Sensitivity Analysis Regarding the Impact of Intentional Mistuning on Blisk Vibrations

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Abstract. The effect of different intentional mistuning (IM) patterns is investigated with respect to the forced response of an academic axial blisk. It could be shown in numerical analyses that a preliminary use of sensitivity algorithms helps to understand the feasibility and efficiency of introducing geometric changes of the blades. The implementation of IM patterns requires conducting intensive sensitivity studies based on FE simulations in order to identify the consequences of slight geometrical blade modifications on natural frequencies. Typical changes might be a modification of fillet radii or partial modifications of blade thickness, which are most suitable to adjust a target natural frequency without a severe loss of aerodynamic performance. A software tool developed at Irkutsk SAU is employed to evaluate the impact of mass and stiffness contributions, and with that, geometric deviations on blade natural frequencies.

Intensive blade vibration due to aerodynamic excitation of blisks is known as major source of high cycle fatigue, which may cause severe failures of turbine and compressor wheels during operation. The problem is relevant for several sectors of industry such as power generation, aviation or vehicle manufacturing. In consequence, there is a broad request of preventing any inadmissible vibration at any time. The application of IM can be regarded as powerful tool to avoid both, large forced responses and self-excited vibration. However, there is a lack of knowledge about how to implement mistuning without strong distortions of the flow passage. The main objective of this work is to close this gap based on comprehensive numerical analyses with regard to the effects of intended geometric modifications of blades on modal quantities.

Using FE models, the effectiveness of the proposed block models of mistuning is analyzed with and without taking into account the operational speed of the axial impeller. In conclusion, the consequences of different IM implementations on the forced response of an academic blisk are discussed. In particular, the most promising IM patterns are identified yielding the least forced response.

Keywords: Intentional mistuning · Sensitivity · Blisk · Forced response

The increasing demand for more efficient, more economic and environmentally friendly operation of turbomachines has led to an increasing application of integrally bladed wheels, meaning that blades and disks of turbo-machine wheels are manufactured as one piece. This technology satisfies the request for introducing light weight solutions in various fields of industries and allows for higher rational speeds, higher aerodynamic

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efficiencies and stage pressure ratios compared to the conventional design with separated blades and disk. However, integral designs are faced to higher stress levels and extremely low mechanical damping due to the lack of friction damping. Additionally, the negative consequences of unpreventable, manufacturing random mistuning are becoming worse, which further facilitates the susceptibility of blisk towards vibration.

Vibrations analyses of mistuned rotors are most frequently addressing the effect of mistuning on dynamic characteristics such as the maximum forced response and consequently on fatigue strength and durability. Blade mistuning manifests itself in small differences between the blades in terms of mass, geometry, material, etc., which violate the cyclic symmetry of rotor. According to [6] the magnitude of mistuning may be quantified by means of

$$\Delta f_i = \frac{f_{j,i} - \bar{f}_j}{\bar{f}_j} \tag{1}$$

where $\bar{f}_j$ represents the arithmetic mean of the blade dominated frequencies assigned to the j^{th} blade mode shape, and $f_{j,i}$ denotes the blade alone frequency of the i^{th} blade ($i = 1, \ldots, N$; N - number of blades).

A significant effect regarding the vibration of mistuned systems is an increase of the maximum forced response in terms of displacements and stresses compared to the ideal system. The factor γ is introduced for a quantitative assessment the maximum increase of amplitudes, which connects the maximum response amplitude of mistuned system with that of tuned system

$$\gamma = \frac{u_{mistuned\,(max)}}{u_{tuned\,(max)}} \tag{2}$$

where amplitude is understood as the maximum displacement or maximum dynamic stress during forced vibrations. The amplitude amplification factor substantially depends on both mistuning magnitude and shape of mistuning distribution. In theoretical calculations Ewins simulated the influence of different mistuning distributions on maximum vibration amplitude, which can vary from 130% to 210% [10] compared to the 100% tuned counterpart. Whitehead introduced a formula to estimate the maximum magnification of the forced response amplitude [5].

