FAST METHODS FOR STRUCTURAL ANALYSIS OF GAS TURBINE BLADES WITH MANUFACTURING IMPERFECTIONS

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Abstract

This paper presents the development and results of modeling turbine blades of aircraft engines with manufacturing imperfections. Approaches are presented that provide a fast analysis of the structure including varying geometrical properties. In particular the starting dynamics and the lifetime are investigated. Resonance conditions during run-up of the aeroengine are identified and the vibration characteristics are analyzed. Maximum vibration amplitudes are determined by superposition of partial frequency responses by linear weighting functions. The estimation of lifetime is based on a semi-analytical approach. Assuming a quasi-stable condition the stresses and the corresponding damage can be analyzed analytically. The damage of the structure caused by fatigue and creep is described by a linear damage accumulation according to Palmgren-Miner and Robinson. Geometrical imperfections are incorporated within a pre-processing routine using Principal Component Analysis. The influence of the scattered parameters on the structural properties is evaluated including probabilistic analysis. The application of the methods on a turbine blade of an air-turbine shows that the characteristic structural properties are captured sufficiently accurately.

1. INTRODUCTION

Gas turbine blades are counted among the most highly loaded components in aircraft engines. Therefore numerical simulations are performed to predict the exact behavior of the structure and to evaluate the blade design.
However, the properties of the manufactured blades may vary from the nominal values of these properties used in the design process. Due to tolerances in manufacturing processes deviations in structural properties of each gas turbine blade are unavoidable. At the same time, already small manufacturing imperfections might have a significant influence on behavior and performance of the structure. As shown by Lange et al. [1] manufacturing variability is not negligible when analyzing turbomachinery performance. Additional influences on the structure are investigated by Vogeler and Voigt [2], who analyzed the vibration characteristics of single blades and highlighted the importance including imperfections in turbomachinery design. Within the structural analysis the determination of the vibrations and lifetime of the blade are of major concern. Beside modal parameters, as natural frequencies and eigenmodes, the starting dynamics has to be considered. Vibration amplitudes during run-up of the engine are strongly influenced by changes in operating conditions and excitation frequency. Also the number of flight cycles a blade is able to resists before failure occurs is of particular importance. Cumulative damage methods allow a quick estimation of blade life.
Nevertheless the structural analysis of turbines blades is still very time-consuming. Especially regarding stochastic analysis a reduction in computation time using suitable simplifications and approaches is required.
In earlier works the basic ideas of efficient structural analysis have been proposed by Rogge and Rolfes [3]. Detailed lifetime estimation were performed by Holl et al. [4]. This present work combines the fast methods developed with stochastic analysis of manufacturing imperfections. It focuses on the modeling of gas turbine blades with geometrical imperfections, simulation of the starting dynamics and estimation of lifetime. Moreover, the application to a turbine blade of an air-turbine shows the numerical efficiency of the methods proposed.

2. STRUCTURAL ANALYSIS OF TURBINE BLADES

When evaluating the structural behavior of gas turbine blades, firstly the manufacturing imperfections of the structure have to be described properly. Furthermore simulations of the starting dynamics are performed considering the transient starting process. The influence of cyclic loads on the structure results in the estimation of lifetime.

2.1. Manufacturing imperfections

The manufacturing process influences various structural parameters. Material properties as well as the geometry can differ from blade to blade. In the present paper the focus is on variability of geometrical imperfections. Material properties of the blade are assumed to be constant.
A common method to include the variability of geometrical properties of the airfoil in structural analysis is the Principal Component Analysis (PCA) [5]. Measurements of the geometry of n blades are converted into a linear combination of r amplitudes a_i and uncorrelated eigenvectors $\vec{v}_i$ by singular value decomposition. The geometrical data of the structure is given by

$$\vec{x} = \sum_{i=1}^{r} a_i \vec{v}_i \qquad (1)$$

The number of amplitudes and vectors is maintained by the rank of the matrix of measured data. Using the PCA the number of input parameters is reduced significantly. Instead of varying the coordinates of every point on the surface of the airfoil the scatter in geometry is captured by the uncorrelated eigenvectors $\vec{v}_i$. The variability of the structure can be considered through the variation of r amplitudes a_i.

2.2. Operating conditions

The operating of the turbine blade result from influences caused by

- centrifugal forces,
- gas temperature,
- and gas pressure.

