Crack propagation analysis of compressor blade subjected to resonant vibrations

Lucjan Witek, Arkadiusz Bednarz, Feliks Stachowicz, Nikita Kazarinov, Ivan Smirnov
lwitek@prz.edu.pl, i.v.smirnov@spbu.ru

Abstract

In this work the crack propagation analysis of the compressor blade of aero engine was performed. During investigations the blade with mechanical defect (notch) was considered. In experimental analysis the blade was subjected to resonant vibration. During transverse vibrations, a high stress occurs in the blade. Pulsation of stress causes the fatigue of material. In results of proposed investigations both the number of load cycles to initiation and also the crack growth dynamics was obtained for the blade working in resonance condition. In second part of work the maximum principal stress distributions in the vibrated blade were determined using finite element method.

1 Introduction

High-cycle fatigue (HCF) is often concerned with vibration of aero engine components. Compressor blades have a small bending stiffness and are particularly susceptible to HCF. During work of engine, blades are excited by an unbalanced rotor. The worst case is when the frequency of excitation overlaps with the resonant frequency of the blade. During resonance, large amplitude of stress causes that the blade can be damaged in relatively short time. The fatigue process is often accelerated by mechanical defects (notches) created during collision of rotated blade with hard objects suctioned from a ground. If aproblem arises in the compressor section it will significantly affect the whole engine function and safety of the aircraft.

The broken blade could cause the puncture of the engine casing. Failures of any high speed rotating components (jet engine rotors, centrifuges, high speed fans, etc.) can be very dangerous to passengers, personnel and surrounding equipment and must always be avoided. The failure analysis of the compressor blade has received the attention of several investigations. The problem of fatigue fracture of the aero engine blades was described in works [1-10].

The objective of presented investigation is to determine both the number of load cycles to crack initiation and also the crack growth dynamic for the compressor blade of aero engine (including artificially created mechanical defects), subjected to resonant vibrations. Created defects (notches) simulate the foreign object damage (FOD) of the blade. An additional aim of work is numerical determination of maximum principal stress values in the blade with the notch subjected to resonant vibration.

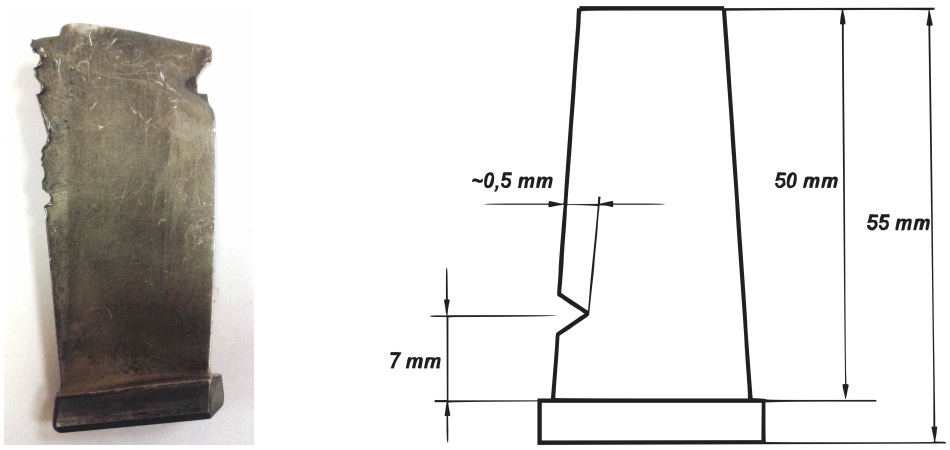

Figure 1: View of blade damaged by foreign object (a), dimension of the investigated blade with v-noth (b).

2 Experimental investigations

In investigated blade a V-notch presented in Fig. 2 was created. The depth of notch was about $0.5mm$ whereas the apex angle 90 degrees. The notch in the blade was created by machining. The compressor blade was made out of $EI-961$ steel ($0.11C$; $11Cr$; $1.5Ni$, $1.6W$; $0.18V$; $0.35Mo$; $0.025S$; $0.03P$) with the following properties (measured in temperature 20^oC): Ultimate tensile strength $900-1000MPa$, Yield stress $800-900MPa$, Young modulus $200GPa$, Poisson ratio 0.3. The high cycle fatigue tests of the blade were made using the Unholtz-Dickie UDCO TA-250 electrodynamic vibration system, presented in Fig. 3 at Laboratory of Turbomachinery of Rzeszow University of Technology. The blade with a notch was horizontally mounted on the movable head of vibrator (Fig. 4). Next the head of shaker was entered into harmonic vibration. In first step of analysis the resonance frequency was determined (for first mode of transverse vibration). The fatigue test was started from frequency close to resonant. During investigations two main parameters were periodically monitored: vibration amplitude of the blade tip and the size (or existence) of the crack. For control of amplitude the laser scanning vibrometer POLYTEC PSV H-400S were used. To measure the length of the crack a nondestructive fluorescent penetrant method was utilized.

