Forced Response Reduction of a Compressor Blisk Rotor Employing Intentional Mistuning

B. Beirow, A. Kühhorn and J. Nipkau

Abstract Using the example of a compressor test blisk with 29 blades different sources of mistuning and their consequences for the forced response are analysed under consideration of aeroelastic effects. In particular the impact of superimposing intentional structural mistuning by both random structural mistuning and aerodynamic mistuning is studied. For this purpose reduced order models of the blisk are adjusted for different mistuning distributions. The mistuning itself is characterized by assigning individual stiffness parameters to each blade. The aeroelastic coupling is included employing aerodynamic influence coefficients. By means of genetic algorithm optimizations, structural mistuning patterns are found which yield a mitigation of the forced response below that of the tuned design reference. Ideally a nearly 50 % reduction of maximum response magnitudes is computed for the fundamental bending mode and large mistuning. The solutions found have been proven to be robust with respect to additional random and aerodynamic mistuning in case of large intentional structural mistuning.

Keywords Blade vibration · Blisk · Mistuning · Forced response · Compressor

1 Introduction

The fabrication of aero-engine compressor rotors as one piece has become increasingly significant in recent years, since it allows for higher rotational speeds associated with higher pressure ratios and an enhanced efficiency. However, regarding the forced response computation of these blade integrated disks (blisks) due to aerodynamic excitation, engineers are exposed to a number of particularities.

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Apart from the extremely low structural damping level of blisks due to the lack of frictional damping, small but unavoidable differences of mechanical characteristics from blade to blade, which are denoted as mistuning can cause severe amplifications of the forced response compared to the ideal design with identical blades. Typical sources of mistuning are geometric deviations due to manufacturing tolerances, wear, damage or even strain gauge instrumentation. Engineers have been concerned with the mistuning phenomenon of bladed disks for about 50 years. As long ago as 1966 Whitehead [1] introduced a theoretical limit for an estimation of the maximum displacement amplification only depending on the number of blades. Martel and Corral [2] formulated a modified and less conservative limit in which the number of blades is replaced by the number of active modes in order to take into account the degree of modal coupling within a family of blade modes. In addition Figaschewsky and Kühhorn [3] assume normally distributed individual blade frequencies with a chosen standard deviation of mistuning. In doing so the mistuning strength is taken into account generally yielding a more realistic calculation of the forced response amplification due to mistuning. Petrov and Ewins [4] used optimization algorithms to find the worst forced response of bladed disks in terms of academic studies. However, the majority of analyses measured or preset mistuning patterns yielded amplification factors from 1 to hardly greater than 2 as exemplarily reported in [5].

Aiming at a reduction of vibration amplification amplitudes one idea has been to design blisks with intentional mistuning. Here the intention is to take advantage of the dependence of aerodynamic modal damping ratios on the inter-blade phase angle, which can yield an increased resulting damping level and lead to a reduction of the forced response level for particular engine order excitations even below that of the tuned counterpart [6]. In the current paper the effect of intentional mistuning is addressed with respect to a mitigation of the forced response. Reduced order models based on the subset of nominal system modes (SNM) [7] are employed in which mistuning is quantified by stiffness variations. Aeroelastic coupling effects are considered employing the method of aerodynamic influence coefficients (AIC) which are put into the SNM-model as described in [8]. Mistuning patterns which are derived from genetic algorithm optimizations are exemplarily analyzed for a 29-bladed high pressure compressor blisk rotor (Fig. 1a) with focus on the

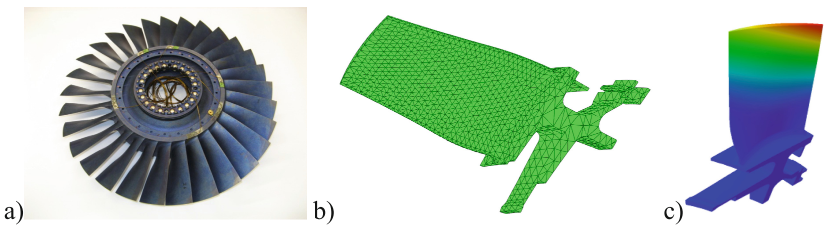

Fig. 1 **a** High pressure compressor blisk, **b** finite element sector model and **c** fundamental blade mode shape (1st flap)

fundamental blade bending mode. Additional random mistuning is superimposed in order to evaluate the robustness of the forced response reduction. Finally, the effect of aerodynamic mistuning is analysed.