Contrary, the application of intentional mistuning has proved to mitigate the negative impact of random mistuning on the maximum forced response [4, 9, 11, 12]. Commonly, sensitivity analyses are carried out to identify suitable intentional mistuning patterns. The basic idea in this regards is to evaluate the consequences of changing an original parameter X_0 by means of ΔX. Hence, the modified system is characterized by $X = X_0 + \Delta X$ which has to be substituted in the static or dynamic equations. The main theory and algorithms for optimum design of turbine blades based on sensitivity analyses have been worked out e.g. by Kaneko, Mase and Fujita [1], Repetski and Zainchkovski [2], or Repetckii, Ryzhikov and Nguyen [3].

1 Numerical and Experimental Analyses of the Blisk

Numerous studies of different authors have shown that the phenomenon of mistuning may significantly affect the operation of power and transport turbine engines since the maximum displacement amplitudes and dynamic stresses can sharply increase during forced vibrations. An academic blisk with 10 blades, manufactured at Brandenburg University of Technology, was chosen as the object of study. The blisk is made of steel featuring a Young's modulus of $2.1 \cdot 10^5$ MPa, a Poisson's ratio of 0.3, and a density of 7850 kg/m^3. The general view of the rotor and one sector is shown in Figs. 1a and b. Supported by experimental data of authors, numerical studies in this work were carried out using both ANSYS and ABAQUS software packages. Figure 1c shows a finite element model of one sector, which uses ANSYS TET10 triangular finite elements, each having 3 degrees of freedom per node with a total of 2515 finite elements (FE) and 14616 degrees of freedom (DOF). The designated model was rigidly fixed at the rim of disk.

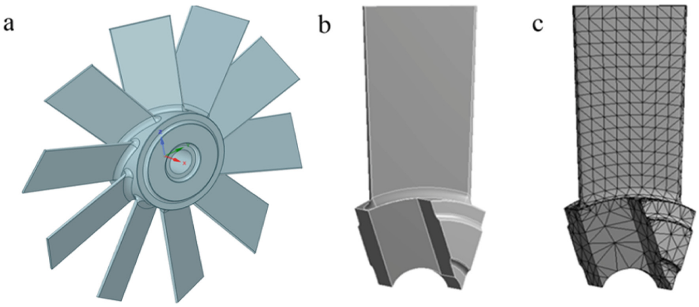

Fig. 1. Academic blisk with 10 blades (a) full disk; (b) one sector; (c) sector with FEM.

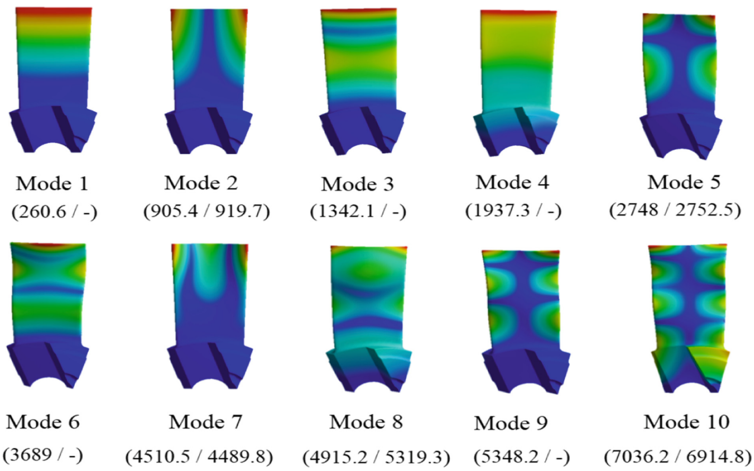

Fig. 2. Vibrational modes of one blisk sector without rotation (FEM/Test) in Hz.

Both natural frequencies and vibrational modes represent dynamic properties of blades. Thus, numerical analyses of natural frequencies and vibrational mode of the blades is a key task in designing turbomachines. Figure 2 shows the first ten sector mode shapes.

