

Detection of blade mistuning in a low pressure turbine rotor resulting from manufacturing tolerances and differences in blade mounting

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Abstract

For a serious prediction of vibration characteristics of any structure a detailed knowledge of the modal characteristic is essential. This is especially important for bladed turbine rotors. Mistuning of the blading of a turbine rotor can appear due to manufacturing tolerances or because of the blading process itself due to unequal mounting of the blades into the disk. This paper investigates the mistuning of the individual blades of a low pressure turbine with respect to the effects mentioned above. Two different rotors with different aerodynamic design of the blades were investigated. The blades were mounted to the disk with a so called hammer head root which is especially prone to mounting irregularities. For detailed investigations, the rotor was excited with a shaker system to detect the forced response behavior of the individual blades. The measurements were done with a laser vibrometer system. As the excitation of rotor structure was held constant during measurement it was possible to detect the line of nodes and mode shapes as well. It could be shown, that the assembly process has an influence on the mistuning. The data were analyzed and compared with numerical results. For this, different contact models and boundary conditions were used. The above described characterization of modal behavior of the rotor is the basis for the upcoming aeroelastic investigations and especially for the blade vibration measurements of the rotor, turning with design and off-design speeds.

Nomenclature

<i>EF</i>	Eigenfrequency	<i>NS</i>	No separation contact
<i>BL</i>	Blade	<i>FL</i>	Frictionless contact
<i>BL-H</i>	Blade model including bore hole	<i>PP</i>	Pure penalty
<i>EXP</i>	Experimental	<i>MPC</i>	Multi point constraint
<i>B</i>	Bonded contact	<i>NL</i>	Normal lagrange
<i>F</i>	Frictional contact	<i>AL</i>	Augmented lagrange

1 Introduction

The determination of the amplification factor due to mistuning is an important task for the safe design of turbomachinery [1]. Rotor disks of turbomachines are designed either to have an intentional mistuning, which is achieved e.g. by small blade modifications or without mistuning. However, if a rotor is designed with identical blades, actually there are small imperfections due to the manufacturing process and tolerances, material inhomogeneity and in service wear. This small imperfections cause mistuning and can increase the forced response vibration amplitudes with respect to that of the ideal case with identical blades. The risk of high cycle fatigue is then increased, therefore mistuning is subject of several experimental and numerical investigations. Ewins [2], Ewins [3], and Castanier and Pierre [4] investigated blade vibrations of bladed disks numerically. Additionally, Castanier and Pierre [4] give an overview of the entire mistuning literature. Also Chan [5] summarises the status of mistuning research.

Khemiri et al. [6] investigated the effect of small mistuning numerically using their asymptotic mistuning model that was successfully used by Martel et al. [7] and Martel and Corral [8].

Because mistuned simulations are extremely costly and time consuming many authors have tried to reduce the size of the computations by means of simplified descriptions of the problem.

For example simple mass-spring systems with just a few degrees of freedom have been frequently used as can be seen in [9], [10], [11], and [12]. The fundamental mistuning model has been introduced in recent years by Feiner and Griffin [13], Feiner and Griffin [14], and Feiner and Griffin [15]. Kaza and Kielb [16], [17], [18] have been the first who investigated the effect of intentional mistuning on flutter.

Heinz et al. [19] investigated experimentally and analytically a low pressure model turbine with intentionally mistuning. Heinz et al. [20] figured out the influence of different mistuning configurations on the circumferential blade amplitude distribution. Wagner [21] has shown that the coupling through the rotor is a very important factor with regard to the blade amplitude distribution in a blade row.

Random mistuning due to manufacturing tolerances is normally treated by means of Monte-Carlo simulations (Beck et al. [22]) or sensitivity analysis (Griffin and Kenyon [23], Petrov [24]). Larin [25] suggested an approach for the analysis of forced vibrations for randomly mistuned bladed disks that is based on stochastic reduced basis method and hypothesis of locally distributed mistuning. In the paper of Hemberger et al. [26] an overview of the work regarding real mistuned bladed structures can be found. Pohle et al. [27] compared numerical and experimental results of a mistuned system and tried to verify the numerical results.

In this work the mistuning pattern due to manufacturing tolerances will be shown. Two different low pressure turbine rotor disks with hammer head blade root design are experimentally investigated and the results are compared with numerical results considering different contact models.

2 Experimental Setup

Two different low pressure turbine rotors (rotor #1 and rotor #2) of the test rig at Graz University of Technology, which have identical overall blade numbers but different geometries are under investigation within this paper. The blades were insert during assembling process into the channel of the rotor disk the and finally fixed with a steel key. Figure 1 shows schematically the design of the hammer head root blading.

