ISSN: 1231-4005
e-ISSN: 2354-0133
DOI: 10.5604/12314005.1213707

INFLUENCE OF ENGINE ROTATIONAL SPEED ON THE NATURAL FREQUENCIES OF THE TURBINE BLADE

Lucjan Witek

Rzeszow University of Technology, Department of Aircraft and Aero Engines
Powstancow Warszawy Avenue 8, 35-959 Rzeszow, Poland
tel.:+48 17 865 1324, fax. +48 17 854 3116
e-mail: lwitek@prz.edu.pl

Abstract

In this work, the influence of the engine rotational speed on the natural frequencies of the turbine blade was investigated. In first part of the work the experimental modal analysis of the blade were made using the vibration system. The investigated blade was fixed to the movable head of the shaker. The blade amplitude was measured using the piezoelectric accelerometers. As results of experimental modal analysis, the resonant frequencies for first three modes of vibration were determined. In the next step of the work, the numerical models of the blades with different size of the finite elements were performed in order to check the convergence of the numerical solution. For each models the natural frequencies were determined for the non-rotated blade. The numerical results were next compared to the results of experimental modal analysis. After this comparison, one model only was selected for the further computations. In the last part of the work, the complex multi-steps analysis was performed in order to investigate the influence of turbine rotational speed on the modes and natural frequencies of the blade. Obtained results are important from both the research and also the practical point of view and have an influence on reliability of the engine.

Keywords: *turbine blade, modal analysis, resonant frequency, finite element method, experimental analysis*

1. Introduction

The turbine blade belongs to the group of critical engine parts, which have a limited fatigue life. The major function of the turbine blades is to extract energy from the hot gas flow to drive the compressor and the main helicopter gearbox. The main mechanical load is concerned with the centrifugal forces acting on the blade during the rotation of the turbine. The centrifugal forces cause that in the blade the tension stress occurs. This stress has a large influence on the blade's natural frequencies.

The turbine blades have a low bending stiffness and are particularly susceptible to vibrations. Engine vibrations are associated with both the imbalance of the rotor and the pulsation of an air in the combustor chamber. As results of mentioned phenomena, a high-cycle fatigue occurs in the blade. In many cases the vibrations and related to vibrations the high-cycle fatigue was a main reason for the blade fracture. The problem of turbine failure was the subject of several investigations [6, 10, 12, 13]. The results of stress, fracture and modal analysis of the blade were described in the papers [2-5, 9, 11, 14].

The main objective of this work is an experimental and numerical determination of the resonant frequencies and modes of free vibrations of the turbine blade. An additional aim of the study is investigation of influence of the turbine rotational speed on the natural frequencies of the blade.

2. Experimental modal analysis of the turbine blade

The experimental modal analysis of the turbine blade was made using the Unholtz-Dickie UDCO TA-250 vibration system presented in Fig. 1a. The blade was horizontally mounted on the movable head of the shaker (Fig. 1b). During the modal analysis, the frequency of excitation was increased from 500 Hz to 4000 Hz with the sweep rate of 2 Hz/s. In experimental analysis a two measure channels were activated. The first channel was used for measure the amplitude of acceleration of the shaker head (signal name A_1). The acceleration in this channel was measured by piezoelectric accelerometer. The signal (A_2) in the second channel was delivered from the miniaturized small mass (0.2 gram) piezoelectric acceleration sensor. This sensor was connected to the blade with the use of the special wax (Fig. 1b).

