# NUMERICAL VIBRATION ANALYSIS OF TURBINE ENGINE COMPRESSOR BLADES DEPENDING ON GEOMETRY AND POSITION OF THE DAMAGE

**Paweł Rygiel, Wojciech Obrocki, Jan Sieniawski**

S u m m a r y

Paper presents the description of the Finite Element Method used to determine the first frequency shape of the Turbine Engine compressor blade. Numerical calculations were conducted on blades with damages modelled on the CAD (Computer Aided Design) model at different positions and different depth. Reduced stress and frequency of the damaged and undamaged blades are the results of presented work.

**Keywords:** compressor blade, foreign object damage, FOD, Finite Element Method, FEM, ANSYS

**Analiza numeryczna drgań łopatki sprężarki silnika lotniczego w zależności od geometrii i położenia jej uszkodzenia**

S t r e s z c z e n i e

W pracy stosowano metodę elementów skończonych do określenia częstości drgań własnych łopatek roboczych sprężarki silnika lotniczego. Obliczenia numeryczne wykonano z zastosowaniem modeli CAD. Przyjęto uszkodzenia na wybranych wysokościach krawędzi łopatki dla różnych typów uszkodzeń. Uzyskano zestawienie naprężeń zredukowanych oraz częstości drgań własnych pierwszej postaci łopatek uszkodzonych oraz nieuszkodzonych.

**Słowa kluczowe:** łopatki sprężarki, uszkodzenia łopatek spowodowane ciałem obcym, FOD, metoda elementów skończonych, MES, ANSYS

## 1. Introduction

Foreign object damage (FOD) is what it sounds like – some object got into the turbine engine compressor. Under normal conditions the compressor rotates at tens of thousands RPM. At such speed an object impacting a blade causes sharp housing penetration, then get through the impeller until the object is either

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completely pulverized or the compressor blades are rolled enough to allow the object to pass. Once first contact occurs, complete engine destruction can happen within seconds (Fig. 1) [1, 2].



Fig. 1. View of the compressor rotor with damaged blades (FOD)

Hard objects, such as bolts or rocks, can completely destroy blades. Abrasive matter, such as sand or dirt, will wear-away the coating form blades. Soft material, such as shop rags, will roll the compressor blades backwards. Every blade damage causes unbalance of the rotating assembly, which leads to shaft motion, efficiency loss and noisy operation. When foreign objects impact the rotor, loads can be extreme. The loads can be high enough to result in catastrophic failure. Foreign object damage distributes damages relatively even to all compressor blade edges, evenly to all turbine blade tips [3, 4].

To better understand any structural vibration problem, the resonant frequencies of a structure need to be identified and quantified. Nowadays, modal analysis has become a widespread means of finding the vibration modes of a machine or structure like a turbine engine compressor blade (Fig. 2). Structural dynamics testing of every developed product is used to assess its real dynamic behavior [5].

Fig. 2. Modal analysis of a blade

## 2. The modal analysis in Finite Element Method

### 2.1. The „Modal" Model

Modes are inherent properties of the structure, and are determined by the material properties (mass, damping, and stiffness), and boundary conditions of the structure. Each mode is defined by a natural (modal or resonant) frequency, modal damping, and mode shape (i.e. the so-called „modal parameters"). If either the material properties or the boundary conditions of the structure change, its modes will change as well. For instance, if mass is added the structure vibration will change. To clarify phenomenon, the concept of single-degree-of-freedom system will be used [6].

### 2.2. Single Degree of Freedom

A single-degree-of-freedom (SDOF) system (Fig. 3 where the mass m can only move along the vertical x-axis) is described by the following equation

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = f(t) \tag{1}$$

with m the mass, $c$ the damping coefficient, and $k$ the stiffness. This equation states that the sum of all forces acting on the mass m should be equal to zero with $f(t)$ an externally applied force – $m\ddot{x}(t)$ the inertial force – $c\dot{x}(t)$ the (viscous) damping force – $kx(t)$ and the restoring force. The variable $x(t)$ stands for
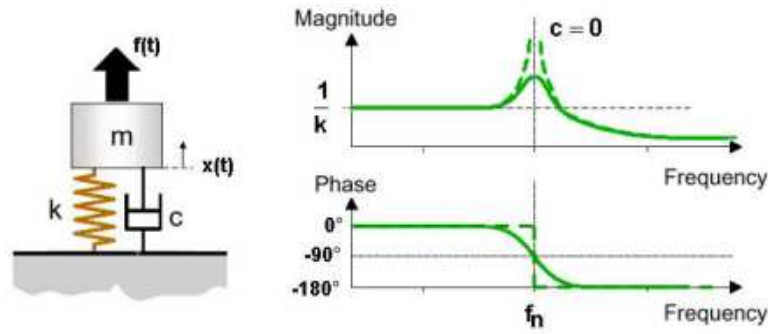
Fig. 3. SDOF system