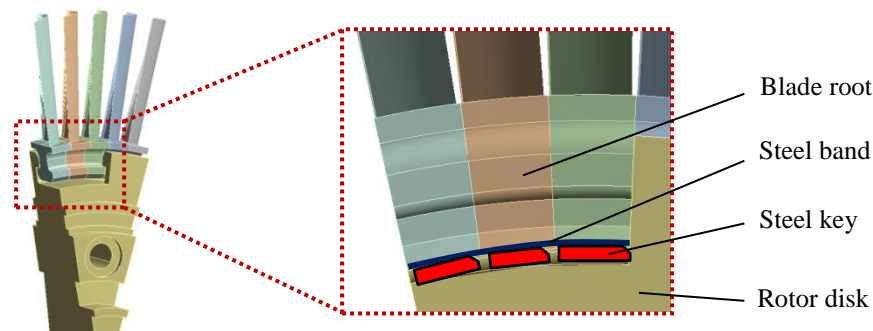


Figure 1: Rotorassembly – Blade fixation

Basically there are a several number of methods for the experimental modal characterization of a structure. Rotor #1 will be characterized by the impact hammer method with roving hammer and compared to the results of shaker tests. The impact hammer characterization shows advantages in relation to a very fast and easy application.

As the shaker test is beneficial for detailed investigation especially regarding mistuning in this case, rotor #2 will be characterized with a shaker.

J.J. Kielb and R.S. Abhari [28] analysed the blade vibratory response with and without aerodynamic loading. They summarized in their conclusion an influence of the aerodynamic damping as part of the total damping. As it is furthermore strongly dependent on the blade root geometries it is not possible to state an influence of aerodynamic damping based to their investigations related to our case, as we are not able to perform vacuum tests up to now.

Due to the fact, that mistuning and vibratory response of the blading has a strong dependency on a several number of influence factors as presented within this paper we assume that aerodynamic influences in our case as part of these investigations are negligible.

2.1 Impact Hammer

The low pressure turbine rotor #1 was bedded on a soft pad to suppress influence of the bedding layer which shall be equal to a free support. Figure 2 shows the test setup where a laservibrometer was placed for the vibration measurement of the blading (structure response). For the modal characterization the rotor was excited via the impact hammer and roving hammer method on different positions on the blades as well as on the disk. The blade vibrations were measured on each blade tip and the measurement executed in 5 iterations as the quality of results is strongly dependent on the hammer impact.



Figure 2: Impact hammer – (Left) Test setup with laser vibrometer, (Right) Excitation with roving hammer

2.2 Shaker

For the shaker tests, rotor #1 and #2 were suspended on a steel frame and coupled with a shaker system shown in figure 3. Again the blade vibration measurements as a result of the excitation were carried out with a laser vibrometer on different positions on the rotor blading and rotor disk as illustrated in figure 4. Basically the input signal is provided by a signal generator which is amplified. As input signal white noise was chosen. In general it is characterized with a constant spectral density within the frequency range. Important for the hanging is the justification of rotor and shaker to avoid additional tilting. This would also result in a wrong excitation of the structure.

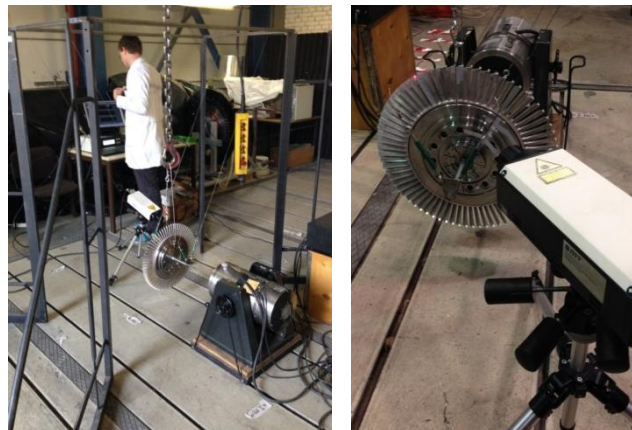


Figure 3: Shaker Test – (Left) Overall test arrangement, (Right) Vibration measurement