a) *b)*

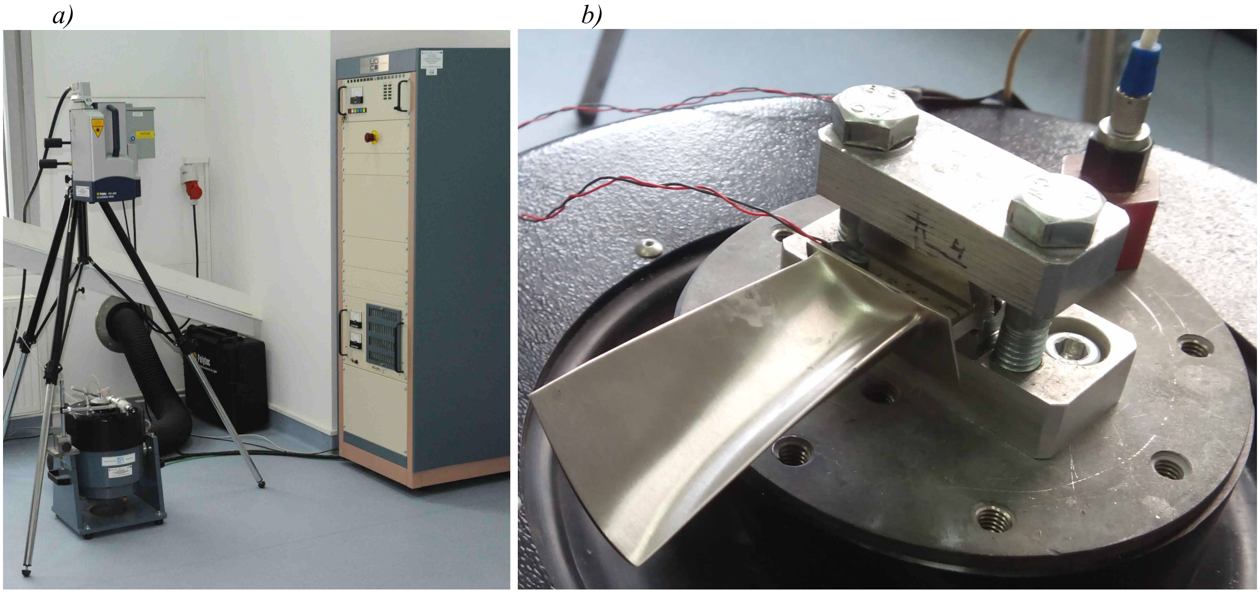

Fig. 1. View of the vibration system (a) and blade fixed to the head of vibrator (b)

During the modal analysis, the frequency of excitation was gradually increased. In this time, the signals from two measured channels were recorded. The main result of the modal analysis is the amplitude-frequency characteristic presented in Fig. 2. On the horizontal axis of the plot, the actual frequency of excitation was defined. On the vertical axis, the relative amplitude of the blade is seen. The relative amplitude means that the amplitude of acceleration from the second channel (A_2) was divided by the signal from the first channel (amplitude of acceleration of the vibrator head, A_1).

As seen from Fig. 2 the large peak, typical for resonance phenomena occurs at 980.8 Hz. This value is the resonant frequency (for first mode of vibrations). The relative amplitude during the mode I equals 130.01. There is visible a second peak on the frequency-amplitude characteristic at 2642.4 Hz. This value is the resonant frequency for the second mode of vibrations. The third resonant frequency equals 3868.2 Hz (Fig. 2).

The large disturbances of the signal between the resonances were concerned with the small sensitivity of the miniaturized acceleration sensor connected to the blade. The second reason for the signal disturbances is the small acceleration amplitude of the vibrator head (from 0.5 g at 500 Hz to 1 g at 4000 Hz) defined in the experimental modal analysis.

3. Numerical models of blade

The numerical models of the blade were made using the MSC-Patran program [8]. The material of the blade (EI-437-y nickel alloy) was defined as linear-elastic with the following properties: modulus of elasticity – 200 GPa, Poisson's ratio – 0.3, density – 8200 kg/m^3 [7]. EI-437-y alloy has a good creep resistance and it is used for hot rotational components of the turbine engines.