the position of the mass m with respect to its equilibrium point, i.e. the position of the mass when $f(t) \equiv 0$. Transforming (1) to the Laplace domain (assuming zero initial conditions) yields

$$Z(s)X(s) = F(s) \qquad (2)$$

with

$$Z(s) = ms^2 + cs + k \qquad (3)$$

The transfer function H(s) between displacement and force, $(s) = H(s)F(s)$, equals the inverse of the dynamic stiffness

$$H(s) = \frac{1}{ms^2 + cs + k} \qquad (4)$$

The denominator roots of the transfer function, i.e. $\mathrm{d}(s) = $ $ = m \cdot \mathrm{d}(s) = ms^2 + cs + k$, are the poles of the system. In mechanical structures, the damping coefficient $c$ is usually very small resulting in a complex conjugate pole pair

$$\lambda = -\sigma \pm i\omega_d \qquad (5)$$

with $f_d = \dfrac{\omega_d}{2\pi}$ – the damped natural frequency, $f_n = \dfrac{\omega_n}{2\pi}$ – the (undamped) natural frequency where $\omega_n = \sqrt{\dfrac{k}{m}} = |\lambda|$, and $\zeta = \dfrac{c}{2m\omega_n} = \dfrac{\sigma}{|\lambda|}$ the damping ratio ($f_d = f_n\sqrt{1-\zeta^2}$).

If, for instance, a mass $\Delta m$ is added to the original mass m of the structure, its natural frequency decreases to $\omega_n = \sqrt{\dfrac{k}{m+\Delta m}}$. If $c = 0$, the system is not damped and the poles becomes purely imaginary, $\lambda = \pm i\omega_n$. The Frequency Response Function (FRF), denoted by $H(\omega)$, is obtain by replacing the Laplace variable $s$ in (4) by $i\omega$ resulting in

$$H(\omega) = \frac{1}{m\omega^2 + ic\omega + k} = \frac{1}{(k - m\omega^2) + ic\omega} \tag{5}$$

Clearly, if $c = 0$, then $H(\omega)$ goes to infinity for $\rightarrow \omega_n = \sqrt{\dfrac{k}{m}}$.

Although very few practical structures could realistically be modeled by a single-degree-of-freedom (SDOF) system, the properties of such a system are important because those of a more complex multiple-degree-of-freedom (MDOF) system can always be represented as the linear superposition of a number of SDOF characteristics (when the system is linear time-invariant).

ANSYS Mechanical APDL and Mechanical (Workbench) can perform harmonic analysis on a structure, determining the steady-state sinusoidal response to sinusoidal varying loads all acting at a specified frequency. Some load types can be applied with a phase offset.

Many structures are tested per customer specification with base accelerations swept across a range of frequencies. Such a test can be implied by fixing a base in ANSYS, and applying an acceleration load to the whole model with an ACEL command. The resulting movement at a point on the FEA model is relative to a fixed non-moving base, and is not what would be picked up at the point with an accelerometer. To replicate acceleration movements at a base would require an acceleration input at selected geometry, which ANSYS does not directly support in Harmonic analysis [7].

### 2.3. Dynamic finite element analysis of the blade

Dynamic finite element analysis of the blade mainly refers to the vibration modal analysis using the finite element theory. Modal analysis is used for natural

frequencies identification. The modal analysis can indicate the frequency range in which blade is the most sensitive to vibration [8].

The paper presents modal analysis of the turbine engine compressor blade to verify whether the blade mechanical properties meet its requirements. The blade finite model was established in ANSYS program by importing a model, previously created in CAD program.

Safety critical components such as compressor blades are rigorously tested to prove their ability to withstand all safety regulations. Model dynamic behavior validation requires experimentally obtained estimates of the modal parameters are required.