During one load cycle the operation conditions change with the rotational speed of the turbine. Centrifugal forces at each operation point are includesd in the analysis as quasi-static loads.
The gas temperature field is determined in stationary computational fluid dynamics (CFD) simulations and imposed by heat convection. The temperature within the structure results from heat distribution by conduction. The gas forces which lead to the main excitation mechanism of the nozzle excitation are described as the sum of static and dynamic pressure. The dynamic pressure is given by an harmonic function and the amplitudes are estimated by the static pressure multiplied with the empirical stimulus.

2.3. Starting dynamics

During run-up the operating conditions change with increasing rotational speed of the engine. The relation between the natural frequencies of the blade and the rotational speed is shown in a Campbell diagram in FIGURE 1. The natural frequencies are computed by modal analysis including pre-stress based on linear eigenvalue analysis at each supporting point n_j. The corresponding eigenmodes are associated with the natural frequencies using the Modal Assurance Criterion (MAC).

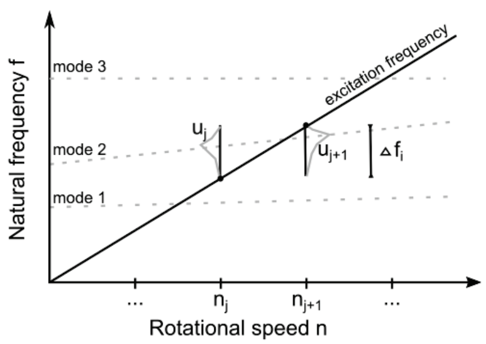

FIGURE 1. Superposition of partial frequency responses u_j and u_{j+1}

Potential resonance conditions are identified where the natural frequency of the structure and the excitation frequency coincide. Passing through resonance leads to increasing vibration amplitudes. Hence, the maximum vibration amplitudes are analyzed depending on the excitation frequency. The approach proposed is based on the frequency response analysis using modal superposition and frequency clustering. Since this can only be performed for steady state vibrations the vibration amplitude is approximated by superposition of partial frequency responses.
Vibration amplitudes in interval i result from partial frequency responses u_j and u_{j+1} computed at the supporting points n_j and n_{j+1}. The superposition between two supporting points is done using linear weighting functions W_{ij}

(2) $$u_i = \sum_{j=1}^{2} W_{ij} u_{ij}$$

The assembly of frequency responses for each interval corresponds to the vibration envelop during run-up.

2.4. Lifetime estimation

The damage the turbine blade has to suffer during service life is described using a linear damage accumulation rule according to Palmgren-Miner and Robinson. Loads resulting from fluctuating gas forces are assumed to be quasi-static. The forces are described by the static force component p_{stat} multiplied with a magnification factor M

(3) $$p = p_{stat}[1+M]$$

The magnification is specified as the ratio of vibration amplitude u and static displacement $|u_{stat}|$. On the basis of assumed loads the cycle at which structural failure occurs is analyzed. The total damage with damage contribution D_f and a creep contribution D_c is given by

(4) $$D_{tot} = D_f + D_c$$

According to Miner's rule [6] for cumulative fatigue damage the damage results in

(5) $$D_f = \sum_{i=1}^{l} \frac{n_i}{N_i(\sigma_i, \theta_i)}$$

where, n_i is the number of cycles of stress σ_i and temperature θ_i applied to the structure, N_i is the fatigue parameter corresponding to load σ_i and θ_i, and l is the number of relevant load cycles.
Similarly to EQ (5), the creep damage results from the ratio of actually experienced duration t_i of stress and temperature and the corresponding duration T_i

(6) $$D_c = \sum_{i=1}^{k} \frac{t_i}{T_i(\sigma_i, \theta_i)}$$

which is summed up over all relevant load cycles k. Due to the interaction between load cycles, the stresses change over the cycles. Thus, the whole service life of the blade has to be simulated, which is numerically costly. Reducing the computational effort a stress prediction is introduced. Assuming that every structure reaches a quasi-stable cyclic stress condition after load cycle $\tilde{n}$, all further stresses in the following load cycles can be calculated analytically. From load cycle $\tilde{n}$ each stress quantity is assumed to have a linear and therefore predictable progression. The gradient of the linear stress prediction is found by numerical calculation of stresses during load cycle $\tilde{n}$ and $\tilde{n}+1$. Hence, the evaluation of damage can be divided in an numerical and analytical part.

3. APPLICATION AND RESULTS

The two approaches presented for starting dynamics and lifetime estimation are used for the structural analysis of turbine blades of the axial air-turbine of the Institute of Turbomachinery and Fluid Dynamics (TFD) of the Leibniz Universität Hannover. The air-turbine is primarily designed for aeroelastic and aerodynamic investigations and therefore the maximum loads shown in TAB 1 are relatively small [7].