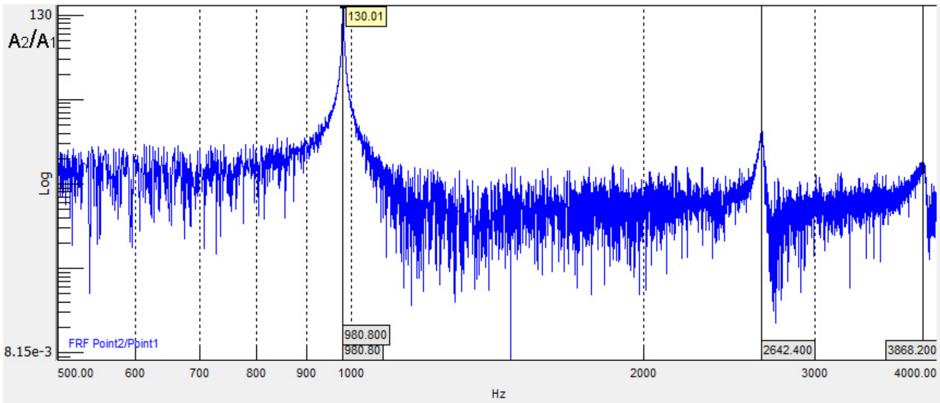

Fig. 2. Amplitude-frequency characteristic for investigated turbine blade

The results of the numerical analysis performed for the first model (model no. 1, size of element in airfoil section: 4 mm) were not converged with the experimental results (Tab. 1 and 2). From this reason the additional models (numbered as 2, 3 and 4) with the smaller size of the finite elements were performed for better convergence of the numerical results (Fig. 3, Tab. 1). In all discrete models of the blade, the TET-4 finite elements with the linear shape function were used [1]. In the analysis the surfaces of fir-tree slots (blade lock) were fully constrained (T_x=0, T_y=0, T_z=0).

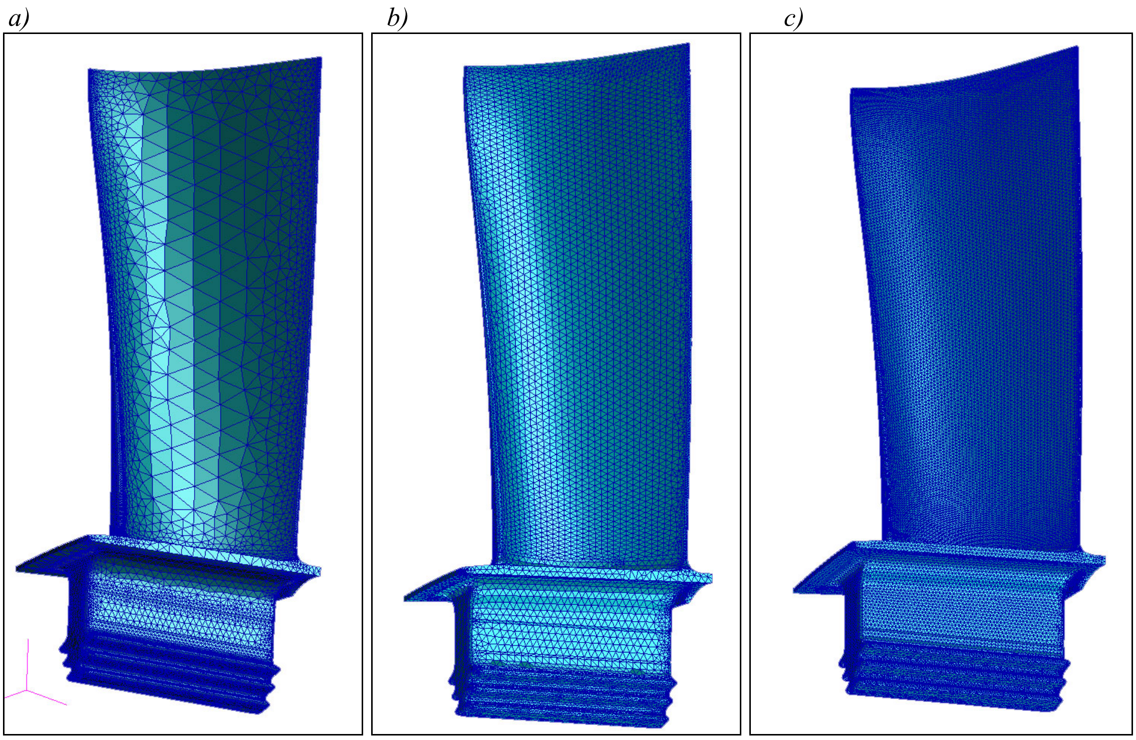

Fig. 3. Numerical models of the blade with different mesh size. Model no. 1 (a), no. 2 (b) and no. 4 (c)

4. Results of numerical modal analysis for the blade

The numerical modal analysis was made using the Abaqus solver [1]. As results of the computations the first 5 modes and natural frequencies were obtained for the blade nos. 1-4 (for rotational speed of turbine n=0 RPM). The resonant frequency for model no. 1 (mode I) equals 1034 Hz. The result of experimental test is equal to 980.8 Hz.

For estimate the quality of the numerical solution, the relative error E was defined according to the formula (1):

$$E = \frac{F_{RES} - F_{NF}}{F_{RES}} \times 100\%, \tag{1}$$

where:
F_{RES} – resonant frequency (experimental result),
F_{NF} – natural frequency (numerical result).