The control parameters of vibration system and results obtained for compressor blade are shown in Tab. 1. The resonant frequency (F_{rez}) of blade was $796.6Hz$. As seen from Tab. 1, the fatigue test started from frequency $2.2Hz$ higher than F_{rez} ($798.8Hz$). Just for this frequency, the vibration amplitude $A = 1.2mm$ was achieved. After 12.46×10^6 total number of cycles (N), an amplitude of blade tip decreased from $1.20mm$ to $1.08mm$. During fracture, the bending stiffness of blade is not constant. This information is important from practical point of view, because decrease of amplitude at constant intensity of excitation is always related to start of crack initiation process. In present case $2.5mm$ long crack (a dimension in Fig. 8) was detected. From $N = 12.82 \times 10^6$ number of load cycles, the excitation frequency decreased with different rate. Preliminary, the rate of change of frequency was $0.025Hz/s$. It allowed to maintain the vibration amplitude on constant level

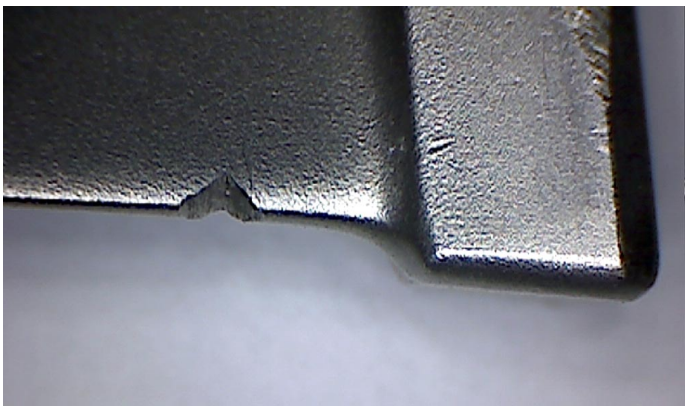

Figure 2: V-noth created on the attack edge of investigated blade.

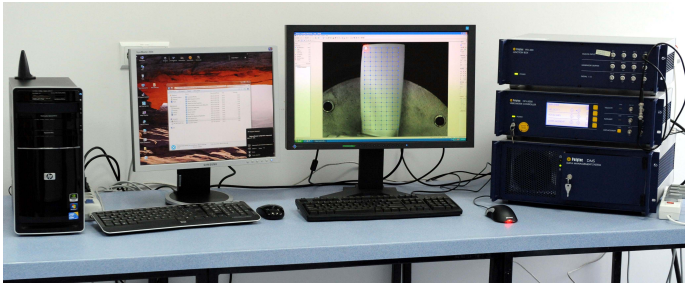

Figure 3: View of control systems of both laser scanning vibrometer and the shaker used in experimental investigations.

(about $1.2mm$). In first stage of blade fracture the intensity of acceleration of vibrator head was constant to crack length $a = 6.5mm$. After that the intensity of head acceleration was increased to $12g$ and $14g$ adequately. In spite of increase of acceleration, the blade amplitude was not constant in the final stage of fracture (for crack length $a = 6.5 - 19mm$).

Obtained results (Tab. 1) showed that the blade with v-notch created by machining, needs $N = 12 \times 10^6$ total number of load cycles to crack initiation. The crack propagation process was much shorter. The crack needs $N = 1.87 \times 10^6$ number of load cycles for propagation from length $a = 0$ to final crack size a = 19 mm (at which the blade was broken). Thus, in presented case the crack initiation process ($N = 12 \times 10^6$) is a main part of fatigue life of the blade ($N = 13.87 \times 10^6$).