Description	Value
Number of vanes	29
Number of blades	30
Max. rotational speed	7500 rpm
Max. gas temperature	332 K
Max. static gas pressure	121 kPa

TAB 1. Parameters of the 5th stage of the axial air-turbine

Structural analysis is performed for a rotor blade of the last stage of the 5-stage air-turbine. The CAD-Modell of the turbine blade, which is made out of high strength aluminum alloy (Certal EN AW 7022) is depicted in FIGURE 2(a).
Imperfections of the surface of the blade due to manufacturing tolerances are measured with three-dimensional optical scanners. The deviations between ideal and real geometry are exemplarily shown in FIGURE 2(b). In this particular example the highest deviations are identified at the leading edge of the blade. Optical measurements are performed for 20 rotor blades of the 5th stage. The geometrical imperfections are mapped and prepared for probabilistic analysis using PCA.
For the finite element simulation performed with ANSYS the structure is meshed with fully-integrated tetrahedron elements with quadratic shape functions. The tetrahedral elements are used in order to capture the complex geometry of the blade. As shown in FIGURE 2(c) the local density of the finite element mesh increases at the blade root. Thereby the failure, which is expected to occur in the fir-tree region is captured accurately. In total, the chosen mesh comprises about 19600 nodes and about 11900 elements.

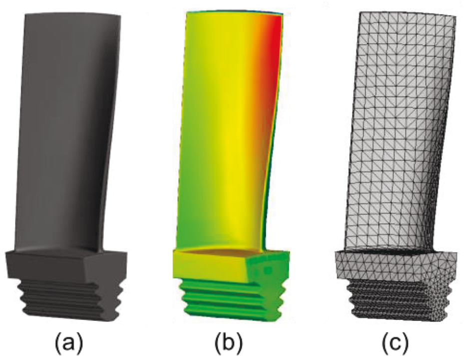

FIGURE 2. (a) CAD-Model, (b) three-dimensional optical scan and (c) finite element model of the blade

The boundary conditions at the blade root are simplified. During engine operation, the forces are transferred to the attached disk via the contact between the upper flanks of the fir-tree teeth and the groove. In order to avoid the numerically costly contact analysis, the degrees of freedom of the displacement field of the upper flanks are constrained in radial and peripheral direction. Fixing the structure in axial direction the displacement of the nodes located at the mid plane of the fir-tree are constrained in axial direction. The material characteristics of the aluminum alloy are described using a visco-elasto-plastic material law.

3.1. Starting dynamics

Analyzing the starting dynamics, the natural frequencies of the single turbine blade are illustrated in the Campbell diagram shown in FIGURE 3. The relevant natural frequencies corresponding to the first bending mode in flapwise direction, the first torsional mode and the first bending mode in edgewise direction are given as a function of rotational speed. Increasing the rotational speed leads to higher centrifugal forces, which can cause blade stiffening. However, due to the relatively low loads no significant stiffening effect can be observed.
The excitation of the blade is represented by the 29th engine order which corresponds to the 29 upstream vanes (TAB 1). This frequency leading to the major vibration excitation is also known as the nozzle passing frequency (NPF).

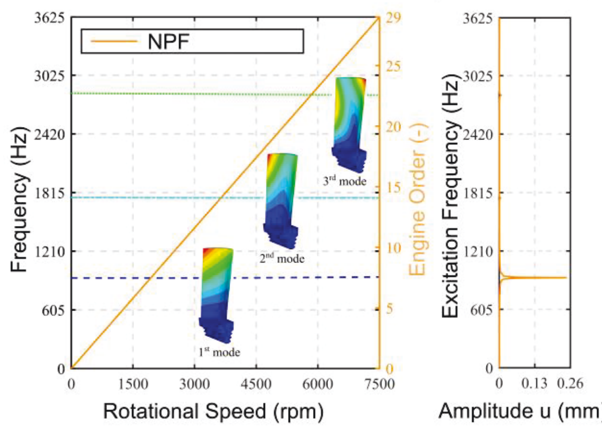

FIGURE 3. Campbell diagram (left) and frequency response (right)

In this particular example 3 potential resonance point are identified in the operating range between 0 and 7500 rpm. The vibration characteristics during run-up is computed by frequency response analysis based on the approach presented. The resulting frequency response of a node located at the trailing edge next to the blade tip is shown in FIGURE 3. At this node the highest vibration amplitudes can be observed. Especially the excitation corresponding to the natural frequencies of first bending mode leads to higher maximum vibration amplitudes. The vibration amplitude computed at the other two resonance points is relatively small.