The relative error E (Model no. 1, mode I) equals 5.42% (Tab. 2). For the second and third mode of vibrations, the error is much larger (34.08% and 34.04% adequately). So large difference between the numerical and experimental results is not acceptable for both the research and also the engineering applications.

The additional information is that the model no. 1 has a mesh concentrated in the fir-tree lock of the blade. On the airfoil section, the size of the finite elements equals 4 mm. This kind of mesh seed is typical for the stress analysis. Obtained results indicate that increase the number of finite elements in the model not always causes the decrease of the error (Tab. 1). Results presented in Tab. 1 and 2 showed that is a large correlation between the size of the finite elements (especially in the airfoil section zone) and the relative error of numerical solution.

As seen from Tab. 2, for the model no. 2 the relative error (mode II) decreases to the value of 6.87%. This model has a size of the finite elements of 0.83 mm (Tab. 1). In the model no. 3 (element size of 0.54 mm), the error for mode II equals only 0.78%. For the model no. 4, the error has a value -3.43%. Negative error value means that the numerical result is smaller than an experimental. For the third mode of vibrations the errors for models nos. 1-4 are 34.04%, 8.45%, 6.2% and 5.11% adequately.

Tab. 1. Natural frequencies of the blade computed using numerical models with different size of the finite elements in airfoil section (for n= 0 RPM)

Model no.	Size of elem. [mm]	Number of elements	Number of nodes	Time of comput. [s]	Natural frequency [Hz]				
					Mode I	Mode II	Mode III	Mode IV	Mode V
1	4.1	295 700	58 082	282	1034	3543	5185	5896	10784
2	0.83	167 631	31 764	161	1014	2824	4195	5441	7187
3	0.54	353 478	70 754	349	992	2663	4108	5329	6645
4	0.41	834 486	159 726	1018	982	2552	4066	5252	6302

Tab. 2. Comparison of experimental and numerical results obtained for different models (n=0 RPM)

Model no.	Natural frequency / Resonant frequency								
	Mode I			Mode II			Mode III		
	Numerical [Hz]	Experimental [Hz]	Rel. error E [%]	Numerical [Hz]	Experimental [Hz]	Rel. error E [%]	Numerical [Hz]	Experimental [Hz]	Rel. error E [%]
1	1034	980.8	5.42	3543	2642.4	34.08	5185	3868.2	34.04
2	1014	980.8	3.38	2824	2642.4	6.87	4195	3868.2	8.45
3	992	980.8	1.14	2663	2642.4	0.78	4108	3868.2	6.20
4	982	980.8	0.12	2552	2642.4	-3.42	4066	3868.2	5.11

From presented above comparison is visible that the results for the models nos. 3 and 4 have the smallest errors. The large time of computations for the model no. 4 (1018 seconds for one job, Tab. 1) causes that in further analysis presented in chapter 5 the model no. 3 will only be considered. For this model, the time of computations equals 349 seconds.

In Fig. 4, the von Mises stress distributions for the first, second and third modes of free vibrations of the blade are presented. The results are related to the maximum amplitude of blade vibrations close to 1 mm. The maximum von Mises stress area (351 MPa) is located (for mode I) in the zone above the blade lock. The mode I can be classified as pure flexural (Tab. 3).

During the second mode of free vibrations, the maximum stress area (294 MPa) is located on the blade airfoil section. It explains why smaller size of the finite elements in this zone causes the decrease of relative errors for the models nos. 3 and 4. The mode II can be classified as torsional-flexural (combination of torsion and bending).

The third mode of free vibration is pure torsional. The large stress area in this case is located on the attack edge of the blade (842 MPa). The maximum stress zone (1264 MPa) is located above the blade lock. The maximum stresses obtained in the modal analysis should be treated as a quality results. The real stress during the blade resonance depends on the real amplitude of blade vibrations.