## 3. Numerical calculations

### 3.1. Compressor blade

Coupled modal and harmonic analysis were performed to simulate the first vibration mode of the compressor blade with no damages and two types of damages – ,,V'' and ,,I'' shape (Fig. 4).
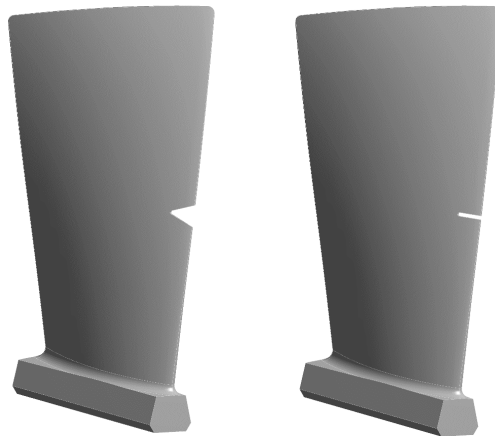


Fig. 4. Compressor blade with the ,,V'' and ,,I'' shape damage

The blade were divided into five regions – ,,A'' to ,,E'' (Fig. 5). Damages were performed in each region on a leading edge at three different levels (bottom, middle, top) and five different depths (0,1; 0,499; 0,999; 1,999; 3,2 mm).

As a result, the values of the reduced stress and frequencies of first vibration mode were obtained. Results allowed precisely planning the fatigue strength tests and eliminating blades with damages which would bring no relevant information.

The procedure for solving the problem [9]:
* Create the geometry.
* Mesh the domain.
* Set the material properties and boundary conditions.
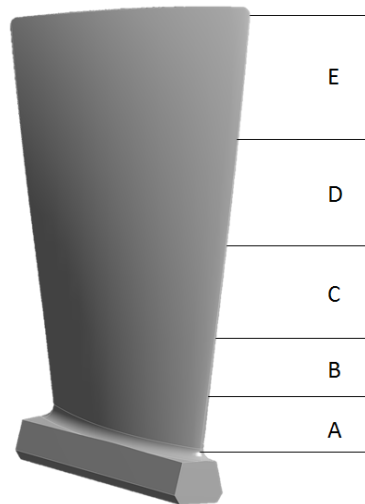* Obtaining the solution

Fig. 5. Areas distribution on the blade

## 3.2. Create the geometry – preprocessing

Preprocessing include CAD model, meshing and defining boundary conditions.

To create the geometry NX Siemens and SpaceClaim (ANSYS) were used. SpaceClaim, the CAD module in ANSYS program aids to simplify the model geometry by dividing, joining or deleting surfaces or edges in order to facilitate the further mesh creation [10].

Figure 6 shows the CAD model of the non-cracked compressor blade.

## 3.3. Mesh the domain – preprocessing

To build the discrete model it was used a tetrahedron elements were used – a higher order 3-D, 10-node element (Fig. 7). The element is defined by 10 nodes having three degrees of freedom at each node [11].

Figure 8 shows the FEM model of the blade with no damage included. The model was built with 205 236 nodes and 130 369 elements.
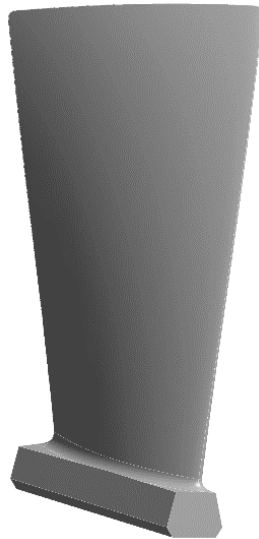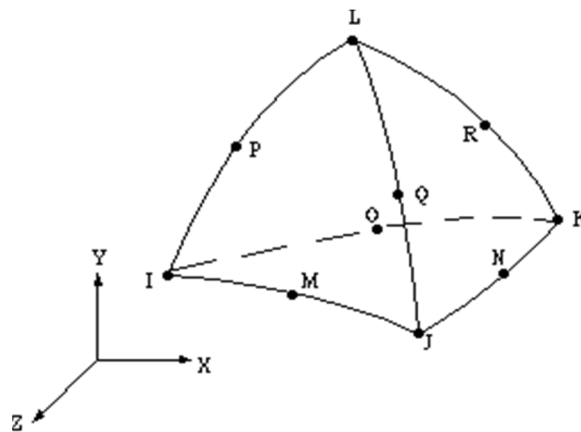
Fig. 6. Compressor blade