In order to prepare the design of suitable intentional mistuning patterns yielding a mitigated forced response, the so-called sensitivity function has been investigated to control and improve the efficiency of intentional mistuning. Some authors [2, 3, 6, 8, 13] show that blade sensitivity analysis can help to determine the location of mistuning zone being useful for the design of promising intentional mistuning patterns. Consequently, stress levels in the blade can be reduced by means of the results of sensitivity analyses, and therefore, the vibration susceptibility is decreased. For this reason, additional masses for detuning the blisk are employed in order to determine locations of suitable mistuning zones affecting the maximum effect on the forced response of the structure (Fig. 3). Here it is shown how the location of the mass affects the change of natural frequencies (bold blue points represent the zone of maximum frequency decrease, whereas bold red points represent the maximum frequency increase due to employing the mass in this zone. The sensitivity analysis of free and forced vibration blades for changing of thickness is performed in [6].

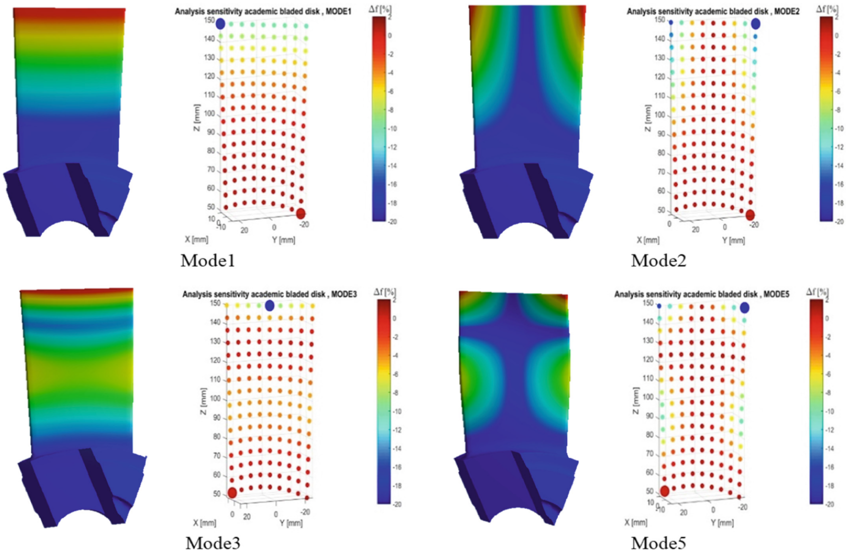

Fig. 3. Mode shapes (left) and sensitivity analysis (right) of academic blisk.

The analyses of the sensitivity of the mode shapes in Fig. 3 show that the zones of maximum sensitivity regarding the location of the masses on the blade coincide with the zones of maximum displacement during vibrations. The arrangement of the masses on nodal lines of the blade does not actually change the frequency of its natural frequencies. The location of the masses close to the blade tip reduces the natural frequencies (blue), and to maximize the frequency of natural vibrations, it is necessary to place the mass in the root part (red) of the effect of mass, in this case, is equivalent to a change in stiffness. The mistuning values are random values. Using experimental method to assess the effect of mistuning on the dynamics of rotor is a difficult task because it is necessary to analyze a large number of blisk hardware for experimentally determining mistuning patterns during the experiment. Numerical methods such as the Monte Carlo method can be used to study random mistuning [9]. The measurement system for experimentally determining mistuning is shown in Fig. 4a. The experimental setup consists of blisk, control device, laser vibrometer, modal hammer and foam pad [6]. The part of experimental data is shown in Fig. 2.

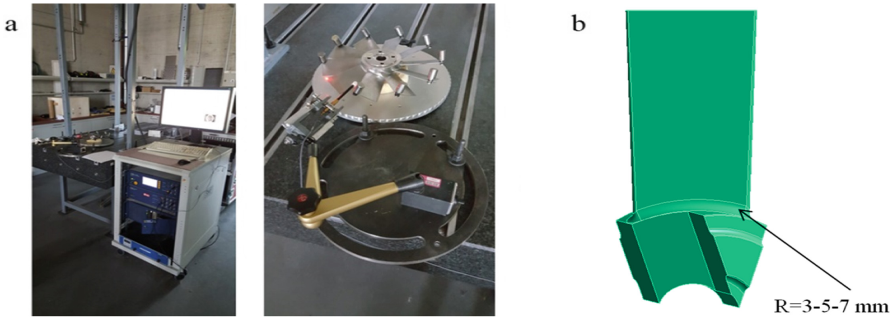

Fig. 4. (a) Experimental setup; (b) The changing of the radius.