Tab. 3. Identification of the modes of blade free vibrations

Number of mode	I	II	III	IV	IV
Kind of mode (identification)	flexural	torsional-flexural	torsional	complex	complex

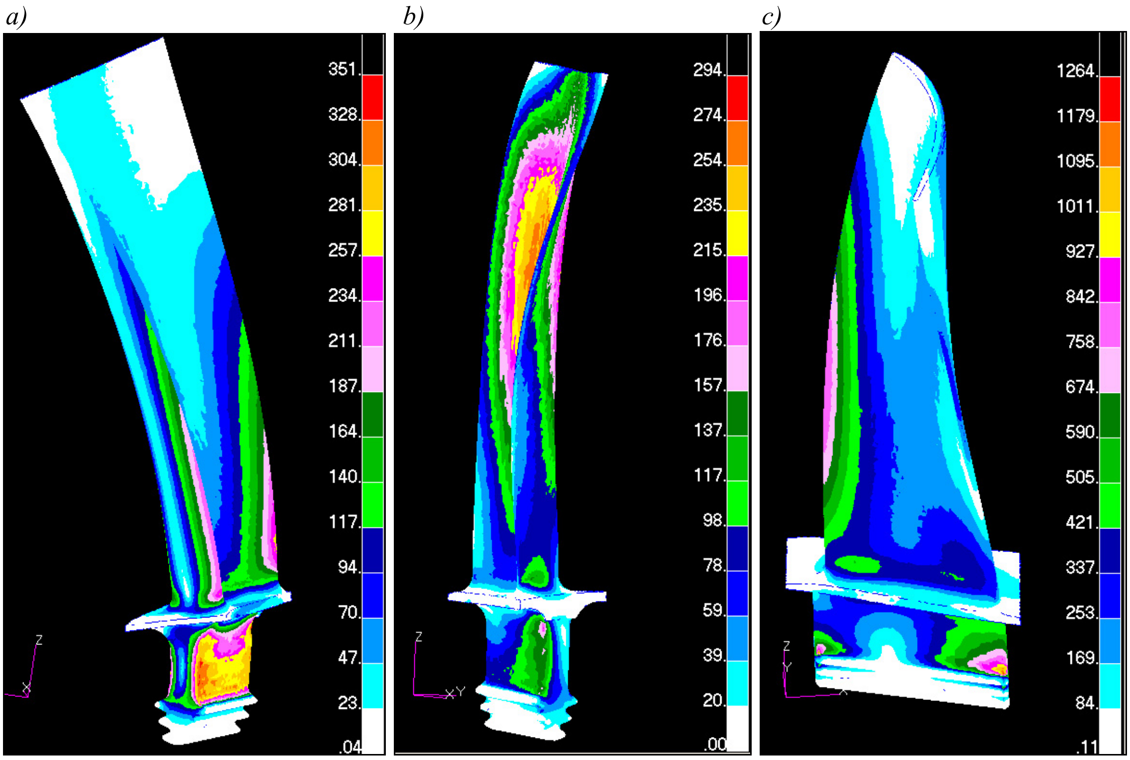

Fig. 4. Von Mieses stress distributions for the turbine blade during first (a), second (b) and third (c) mode of free vibrations (for blade amplitude close to 1 mm, blade no. 3, n=0 RPM)

5. Influence of turbine rotational speed on the natural frequency of the blade

During rotation of the turbine a large centrifugal stresses occur in the blade. The tension stress in the blade causes that the natural frequencies increase during an acceleration of the engine rotor. In this chapter, the influence of the rotational speed of turbine on the natural frequencies of the blade was investigated.

In order to solve the problem, the 3 separated steps were defined in one job of the numerical analysis. In the first step, the centrifugal forces were applied to the blade subjected to rotation. As a result, the stress distribution for the rotated blade was obtained (Fig. 4). In the second step the modal analysis of the blade were performed using Abaqus command: “Frequency” [1]. In the third step, the “Complex frequency” analysis [1] was used to extract the natural frequencies of the blade. In this procedure the stress, state obtained in the step no. 1 (for blade subjected to rotation) is considered as preliminary in the modal analysis.

The results of modal analysis of the rotated blade are presented in Tab. 4. The maximum rotational speed of the engine rotor (22 490 RPM) should be related to the value of 100% speed of turbine. In this study, the following speeds: 0%, 50%, 80% and 90% were considered. Moreover the over speed condition of the turbine (speed of 110%) was also defined.

As seen from Fig. 5 the natural frequency of the blade for the rotor speed of 0% equals 992 Hz. After acceleration of the turbine to the speed of 100%, the natural frequency of the blade is equal to 1147 Hz (increase at about 16%). During the second mode of free vibration, the natural frequency increases from 2663 Hz to the value 2787 Hz (Fig. 6).

During the torsional mode of vibrations (mode III) there is visible a small influence of the rotational speed of turbine on the natural frequency of the blade (increase of frequency at about 1%) (Fig. 6). In this case, the tension stress in the rotated blade has no large influence on the shear stress occurring during the blade torsional vibrations.