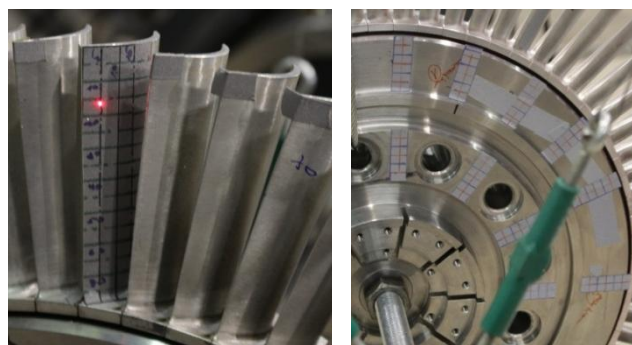


Figure 4: Different measurement positions on the blade (left) as well as on the disk (right)

3 Experimental test results – Impact hammer

The basis for analysing the experimental data is the transfer function which was calculated by the results of the FFT - Analysis of input and output signal. To avoid random noise the results of the input signal FFT and output signal FFT were averaged. Therefore the data are separated in identical segments. These segments are then subject of an individual FFT – Analysis. Finally all data were averaged across all transformations as illustrated in figure 5.

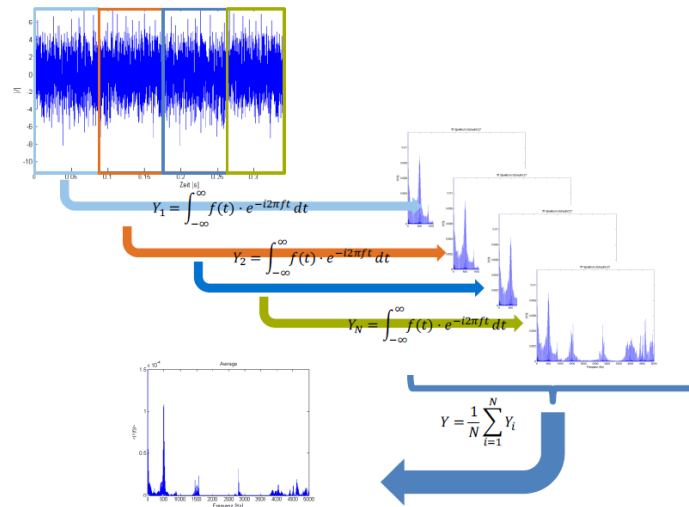


Figure 5: Signal averaging across a number of Fourier transformations

3.1 Blade Eigenfrequencies - Rotor #1

The results of all rotor blades can be seen in the waterfall diagram on the left hand side in figure 6. The same results can be also transferred in a polar diagram shown on the right hand side.

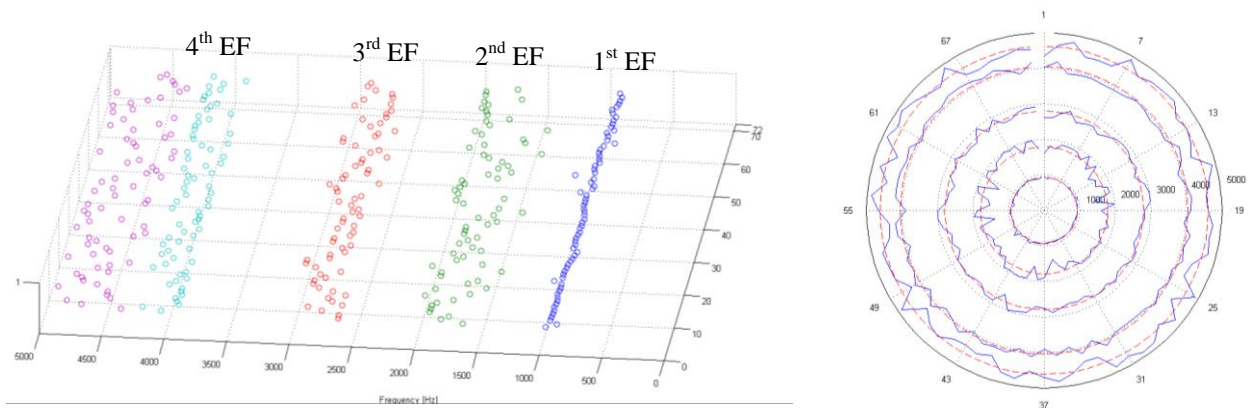


Figure 6: (Left) Waterfall diagram, (Right) Polar diagram of Rotor # 1

Clearly visible are the first 4 eigenfrequencies and the related modes shapes are shown in figure 13. In general there is a high fluctuation range of results which is also characteristic for the impact hammer method. Due to not reproducible impacts of the hammer a deviation of results occurs. An averaged value can be found for sure but furthermore no analysis in respect to mistuning of the low pressure turbine rotor is possible. It also can be stated that the highest fluctuation range can be noticed at the 2nd eigenfrequency. This can be also clearly seen in the polar diagram. In the following chapters it will be shown that the mistuning effect especially influences the 2nd mode, so a tendency can already be noticed with the impact hammer method.

