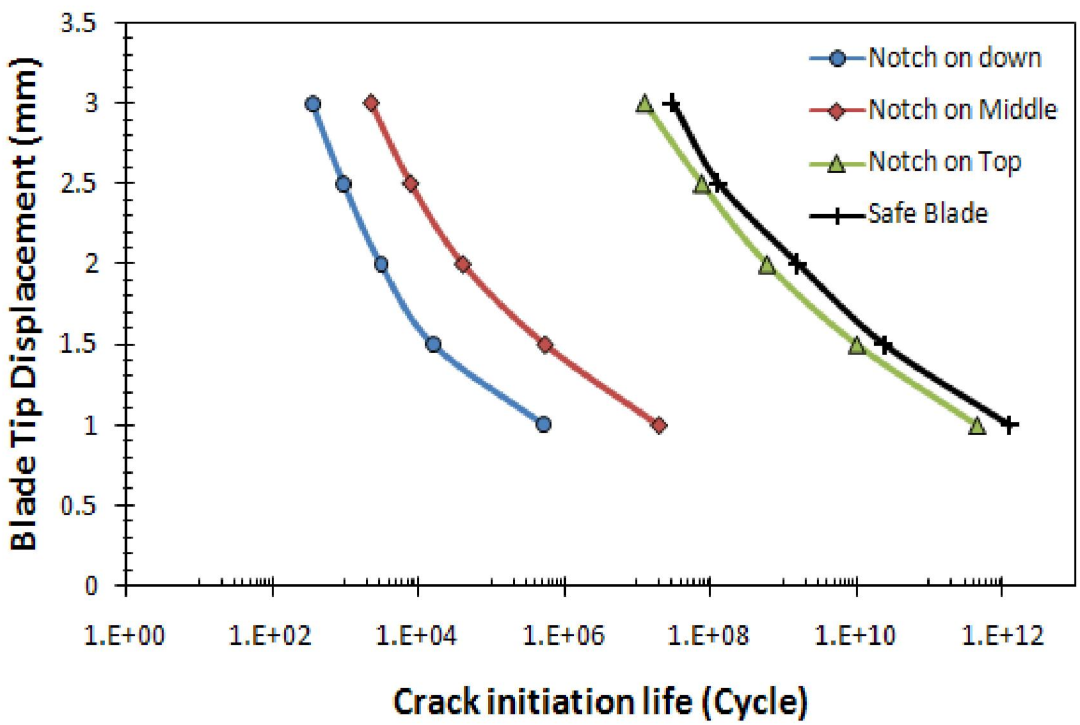

Fig. 13. Crack initiation life in 5 mm notched blade

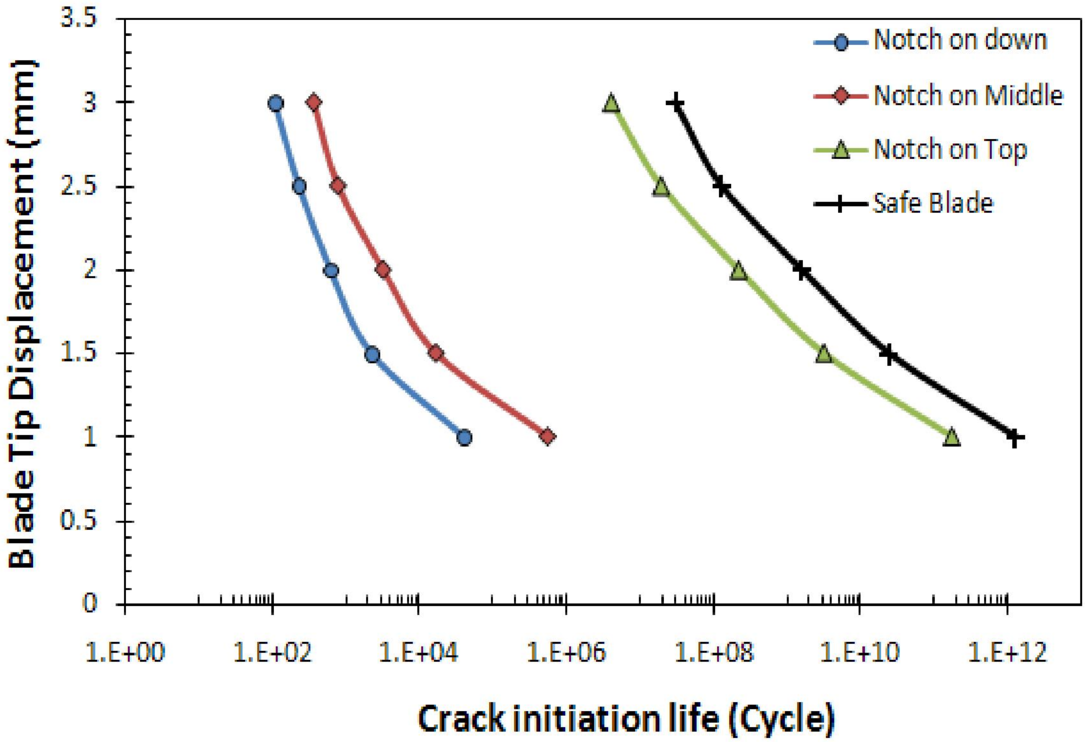

Fig. 14. Crack initiation life in 10 mm notched blade

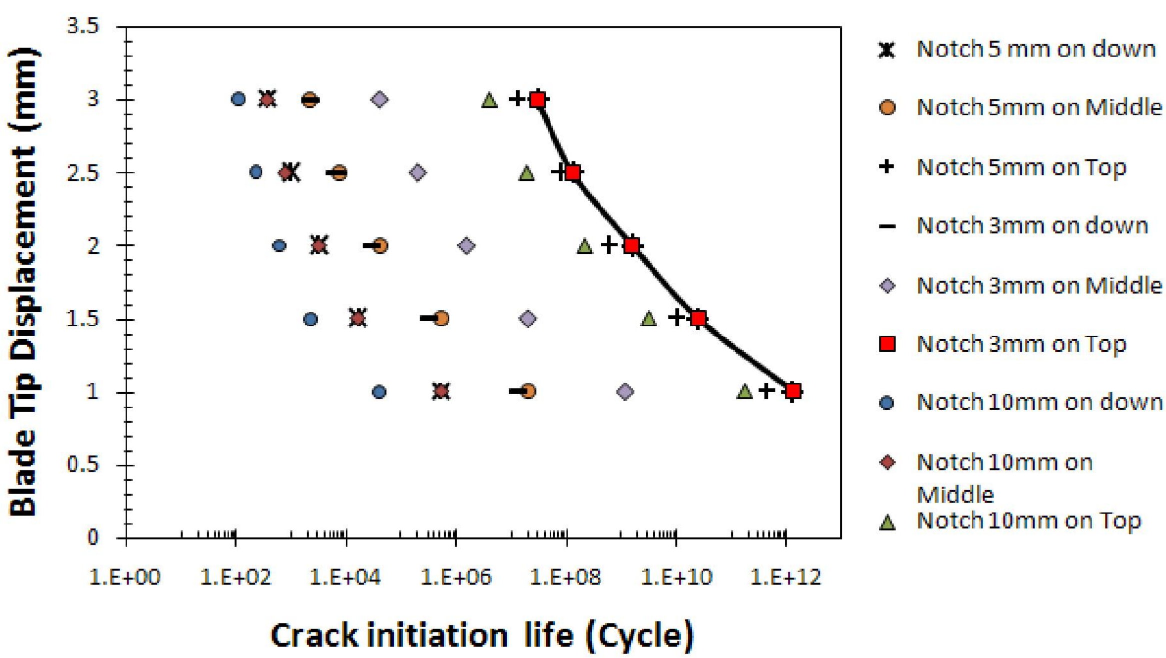

Fig. 15. Total fatigue results with different notches

4. Conclusions

In this study, the first mode resonant condition was considered and numerical analysis was performed to investigate the effect of notch size and location in blade fatigue life reduction based on the results following conclusion can reported:

1) The maximum stress in the non-defected blades is located in profile and dovetail connection region whereas maximum stress in notched blade is accrued in notch tip region in suction side of profile.
2) The larger the notches size the lower the fatigue life.
3) Results of $\varepsilon - \mathrm{N}$ analyis showed that the fatigue life strongly depends to notch location. Notch in lower section of blades profile is more dangerous than the other region.
4) Small notch in top section of blade profile had not significant effect on blade fatigue life and can be neglect

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