Tab. 4. Natural frequencies of the blade (model no. 3) for different rotational speed of turbine

Turbine speed [%]	Turbine speed [RPM]	Natural frequency [Hz]				
		Mode I	Mode II	Mode III	Mode IV	Mode V
0	0	992	2663	4108	5329	6645
50	11245	1032	2694	4116	5361	6690
80	17 992	1093	2743	4132	5413	6763
100	22 490	1147	2787	4148	5461	6829
110	24739	1177	2812	4157	5489	6867

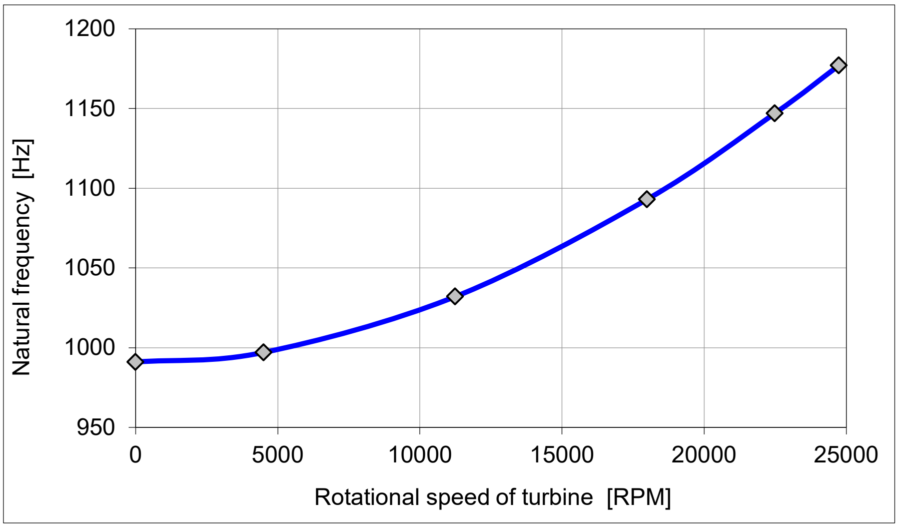

Fig. 5. Natural frequency of the blade (mode I) as a function of the rotational speed of turbine (model no. 3)

6. Conclusions

In this work, the influence of the rotational speed of turbine on the natural frequencies of the blade was investigated. In the first part of the study, the experimental analysis was performed in order to determine the real resonant frequencies of the turbine blade. Next, the convergence of the numerical results was checked for selected models with different size of the finite elements. In the last part of the study the numerical modal analysis of the blade (with consideration of the turbine rotational speed) was performed. During the work preparation, the following conclusions were formulated:

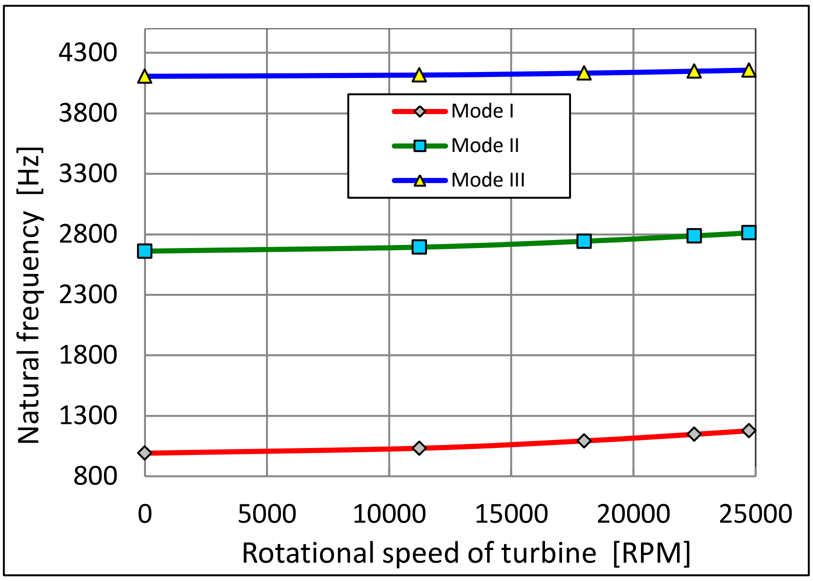

Fig. 6. Natural frequencies of the blade for first, second and third mode of vibrations as a function of rotational speed of turbine (model no. 3)