Fig. 7. Tetrahedral element – SOLID187

Model with "V" shape damage included counts 205 100 nodes and 130 379 elements. Mesh around the damage was applied with smaller elements to achieve more accurate results (Fig. 9).
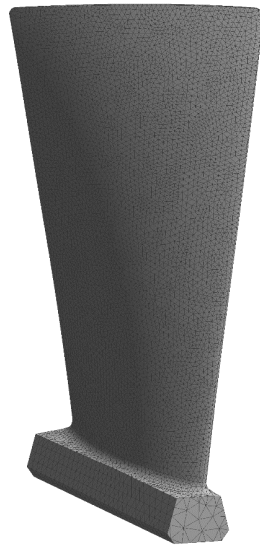
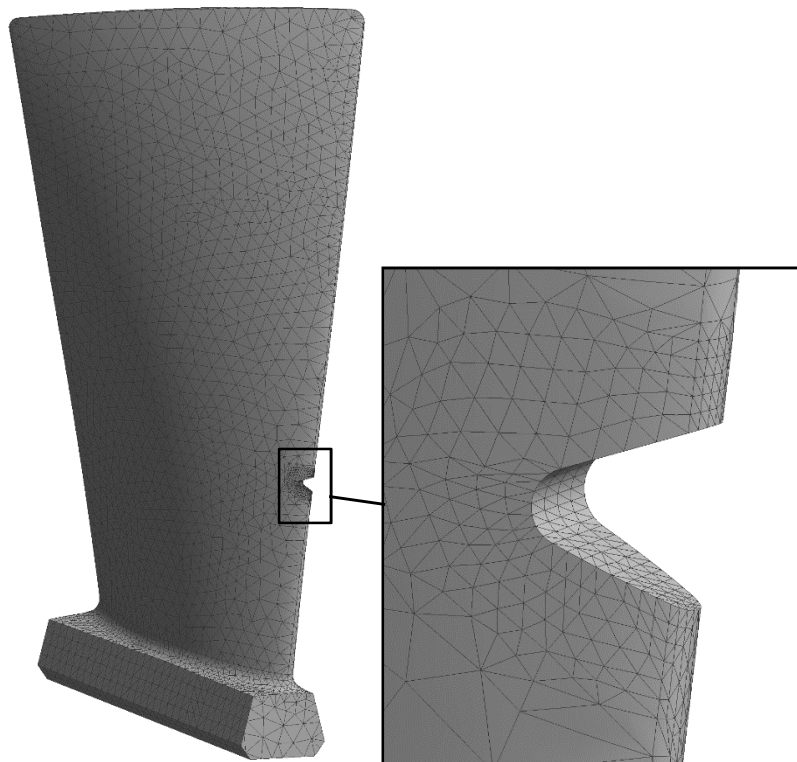Fig. 8. Tetrahedral mesh – no damage model

Fig. 9. Tetrahedral mesh with the V shape damage modeled

## 3.4. Material properties and boundary conditions
## – preprocessing

The material applied to the model was Structural Steel with parameters:
- Young's Modulus: 200 GPa
- Poisson's Ratio: 0,3
- Density: 7850 kg/m$^3$.

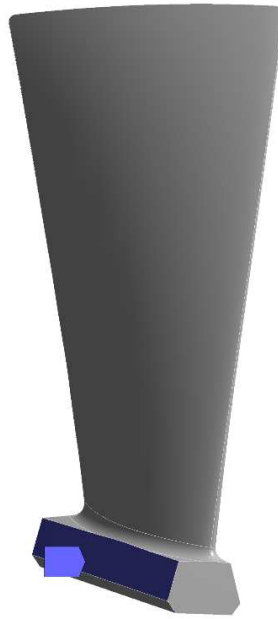Boundary conditions: blades were fixed at two symmetrically placed areas of the root – see Fig. 10.

Fig. 10. Boundary condition

## 3.5. Obtaining the solution

To obtain the solution, coupled analysis (modal and harmonic) were performed. Results of such analysis determine the dynamic behaviour – value of the first frequency mode and stress distribution on a blade.

46 analyses were conducted to calculate the behaviour of variously damaged turbine engine compressor blade in reference to non-damaged blade.