The assumption of work was to maintain the blade tip displacement amplitude (vibration amplitude) on constant level. However this condition is difficult for satisfy during all fatigue test. The vibration amplitude in the blade was constant until about $N = 12 \times 10^6$ number of cycles (Fig. 6). Just after crack initiation, the blade stiffness decreases and in consequence of them the lower value of blade amplitude ($A = 1.08mm$) was observed (at $N = 12.46 \times 10^6$). To maintain the blade amplitude on level $A = 1.2mm$ the frequency of excitation was next decreased. In the range of $N = 12.82 - 13.10 \times 10^6$ number of load cycles, the blade amplitude was close to initial value, but after $N = 13.42 \times 10^6$ the vibration amplitude decreased more quickly. The last part of fracture is highly unstable process. The increase of intensity of vibration (to value of $12g$ and $14g$) in finish part of fatigue (Fig. 7) caused that

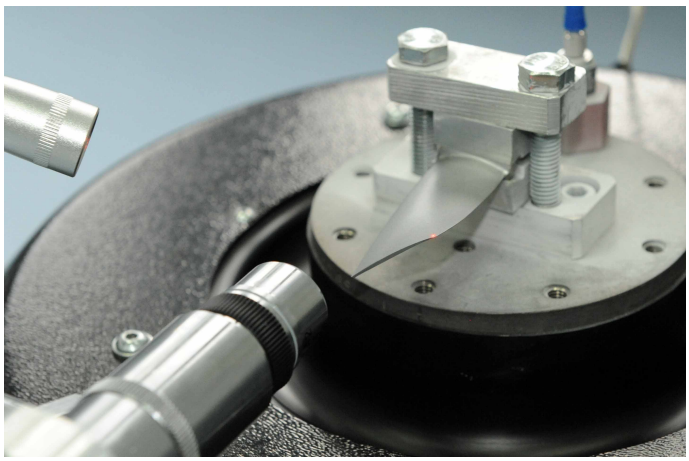

Figure 4: Compressor blade fixed to movable head of vibrator.

Table 1: Control parameters of vibration system and results of fatigue test of the blade.

Initial freq.	Final freq.	Rate of change of freq.	Intensity of excitation	Partial no. of cycles	Total no. of cycles	Total no. of cycles (crack prop.)	Crack length	Amplitude of crack tip
F_{init} [Hz]	F_{fin} [Hz]	dF/dt [Hz/s]	[g]	N_{part} $\times 10^6$	N $\times 10^6$	N_{cp} $\times 10^6$	a [mm]	A [mm]
798.8	798.8	0	10	0	0	-	0	1.20
798.8	798.8	0	10	3	3	-	0	1.20
798.8	798.8	0	10	3	6	-	0	1.20
798.8	798.8	0	10	3	9	-	0	1.20
798.8	798.8	0	10	3	12	0	0	1.19
798.8	798.8	0	10	0.46	12.46	0.46	2.5	1.08
798.8	789.3	0.025	10	0.36	12.82	0.82	4.0	1.21
789.3	770.0	0.036	10	0.40	13.22	1.22	6.5	1.14
770.0	699.0	0.260	12	0.20	13.42	1.42	9.0	1.03
699.0	500.0	0.370	14	0.32	13.74	1.74	15.0	0.95
500.0	100.0	0.130	14	0.13	13.87	1.87	19.0	0.91

the vibration amplitude was still not constant (Fig. 6).

Shape of crack in preliminary phase of growth is presented in Fig. 8a. As seen from this figure, the crack in first phase of growth propagates more quickly along the concave surface of the blade profile. In Fig. 8a is also distinguished the crack length (a dimension) used to description of vertical axis of plot presented in Fig. 5. The blade after finish of the fatigue test is visible in Fig. 8b. The crack direction is not parallel to blade lock. The crack starts from the notch located 7mm above the lock. The crack in finish part of fracture achieved the trailing edge of the blade, about 5mm above the lock.

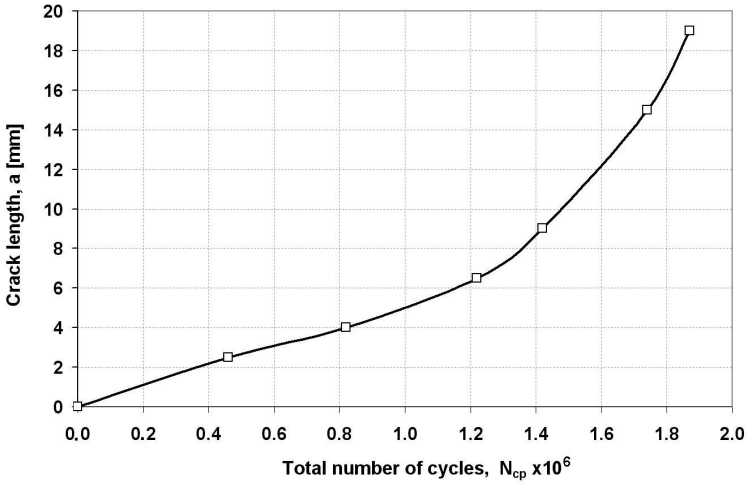

Figure 5: The crack length in function of number of load cycles N_{cp} (counted from crack initiation) for blade subjected to resonant vibration (first mode, $A = 1.2mm$).