1. The error for the model no. 1 achieves the value of 34% (for mode II and III). The numerical model used for stress analysis (with the mesh concentrated in fir-tree blade area and large finite elements in airfoil section) is not adequate for the modal analysis.
2. The model of blade for the numerical modal analysis should have a small size of the finite elements in the airfoil region, in which the large stresses are observed at higher modes of free vibrations. In presented analysis the sufficient quality of numerical results were obtained for the model no. 3 with the element size of 0.54 mm. In this case, about 4 elements were composed on the profile thickness.
3. The natural frequencies of the blade for the mode I and II (after turbine acceleration to the speed of 22 490 RPM) increase adequately at 16% and 4.6%.
4. The natural frequency (for mode III) increases in rotated blade at about 1%. The centrifugal forces in rotated blade have no large influence on the shear stress, which occurs during pure torsional vibrations.
5. Results of the work [15] show that after blade heating to the working temperature (400-580ºC) the natural frequencies of the rotated blade decrease at about 6%. This phenomenon is related to the decrease of the modulus of elasticity of the blade hot material. Additionally, the hot blade has a larger size what has also influence on decrease of the natural frequencies.

Acknowledgement

The research leading to these results has received funding from the People Programme (Marie Curie International Research Staff Exchange) of the European Union's Seventh Framework Programme FP7/2007-2013/ under REA grant: PIRSES-GA-2013-610547.

References

[1] ABAQUS ver. 6.9 *Users Manual*, Abaqus Inc., 2009.
[2] Akash, S. Harsha, S. P., *An Experimental and FEM Modal Analysis of Cracked and Normal Steam Turbine Blade*, Materials Today: Proceedings, Vol. 2, pp. 2056-2063, 2015.
[3] Chan, S. K., Tuba, I. S., *A finite element method for contact problems of solid bodies – Part II: Applications to turbine blade fastenings*, International Journal of Mechanical Sciences, Vol. 13, pp. 627-639, 2007.

[4] Gwo-Chung, T., *Rotating vibration behavior of the turbine blades with different groups of blades*, *Journal of Sound and Vibration*, Vol. 271, pp. 547-575, 2004.
[5] Kowalski, T., Mamrowicz, J. Witek, L., *Numerical stress and fatigue analysis of the first stage power turbine for helicopter engine*, Proceedings of the 24-th International Committee on Aeronautical Fatigue (ICAF) Symposium, Naples, Italy 2007.
[6] Maktouf, W., Sai, K., *An investigation of premature fatigue failures of gas turbine blade*, Engineering Failure Analysis, Vol. 47, pp. 89-101, 2015.
[7] Michailov, P. B., *Sprawocznik po Metaliczeskim Matierialam Turbino-i Motorostroenija*, Petersburg, 1961.
[8] MSC-Patran 2009 *Users Manual*, MSC Corporation, Los Angeles 2009.
[9] Papanikos, P., Meguid, S. A., Stjepanovic, Z., *Three-dimensional nonlinear finite element analysis of dovetail joints in aero-engine discs*, Finite Element Analysis and Design, Vol. 29, pp. 173-86, 1998.
[10] Park, M., Hwang Young, H., Choi, Y. S, Kim, T. G., *Analysis of a J69-T-25 engine turbine blade fracture*, Engineering Failure Analysis, Vol. 9, pp. 593-601, 2002.
[11] Poursaeidi, E., Hosein, B., *Fatigue crack growth simulation in a first stage of compressor blade*, Engineering Failure Analysis, Vol. 45, pp. 314-325, 2014.
[12] Song, K. S., Kim, S. G., Jung, D., Hwang, Y. H., *Analysis of the fracture of a turbine blade on a turbojet engine*, Engineering Failure Analysis, Vol. 14, pp. 877-883, 2007.
[13] Witek L., *Failure analysis of turbine disc of an aero engine*, Engineering Failure Analysis, Vol. 13, 9-17, 2006.
[14] Witek, L., *Fatigue investigations of the compressor blades with mechanical defects*, Key Engineering Materials, Vol. 598, pp. 269-274, 2014.
[15] Witek, L., Wyzga, B., *Analiza wpływu wybranych rodzajów obciążeń na częstotliwość drgań rezonansowych łopatki turbiny silnika śmigłowcowego*, Logistyka, Vol. 12, pp. 11187-11195, 2014.