It can be summarized that the impact hammer method for the experimental modal characterization of structures is a very fast method for estimating the modal characteristics. It is characterized by a very easy test setup with the problem of the repeatability which causes high fluctuation ranges of results.

4 Experimental test results – Shaker

Test results of the shaker were analysed as already described at 3. Rotor #1 as well as rotor #2 was under investigation at the shaker tests. Due to a constant input excitation signal the shaker test is predestined for investigations regarding mistuning. All analyses within this paper are focussed on the blade eigenfrequencies. Rotor disk – blade coupling effects were not investigated.

4.1 Blade Eigenfrequencies – Rotor #1

The results of all rotor blades can be seen in the waterfall diagram on the left hand side in figure 7. The same results can be also transferred in a polar diagram shown on the right hand side where a mistuning of the rotor due to manufacturing tolerances and the blade mounting become visible.

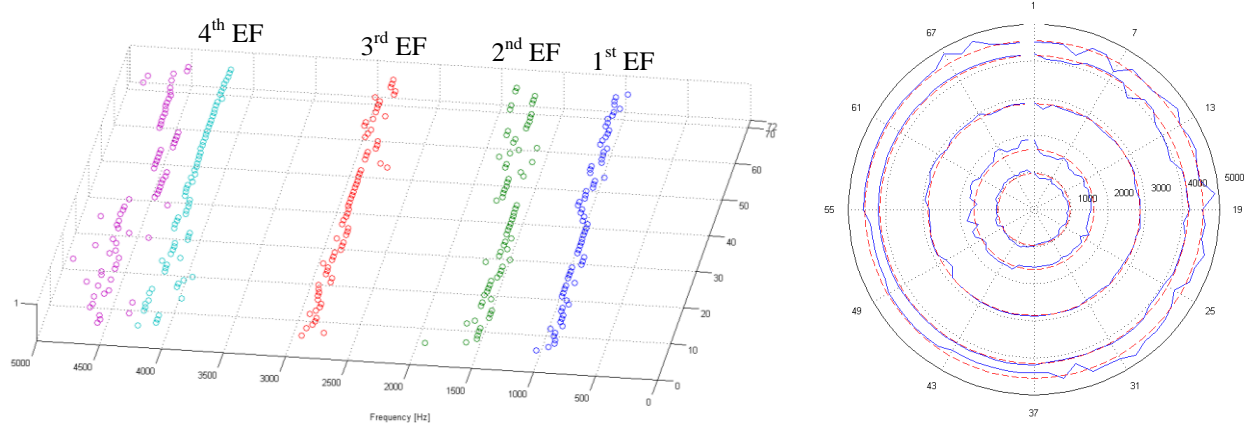


Figure 7: (Left) Waterfall diagram, (Right) Polar diagram of Rotor # 1

Figure 7 shows the first 4 eigenfrequencies. Clearly noticeable in comparison to the results of the impact hammer is a smaller fluctuation range of results due to the constant input signal. By trend again the 2nd mode shows the highest deviations related to the averaged eigenfrequency out of the result of all blades. The polar diagram visualizes this effect as well and also a mistuning especially of the right half of the rotor becomes visible in all modes.

4.2 Blade Eigenfrequencies – Rotor #2

Basically both rotor blade geometries are quite similar. Therefore the results of averaged eigenfrequencies show similar results when comparing figure 7 and figure 8. Nevertheless again the first four eigenfrequencies become clearly visible where the fluctuation range decreases in comparison to rotor #1. Again the 2nd mode shows the highest fluctuation ranges.