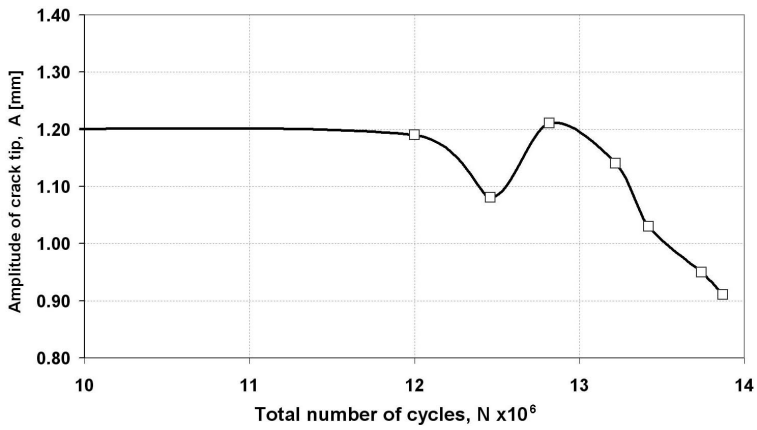

Figure 6: Amplitude of crack tip displacement (vibration amplitude) as a function of number of load cycles N.

3 Numerical stress analysis of the compressor blade subjected to vibration

For definition of stress state in the blade subjected to HCF, the finite element analysis (FEA) was performed. In this analysis the first mode of transverse vibration was considered. To solve this problem, the Patran program was used to both geometrical and the finite element model preparation. In Fig. 9a the discrete model of blade with the notch located $7mm$ above the lock was shown. In the notch vicinity the finite element mesh was concentrated (Fig. 9b). In the next part of work Abaqus software were used for stress and modal analysis of the compressor blade. Results of FEM analysis (Fig. 9c) showed that during first mode of resonant vibration the blade are subjected to cyclic bending. During transverse vibration the maximum value of amplitude of displacement (on blade tip) is equal to $1.2mm$. All numerical results are obtained for the same vibration amplitude ($A = 1.2mm$) and for left

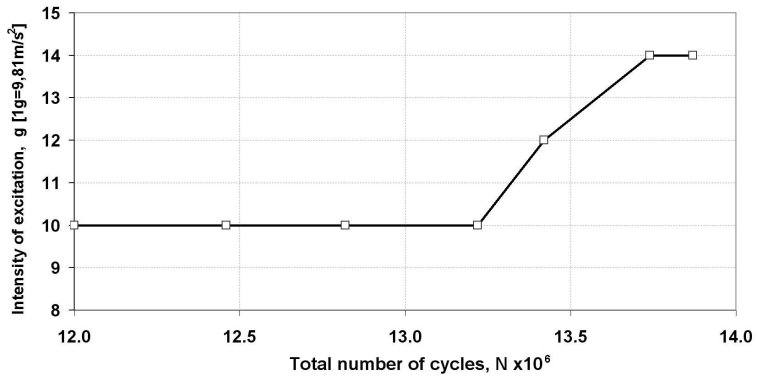

Figure 7: Intensity of excitation (vibration) in function of number of load cycles N.