2 Investigation of Intentional Mistuning

Following analyzes aim at analyzing the impact of intentional mistuning, which is accomplished such that the flow around the blades is hardly. As such types of wheel mistuning, changes in the radius of rounding of the transition of the blade to the disk and changes in the thickness of the blade are considered. Figure 4b shows a sector model of the academic blisk and the changing of the fillet radius from 3 mm to 7 mm. Such a measure basically enables the fine-tuning of real parts of turbomachines with respect to an attenuation of the forced response [7].

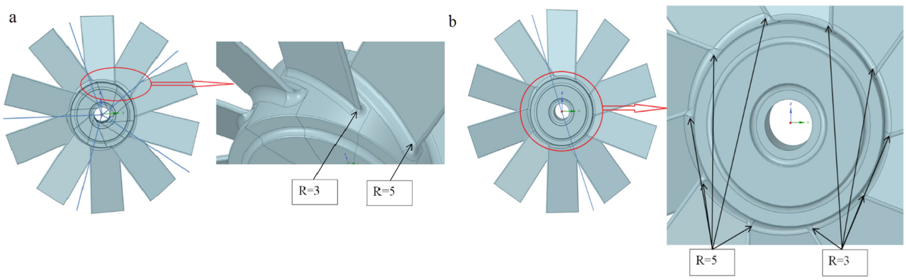

Fig. 5. (a) Block model No. 1 of the blisk mistuning; (b) Block model No. 2 of the blisk mistuning.

Following, we consider the first block mistuning model (BM) of alternate blade mistuning featuring blade root fillet radii of R = 3 mm or R = 5 mm, respectively (Fig. 5a, the variant without mistuning features R = 5 mm). Table 1 indicates the values of the maximum magnification factors due to changing fillet radii between blade root and disk, taking into account the intentional mistuning. The results show that the second block model (5 blades in one mistuning group, Fig. 5b) with intentional mistuning works most effectively. The second BM is effective for both at rest and at maximum speed (n = 100 1/s). The rows of the table show the value of the maximum magnification coefficient for the first forms in torsion mode (1T) and in bending mode (1B) when changing the initial radius of R = 5 mm to radii of 3, 1 and 7 mm.

Table 1. Calculation of the value of the maximum magnification factor due to changing fillet radius at the blade root

Block model (Number)	Mode shapes	R5 - R3 (n = 0 1/s)		R5 - R1				R5 - R7 (n = 0 1/s)	
				(n = 0 1/s)		(n = 100 1/s)			
		γmax	%	γmax	%	γmax	%	γmax	%
1	1T	1,63	−21,63	1,72	−17,31	1,88	−9,61	2,15	+3,34
	1B	1,10	−47,12	1,08	−48,10	1,10	−47,11	1,95	−6,25
2	1T	1,53	−26,44	1,58	−24,04	1,87	−10,10	2,19	+5,30
	1B	1,07	−48,55	1,03	−50,51	1,09	−47,60	1,90	−8,65

Analysis results of Table 1 shows that with the change radius from 5 to 1 mm, it will be possible to reduce the maximum magnification factor of the blades by −50,5% for the bending mode (1B). With increasing radius up to R = 7 mm, an increase of the maximum magnification factor on +5,30% is obtained for the torsion mode (1T). The influence of rotation on the γ_{max} is essential for the torsional vibrations.

3 Fatigue Life Prediction

Mistuning parameters affect the value of static and dynamic stresses, and the fatigue life. Figure 6a shows the process of fatigue life calculation and the place of mistuning in this process. The calculations on the basis of this diagram are presented in the program package BLADIS+ [7]. Figure 6b displays the schematic diagram of fatigue life calculation. A dynamic stress is determined following the scheme load classification, stress amplitude, number of cycles and fatigue life. An analysis of forced vibrations of one sector of an academic bladed disk (Fig. 1) was carried out. This blade is harmonically excited by 10 nozzles for time period 0–5 s [6, 13]. Figure 6b represents the results of dynamical stresses and estimation of fatigue life in time domain that were obtained using the method of schematization of random loading processes, the Rain flow method and a fatigue life computation according to the Palmgren-Miner hypothesis of one of the previously studied blisks. Figure 7 shows the calculation of the dynamic stress and fatigue life of the model with and without mistuning in the change radius from 5 to 3 mm. Table 2 indicates the values of the blisk fatigue life due to changing fillet radius at the blade root.