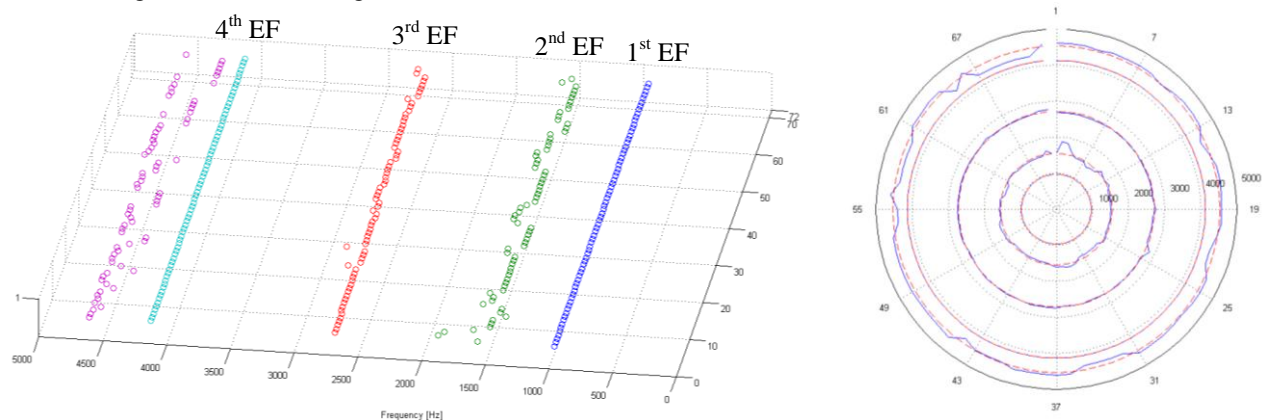


Figure 8: (Left) Waterfall diagram, (Right) Polar diagram of Rotor # 2

When analysing waterfall and polar diagram according figure 8, the mistuning of rotor #2 can be identified as well. In comparison to rotor #1, rotor #2 shows a moderate mistuning. If the tolerances are in the same range, the rotor was manufactured by same supplier; the rotor assembly was improved. As the blade roots are fixed with a key the pre stressing of the blading in fact is not identical.

5 Comparison of the experimental results

Figure 9 shows a comparison between the modal characterization of low pressure turbine rotor #1 with impact hammer method on the left and the shaker test on the right hand side. Average values of the eigenfrequencies show similar results. Conspicuous is the high fluctuation range of the impact hammer due to an uneven excitation process. Both shows the highest fluctuation ranges of mode 2 where the influence of mistuning patterns is high in comparison to all other modes.

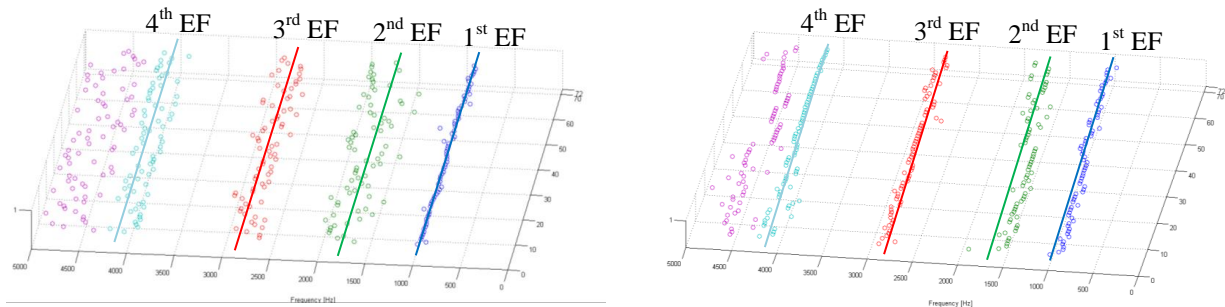


Figure 9: Experimental results: (Left) Impact hammer, (Right) Shaker test of rotor #1

The mistuning effect becomes clearly visible when illustrating the results in a polar plot as shown in figure 10 again for rotor #1. As already seen in the figure 9, mode 2 shows the highest dependency. Due to the geometry of a hammer head root, potential mistuning can be occurred as a result of manufacturing tolerances. Especially the undefined gap of the lateral guidance between rotor disk and blade root, which does exist in fact, results in a mistuning seen in the 2nd EF.

Besides manufacturing tolerances, the blade mounting can cause undefined pre-stressing of the blade root and is another reason for mistuning and is an important factor of the rotor disk – blade coupling. This is furthermore a challenge in the numerical modelling. On one hand the blade is fixed by a key, insert manually and clamping the roots in radial direction. On the other hand the pre-stressing between the blade roots in circumferential direction is an additional result of the clamping process. These effects only can be minimized by an elaborateness assembling process. According figure 10, rotor #1 shows a higher mounting quality on the left half.

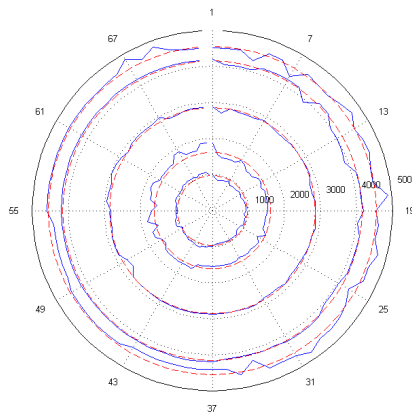


Figure 10: Polar diagram, Rotor #1

Basically the material pairing as 3rd influence of mistuning can be stated. The investigated low pressure turbine rotors #1 and #2 were designed with a rotor disk out of steel and an aluminium blading. The damping of modes furthermore is strongly dependent on this fact.