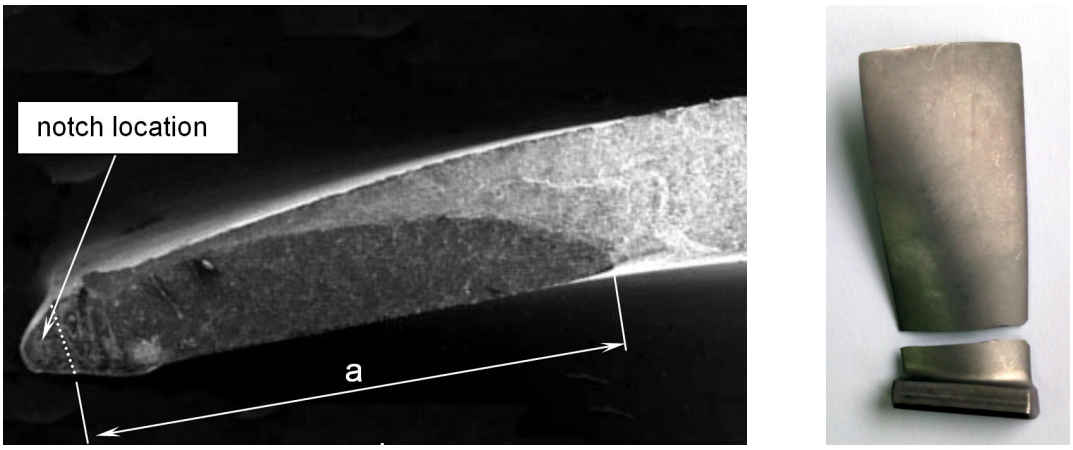

Figure 8: Fracture of blade with $6.5mm$ long crack ($a = 6.5mm$) (a) and the blade after finish of fatigue test (b).

blade deflection at which the maximum principal stress in the blade was observed. Figure 10a showed that value of maximum principal stress value in the zone located near the attack edge of blade is about $225 - 280MPa$. The area of maximum stress ($771MPa$) is located in the notch vicinity (Fig. 10b). Maximum principal stress values in cross-section of blade (in fracture plane) showed that during left blade deflection the tension stress occurred in the zone near concave surface of blade (Fig. 11). Just in this region the crack propagate more quickly than in convex profile area. Obtained results showed that cyclic tension stress in blade cross section is a main reason for crack initiation and crack propagation of the blade subjected to resonant vibration.

4 Conclusions

In this study the experimental analysis were performed to investigate both the crack initiation and the crack propagation process of compressor blade with preliminary defect. This mechanical defect simulates the foreign object damage. The complex

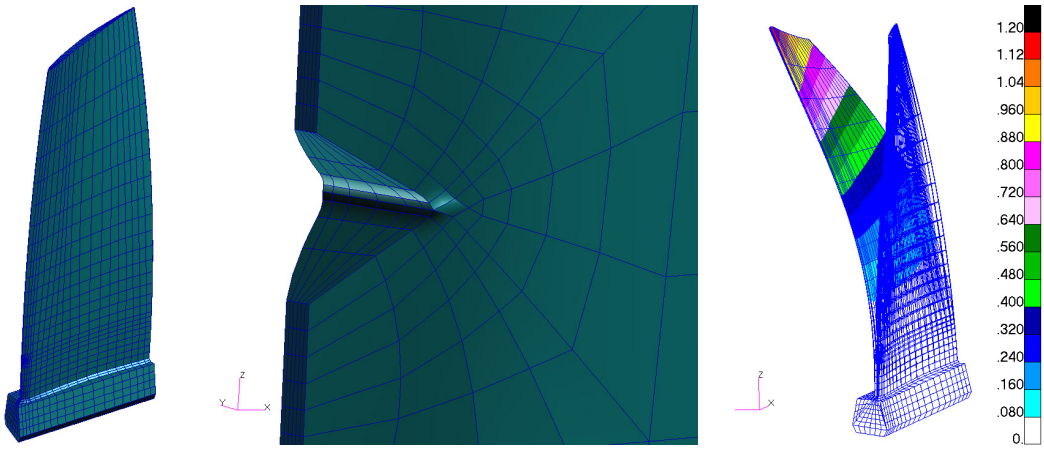

Figure 9: View of numerical model of compressor blade (a), magnified notch area (b) and values of displacement of blade during first mode of free vibrations, [mm] (c).

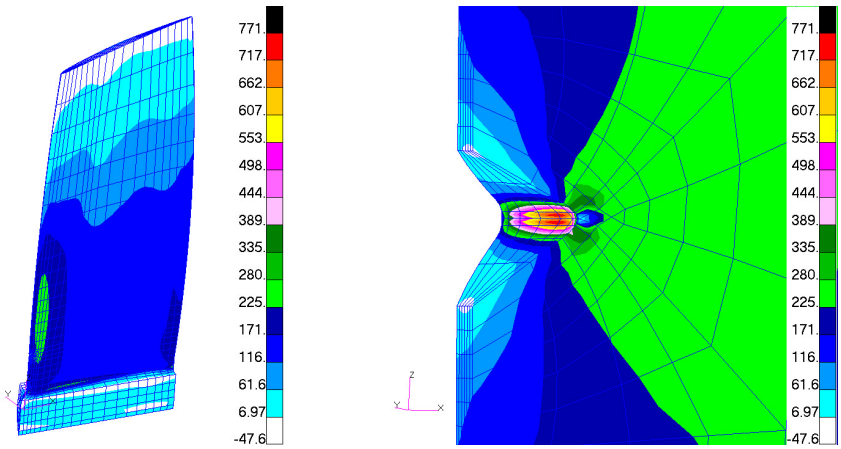

Figure 10: Values of maximum principal stress for the blade (a) and in the vicinity of notch (b), [MPa].