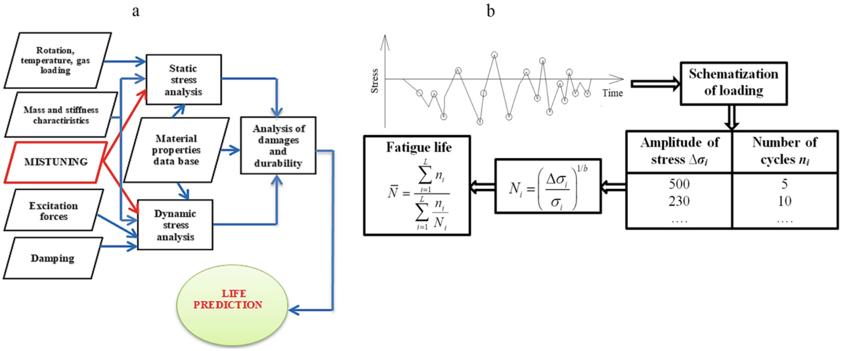

Fig. 6. (a) Mistuning in the life prediction; (b) Diagram of the fatigue life calculation.

Table 2. Calculation of the blisk fatigue life due to changing fillet radius at the blade root with rotational speed (n = 100 1/s)

Block model (Number)	R5 - R3		R5 - R1		R5 - R7	
	$\bar{N}$, *cycles*	%	$\bar{N}$, *cycles*	%	$\bar{N}$, *cycles*	%
1	$1,414 \cdot 10^6$	+2,11	$1,394 \cdot 10^6$	+0,64	$1,361 \cdot 10^6$	−1,72
2	$1,457 \cdot 10^6$	+5,25	$1,442 \cdot 10^6$	+4,12	$1,350 \cdot 10^6$	−2,53

Analysis results of Table 2 shows that the BM 2 of the change radius from 5 to 3 mm gives maximum fatigue life of the blisk, increasing by +5,25%. And the BM2 of the change radius from 5 to 7 mm gives minimum fatigue life, decreasing by −2,53%.

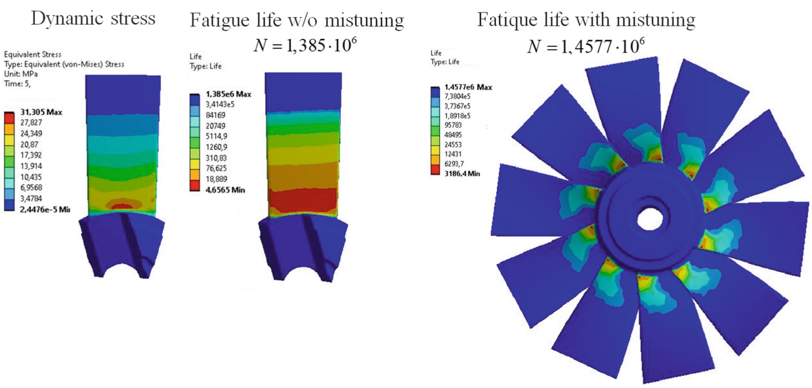

Fig. 7. Dynamic stresses and fatigue of one sector (w/o mistuning) and blisk (BM 2).

4 Conclusion

This paper presents the results of a numerical analysis of the effects of intentional mistuning in order to reduce the maximum amplitude magnification of an axial blisk. Intentional mistuning was implemented by means of small geometric changes namely by changing fillet radii at the blade roots (initial radius R = 5 mm). The results of the study show the reliability and effectiveness of the use of intentional mistuning used for blisks. The blisk fatigue life may be increased by +5,25% due to a reduction of the fillet radius down to 3 mm, and increasing the radius to 7 mm gives a reduction in the blisk fatigue life of −2,53% (BM 2). In addition, variants of introducing intentional mistuning in the form of changing the blade thickness, cropping of the trailing edge and drilling into the blade were investigated. The results of these studies for axial and radial bladed disks will be presented in the following scientific papers.

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