Figures 11 and 12 provide the information regarding first frequency mode for all achieved results. Figure 11 contains data from "I" shape defect analyses, "V" shape defect analysis are presented at Fig. 12.
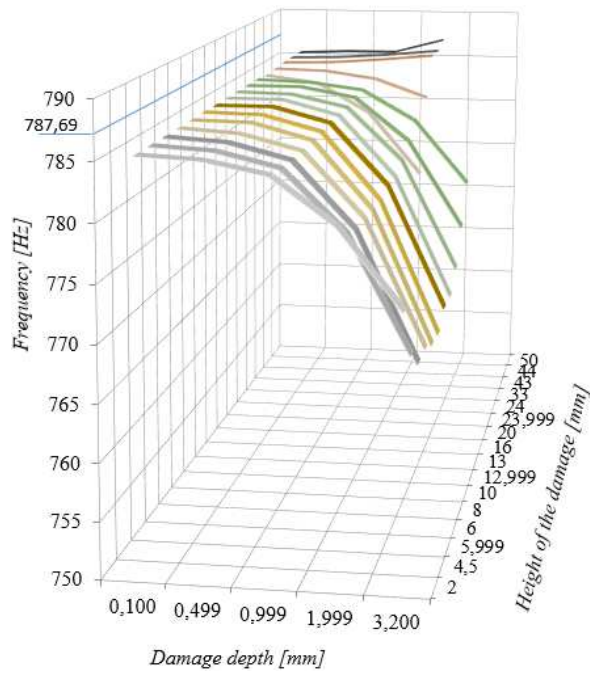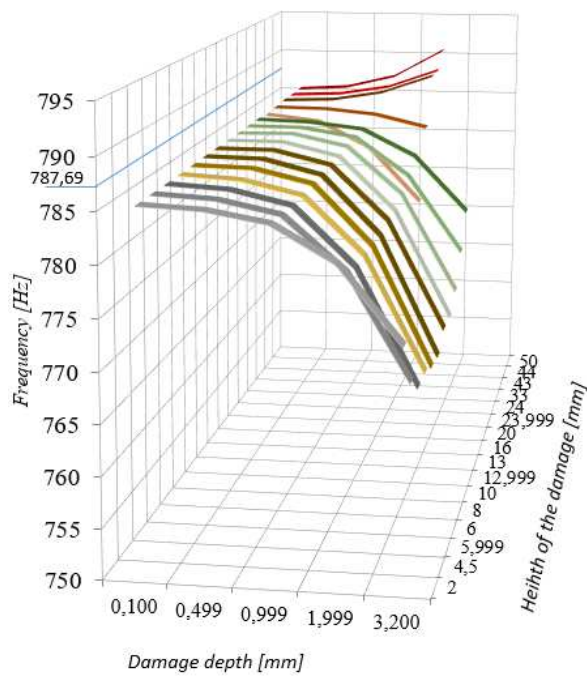
Fig. 11. "I" shape damage



Fig. 12. "V" shape damage

The conclusions based on first frequency mode data analysis:
- Frequencies decreases with damage depth risen
- Damage positions at the top of the blade and near the blade root does not influence the blade natural frequencies.

Figures 13 and 14 show the example of vonMises stress distribution at blades with damages located in a "C" level. Information of the maximum stress value aimed to precisely locate the strain gages at compressor blades for physical tests of fatigue strength research.
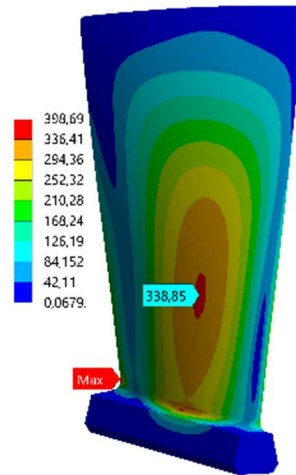


Fig. 13. Strength distribution (VonMises) on a blade with no damage
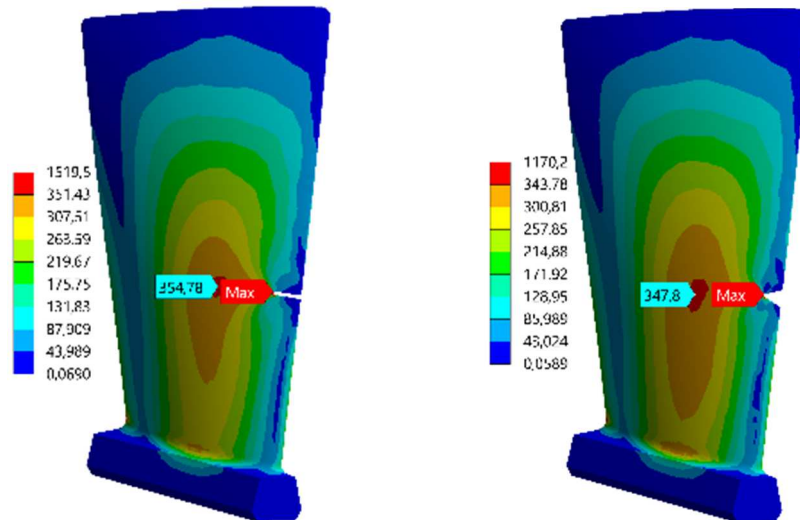


Fig. 14. Strength distribution (VonMises) on blades with "I" and "V" damage shape

## 4. Conclusion

Dynamic calculations with the use of Finite Element Method provide the opportunity for properly plan the research.

Information gathered from conducted analysis profited to adjust the research assumptions by cost reduction – less number of blades were used in the physical tests – and time optimization – avoided performing tests which results would not bring any relevant information.

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