2 Numerical Model

Aiming at a preparation of a numerical model for the forced response computation, a finite element sector model is set up in a first step representing the tuned and cyclic symmetric blisk with identical blades (Fig. 1). Subsequently an eigenvalue analysis is carried out in order to gain the relevant information about the basic vibration characteristics and blade mode families. Focusing on the frequency range around the first fundamental blade mode family, a basic SNM-model with just 31 degrees of freedom is derived. This basic modal model can be easily adjusted to arbitrary blade frequency mistuning by means of a stiffness adjustment of every blade, for details please refer to [7, 8]. These models are valid as long as no change of blade mode shapes appears. Hence, the SNM is well suited for quick forced response computations e.g. within probabilistic analyses.

Assuming small blade displacements and linear conditions, the AIC-technique is well suited to cover effects of fluid structure interaction in a simple way [8]. The computation of AIC starts with unidirectional coupled CFD-/FEM-calculations for an assembly of identical blades and stationary flow conditions. Only one blade, the reference blade, is forced to vibrate in a particular blade mode causing flow disturbances and consequently unsteady surface pressure distributions on all blades in the assembly. Subsequently, the modal forcing acting on each blade is computed and normalized with the modal displacement of the reference blade. This normalization yields the AIC which finally have to be transformed from blade individual coordinates into coordinates of the subset. In this way the whole aeroelastic information is considered in the aerodynamic impedance matrix $\mathbf{Z}$ appearing in the equation of motion as follows:

$$[-\Omega^2\mathbf{M} + j\Omega\mathbf{D} + \mathbf{K} + \mathbf{Z}]\mathbf{q}(j\Omega) = \mathbf{F}^F. \tag{1}$$

$\mathbf{M}$, $\mathbf{D}$ and $\mathbf{K}$ denote modal mass, modal structural damping and modal stiffness matrices, $\mathbf{F}^F$ the vector of external modal forcing and $\mathbf{q}$ the vector of modal displacements. If external forcing and structural damping are neglected, Eq. (1) represents an eigenvalue problem which yields the aeroelastic natural frequencies and aerodynamic damping values given in Fig. 2. Note that if a tuned blisk is excited in a particular engine order only one mode characterized by a particular number of nodal diameters (ND) and assigned to one aerodynamic damping ratio is responding. In case of a mistuned blisk the response is composed by a linear combination of several ND-modes so that the resulting aerodynamic damping deviates from the 'regular' value of a pure response in only a single ND.

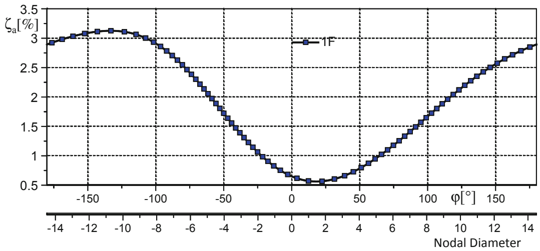

Fig. 2 Aerodynamic damping versus inter-blade phase angle φ and nodal diameter (maximum take off conditions)

3 Real and Intentional Structural Mistuning

The real mistuning distribution has been experimentally determined for the blisk shown in Fig. 3a by use of a patented approach via blade by blade ping tests [6]. It was found that the blade to blade stiffness variation is ranging between ±0.75 %. However, the maximum forced response amplification of the tuned reference is appearing moderately with an 8 % (γ = 1.08 at EO 26) rise in the worst case (Fig. 3c). On the contrary large alternating intentional mistuning (ΔE = ±6.09 %, Fig. 3b) comes along with tremendous differences (Fig. 3c): extreme values are a 115 % rise (γ = 2.15 at EO 13) and a drop of about 40 % (γ = 0.60 at EO 25). Such a behavior is explainable by a modified aerodynamic damping level. In the latter case an EO 25 corresponds to an aliased EO-4 excitation, which excites a ND 4 mode in case of a tuned system. However, since strong mistuning is present, other NDs are involved which contribute more aerodynamic damping (Fig. 2) and hence strongly increase the resulting aerodynamic damping level. Obviously this increased damping contribution works against the commonly response amplifying

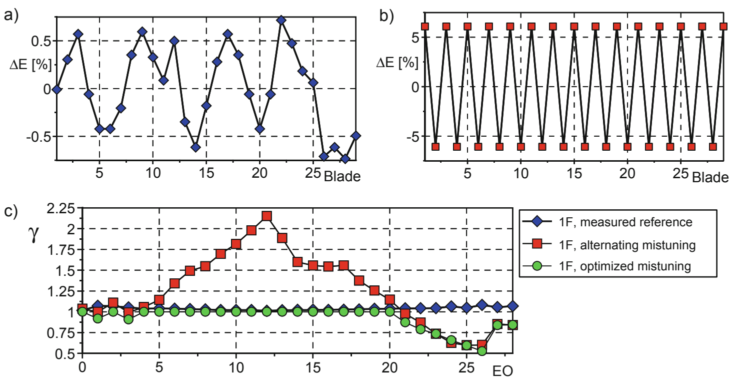

Fig. 3 **a** Measured and **b** alternating mistuning patterns, **c** maximum amplification of mistuned forced response versus engine order (EO)