All in all it is very difficult to separate this several number of influence factors. Neither the manufacturing process nor the assembling process can be performed perfectly.

As concluded above the 2nd eigenfrequency is mostly influenced by mistuning effects. When setting up the numerical model boundary conditions and contact formulations are essential for a serious prediction in general. The following chapter also show the strong dependency of the 2nd eigenfrequency in relation to the boundary conditions of the blade root.

6 Numerical investigations

The numerical analyses were carried out with commercial code of ANSYS 13.0. Different models of the low pressure turbine rotor #2 as illustrated in figure 11 were considered. Starting with a fully assembled turbine rotor model (72BL) also a 1 blade model (1BL), a five blade model (5BL) and different 3 blade models (3BL and 3BL-H) considering varying rigidities of the rotor disk were computed. All models are hexagonal dominated meshed with quad elements. All numerical investigations are at a rotational speed of 0 rpm to be able to compare them with the experimental results. The lack of centrifugal forces which would result in an increase of the tightness of the contact forces is not under investigation within this paper as the experimental results were carried out at latent conditions as well.

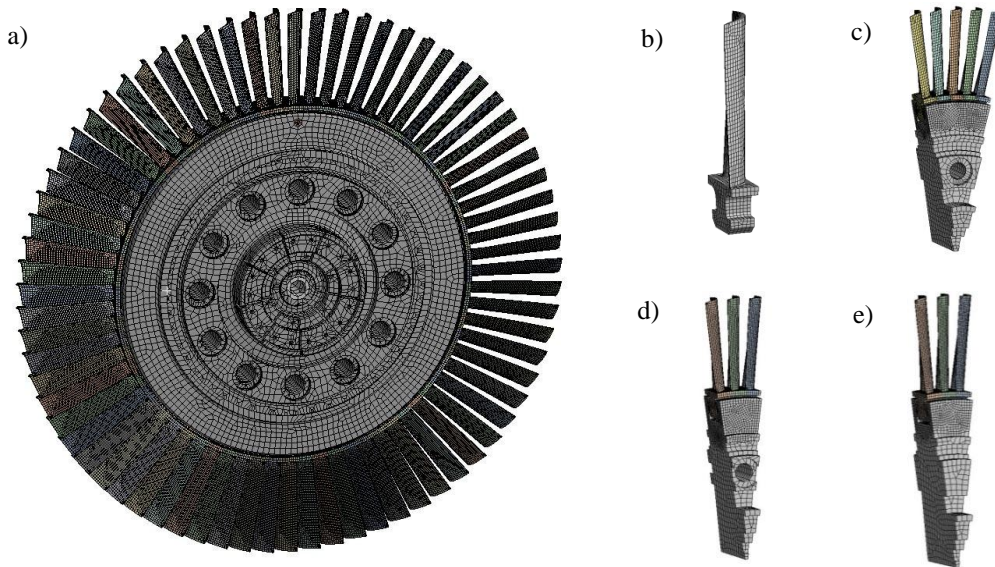


Figure 11: Different FE – Models of the low pressure turbine rotor #2

On selected faces as seen in 12 different contact models and formulation types were applied. For a serious prediction, a face splitting of contact regions as well as a contact sizing are important. The contacts between the blade roots and the lateral guidance of blade root and disk are of particular interest. Mistuning effects are strongly dependent on these contacts and furthermore the eigenfrequencies of related modes are influenced and show the highest deviation in comparison to the experimental results as described at 6.2.