experiment was performed in resonance condition. In experimental investigation a modern vibration system and the laser scanning vibrometer were used. In results of performed work, the following conclusions were formulated:

1. Foreign object damage is very dangerous for the compressor blades. In most cases defects obtained in results of FOD (as V-notches) is potential crack origin. After phase of initiation, the crack propagates from notch inside the structure in relatively short time.

2. The crack in the blade working in resonance conditions (first mode of vibrations, $A = 1.2mm$) initiates after about $N = 12 \times 10^6$ total number of load cycles.

3. The crack needs $N = 1.87 \times 10^6$ number of load cycles for propagation from length $a = 0$ to final crack size $a = 19mm$ (at which the blade was broken).

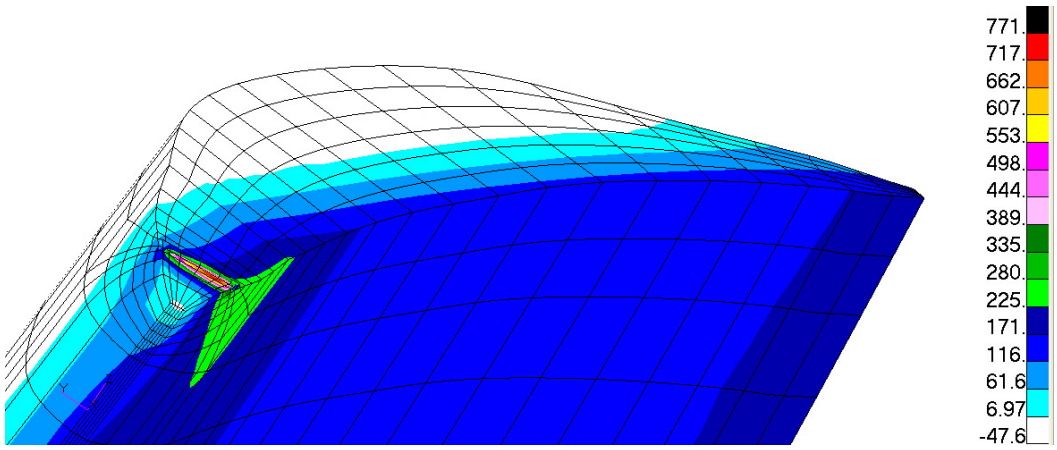

Figure 11: Values of maximum principal stress in cross-section of blade (at level of notch), [MPa].

4. The crack initiation process (number of cycles for initiate of crack from $a = 0$ to blade damage) is a small part (about 13.5%) of total fatigue life of blade ($N = 13.87 \times 10^6$).

5. Maximum principal stress area in the blade is located on tip of notch. In the blade vibrated with amplitude 1.2mm a maximum stress on the notch has a value of 771MPa. This value is close to yield stress of blade material.

6. Maximum principal stress value (for left deflection) in the blade without defects is about 3 times lower then the local stress in the notch.

In the case of old aircraft structures, which are operated according to the damage tolerance method, the information about crack dynamics is very important from practical point of view. In aerospace engineering, structure is considered to be damage tolerant if implemented maintenance program can stop operation of structure with a small (safe) fatigue crack. The operation of structures according to damage tolerance methodology can cause a significant reduction of costs because the aircraft or aero-engine can be safety operated to the real fatigue limit.

Acknowledgements

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Lucjan Witek, Arkadiusz Bednarz, Feliks Stachowicz,
Faculty of Mechanical Engineering and Aeronautics, Rzeszow University of Technology Al. Powstancow Warszawy 8, 35-959 Rzeszow, Poland

Nikita Kazarinov, Ivan Smirnov,
Department of Elasticity Theory, Saint Petersburg State University, Universitetskiy prospekt 28, Peterhof, St. Petersburg, 198504, Russia