Evaluating maximum vibration amplitudes caused by geometrical manufacturing tolerances 250 computations are performed using Latin Hypercube Sampling (LHS). The distribution of maximum vibration amplitude computed by the realizations is shown in FIGURE 4.

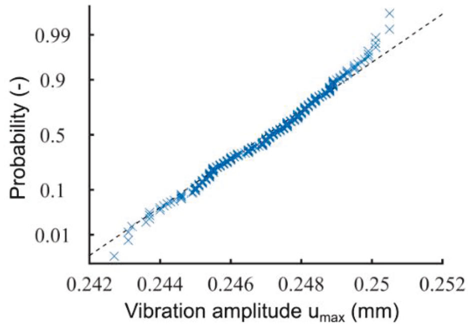

FIGURE 4. Distribution of maximum vibration amplitudes caused by geometrical variability

The scatter of maximum vibration amplitudes during run up lies within a range between about 0.24 and 0.26 mm. The probability plot of amplitudes additionally shows the comparison with a corresponding normal distribution and it can be concluded that the normal distribution fits the computed data quite well.

3.2. Lifetime estimation

For the estimation of lifetime the entire service life of the blade has to be analyzed. Stresses in the structure are determined for each load cycle. According to the damage accumulation rule given in EQ (4) – (6). In FIGURE 5 the damage caused by fatigue and creep is shown as a function of load cycles of the blade. The corresponding location, where maximum total damage is detected, is depicted. As expected the failure occurs in the fir-tree of the blade.

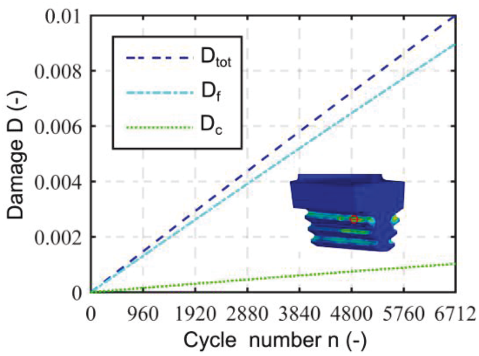

FIGURE 5. Total damage D_{tot} and damage contributes caused by fatigue D_f and creepage D_c

The damage is only presented for a total damage level of 1% and computations are terminated, because the damage process is approximately a linear one. According to the fast approach proposed the blade is damaged by 1% after being loaded during 6712 cycles. The quasi-stable condition required for analytical damage accumulation is reached after only 3 load cycles, therefore the analysis of lifetime can be performed relatively fast.

Moreover, it can be concluded that in this case the damage is primarily influenced by fatigue. Due to the low temperature in the air-turbine the contribution of creep is relatively low compared with damage cause by fatigue.

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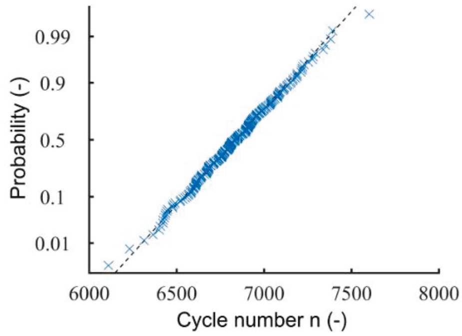

FIGURE 6. Distribution of number of cycles at 1% damage level caused by geometrical variability

The computational results determined using 250 realizations of varying geometrical input parameters are summarized in the distribution plot in FIGURE 6. Due to geometrical variances of the airfoil the cycles corresponding to a damage level of 1% scatter in the range between 6000 and 8000 cycles. The comparison with the normal distribution function again indicates that the output parameters are again normally distributed.

4. CONCLUSION

In this paper the basic ideas for an efficient structural analysis of turbine blades has been presented. The two approaches developed for the evaluation of starting dynamics and lifetime estimation allow the fast computation of structural characteristics. Only in this way it is feasible to perform probabilistic analysis, because various simulations have to be performed. Analyzing the maximum vibration amplitudes and maximum number of load cycles it was found, that geometrical deviations in the surface of the blade can have a significant influence. The scatter in output parameters shows that the influence of geometrical variability is not negligible and therefore has to be considered in the design process.

5. ACKNOWLEDGEMENT

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