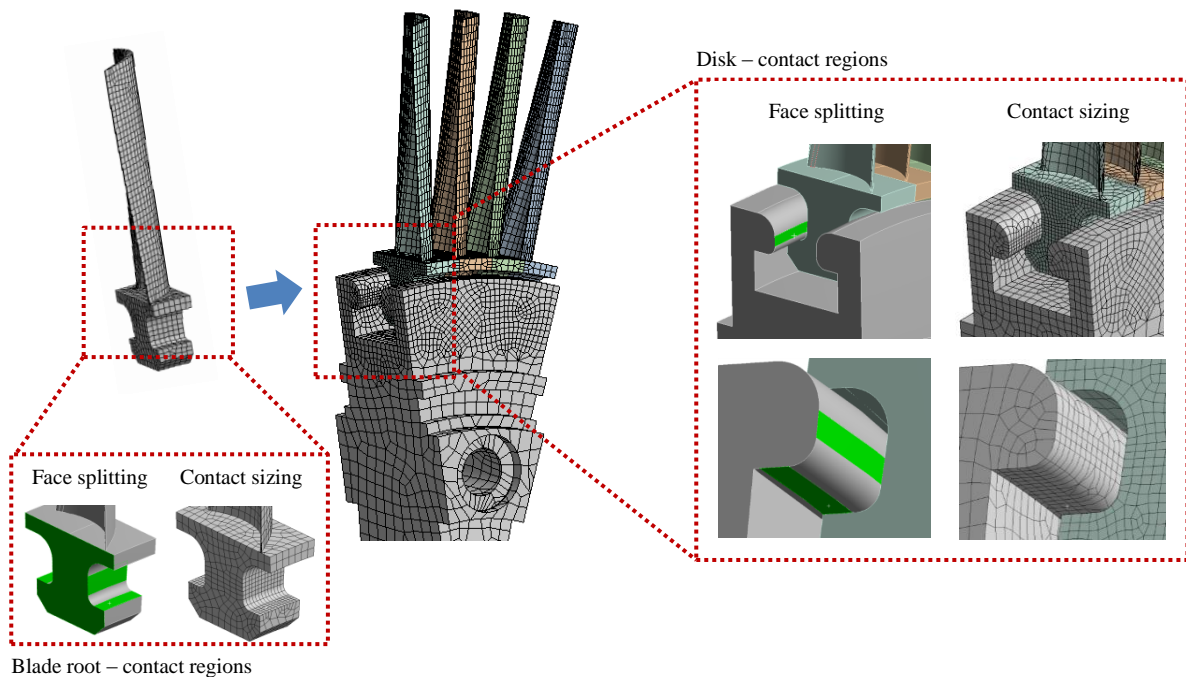


Figure 12: Contact regions and boundary conditions of the FE - Model

6.1 Mode shapes of the rotor blades

Figure 13 shows the mode shapes of the first four eigenfrequencies of the rotor blading of rotor #1 and #2. The first eigenfrequency is the first bending mode in circumferential direction. The second show the first bending mode in flow direction followed by the first torsional mode. The fourth eigenfrequency show the second bending mode in circumferential direction. As all other modes are at higher frequency ranges the result discussion will be focussed on the illustrated modes.

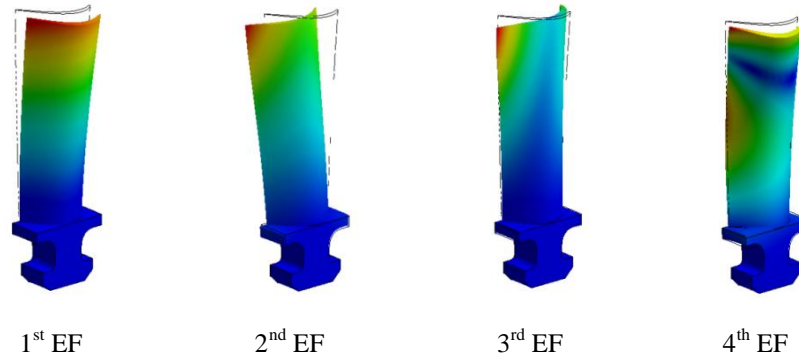


Figure 13: Blade mode shapes

6.2 Contact formulation study

As ANSYS is providing different contact types as well as mathematical formulations, a several number of combinations of these boundary conditions are suitable. All contact types are linearized and therefore non-linear effects were not considered. The main aim was to find a good agreement of numerical results in comparison to the experimental evaluated eigenfrequencies with very simplified numerical models. Table 1 shows the results of the relative deviation of the numerical results in comparison to the averaged experimental values of eigenfrequencies for the first four shape modes and different simplified models as introduced before. All analyses within this paper are focussed on the blade eigenfrequencies. Rotor disk – blade coupling effects were not investigated.