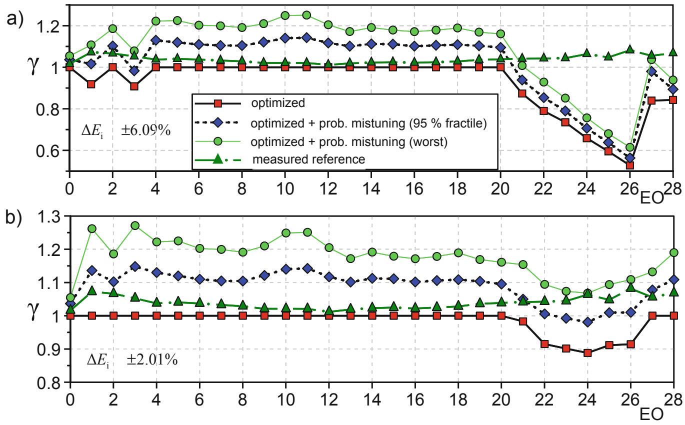

Fig. 4 Amplification of max. blade displacements—optimized intentional mistuning combined with additional random mistuning (50000 samples): **a** large, and **b** small intentional mistuning

effect of mistuning and even causes a mitigation of forced response below that of the tuned counterpart. Similar results are achieved with intentional mistuning patterns ($\Delta E = \pm 6.09$ % allowed) being individually optimized for each EO via genetic algorithms [6]).

Uncertainties within the manufacturing process cannot be avoided even in case of intentional mistuning. For that reason it is assumed that additional and evenly distributed random mistuning ($\Delta E = \pm 1$ %) will be present in order to prove the robustness of the optimized intentional mistuning patterns to reduce the forced response. Again considering large intentional mistuning ($\Delta E = \pm 6.09$ %), the gain achieved for mistuning patterns optimized with respect to EO-excitations from 21 to 28 is the same since the responses remain always below that of the measured and largely below the tuned reference (Fig. 4a). However, this is not the case for small intentional mistuning ($\Delta E = \pm 2.01$ %) where the benefit of the optimization gets widely lost and hence the robustness is not given (Fig. 4b).

4 Aerodynamic Mistuning

Apart from structural mistuning additional aerodynamic mistuning is considered via small random perturbations of AICs. Hence, small perturbations of features like blade stagger angles or the distance of adjacent blades are taken into account. Similar to the approach presented in [9] random perturbations are included in the impedance matrix of Eq. (1) considering three different maximum random deviations: 1, 3 and 10 %. In combination with large intentional mistuning patterns

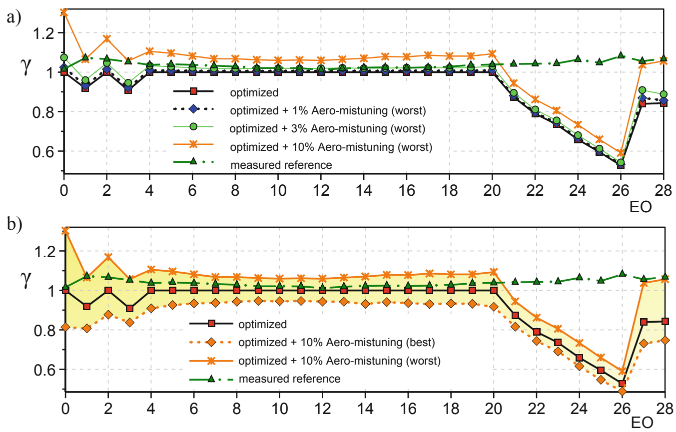

Fig. 5 Amplification of maximum blade displacements—optimized intentional mistuning combined with additional aerodynamic mistuning (50000 samples): **a** varying, **b** 10 %

($\Delta E = \pm 6.09$ %) 1 and 3 % aerodynamic mistuning only marginally affects the forced response (Fig. 5a), which agrees with the findings shown in [9]. Whereas allowing 10 % aerodynamic mistuning both, a partly significant rise and drop of the forced response may occur (Fig. 5b). Nevertheless, the gain achieved with optimized, large intentional mistuning is retained between EO 21 and 28, since the maximum response amplification is never exceeding that of the measured reference.

5 Conclusions

The use of intentional mistuning has been analysed for a compressor blisk rotor in terms of a mitigation of the forced response. It could be shown that even a response reduction beneath the level of the tuned counterpart design is possible for particular engine order excitations for optimized mistuning patterns. In case of large intentional mistuning the potential of encouraging reduced forced response levels has been proven to be robust with respect to additional moderate but realistic random structural and aerodynamic mistuning.

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