Table 1: Relative deviation between experimental and numerical results

EXP	1BL	3BL-H				3BL				5BL			72BL		
EXP	B	B-MPC	NS-PP	FL-AL	F-AL	B-MPC	NS-PP	FL-AL	F-AL	B-MPC	NS-PP	FL-AL	F-AL	B-MPC	NS-PP
967	-9,4%	-2,8%	-3,5%	-3,6%	-3,0%	-2,8%	-4,2%	-4,5%	-3,2%	-2,8%	-4,2%	-4,2%	-2,9%	1,4%	-10,7%
1548	12,3%	22,1%	18,2%	17,3%	19,9%	23,3%	18,9%	17,7%	20,6%	22,9%	18,4%	18,1%	21,7%	24,9%	21,1%
2716	4,0%	5,6%	4,4%	4,3%	5,1%	6,5%	5,1%	4,7%	5,7%	6,2%	4,8%	4,8%	5,9%	-0,6%	-2,9%
4131	1,3%	-3,7%	-4,4%	-4,5%	-4,0%	5,4%	3,8%	3,4%	5,0%	0,7%	-0,7%	-0,7%	0,5%	2,0%	-5,5%

The results of the 2nd eigenfrequency are not satisfying and would need further investigations. Chosen simplification, especially for the contact of the lateral guidance between rotor disk and blade root are not suitable for a serious prediction of this mode. The fact, that manufacturing tolerances and the blade mounting in respect to different blade root pre-stressing cannot be considered with these models make things worse. As these are basically the reasons of a mistuning of the rotor in reality it is furthermore a challenge in modelling the correct boundary conditions with FEM. Further investigations are necessary and could start with a more realistic contact type in future, such as a non-linear frictional contact followed by considering manufacturing tolerances.

Figure 14 shows the comparison of results representative for the 2nd eigenfrequency. All numerical results show higher values by trend and moreover too high deviations. The 1 blade model (1BL) is very simplified. Depending on the disk rigidity the 3 blade models show similar deviations also in comparison to the 5 blade model (5BL) which is recommended for further investigations such as fluid structure interaction analysis (FSI) where simplified models are required when computational resources are limited. This is also the case for the fully assembled rotor model in this study, where simplified contact formulations results in higher deviation of results in relation to the experimental and the other numerical results as well.

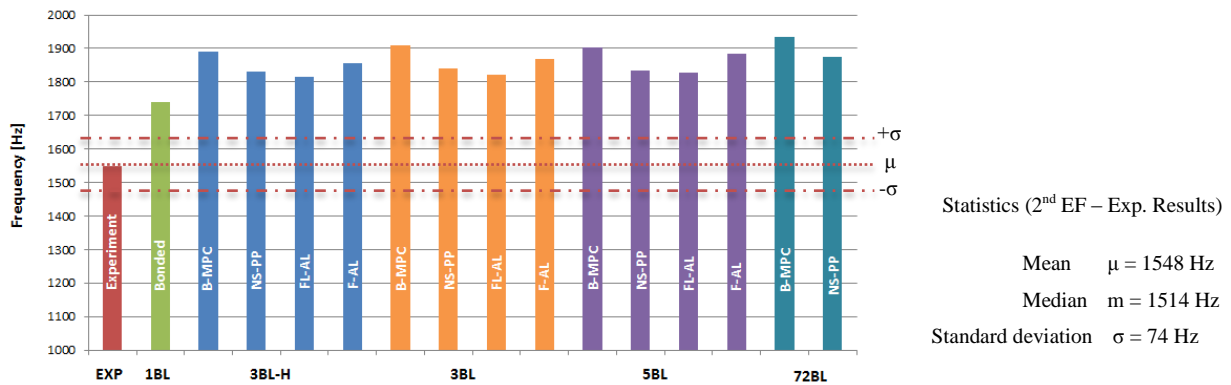


Figure 14: Results for the 2nd Eigenfrequency – Numerical vs. experimental results

If there would be a rating of all investigated simplified models and their results in comparison to the experimental data it can be summarized that the 1 blade model shows very good preliminary results with a fast setup including the weakness of maybe too simplified boundary conditions. The 3 blade model for this turbine rotor geometry is problematic, because of modelling correct disk rigidities. For further investigations such as FSI analysis the 5 blade model is recommend which represents correct model characteristics after all simplifications.

7 Conclusion

This paper presented different experimental and numerical methods for the modal characterization of structures carried out on a low pressure turbine rotor. Impact hammer tests show good preliminary results with a very fast test arrangement and execution. Shaker tests are especially required for detailed analysis regarding mistuning. Mistuning is strongly influenced by manufacturing tolerances, the blade mounting as well as of the material pairing of components. Especially the second eigenfrequency is strongly influenced due to various mistuning effects. This can be also seen in the numerical analysis, where simplified contact modelling of the relevant contact region shows no satisfying results. Furthermore simplified numerical models with different contact types as well as with mathematical formulations were presented. All in all these numerical models show a good agreement with the experimental data.

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