



Vibration Reduction of a Steam Turbine Wheel by Means of Intentional Mistuning

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Abstract. A last stage steam turbine wheel is analyzed with the objective to alleviate the flutter susceptibility by employing intentional mistuning (IM). In particular, the operation at nominal speed under part-load conditions may cause unfavorable flow conditions facilitating flow separation. In consequence, negative aerodynamic damping ratios occur for the first bending mode family in some circumstances. Employing intended alternate mistuning of adequate magnitude has proved to be a promising measure to stabilize rotors in terms of avoiding self-excited vibration phenomena. From the manufacturing point of view, this two-blade design is advantageous as well and hence, chosen here as a first measure to attenuate flutter susceptibility. Two prototypes of bladed disks series have been made, which are exhibiting small but unavoidable deviations from the design intention due to manufacturing. The real blade alone frequencies have been identified within foregoing experimental investigations. Numerical modal analyses carried out for the prototypes as manufactured finally reveal that there is an additional positive contribution of random mistuning in terms of further enhancing the least aerodynamic damping ratio. Another promising and robust IM pattern is found by using generic algorithms to optimize the least aerodynamic damping ratio yielding stable conditions at any time as well. Moreover, it shows that IM combined with random mistuning also mitigates the maximum forced response at part-speed conditions.

Keywords: Intentional mistuning · Blade vibration · Flutter · Forced response · Optimization · Aeroelastics

1 Introduction

Commonly, bladed disks and blade integrated disks of turbine and compressor applications are designed as cyclic symmetric structures featuring identical blades. However, actually the wheels are exhibiting small variations from blade-to-blade, and consequently different mechanical properties of each blade. This characteristic is known as mistuning, which typically results from manufacturing, wear, damage, repair measures or even strain gauge instrumentation. On the one hand, mistuning has a beneficial effect on flutter suppression [1]. On the other hand, unfortunately, unavoidable random mistuning may cause a severe magnification of the maximum forced response compared to the tuned counterpart with identical blades. Already more than 50 years ago, Whitehead

[2] or Ewins [3] contributed trendsetting publications addressing this issue. Until today hundreds of papers dealing with the mistuning phenomenon have been published, which indicates the significance of the problem.

Over the years, it was found that the application of intentional mistuning (IM) is well suited to mitigate the susceptibility of the forced response towards random mistuning. Castanier and Pierre [4] conducted comprehensive structural analyses with a focus on harmonic IM patterns without considering aerodynamic coupling for both academic blisks and an industrial blisk with 29 blades. They proved a clearly enhanced robustness towards large forced response amplitudes for both cases. Lim et al. [5] continued this work for the industrial blisk by considering other set-ups of IM patterns, each effecting a mitigation of the forced response compared to the blisk without IM. Han et al. [6] proved that optimized IM patterns composed by only two different blade types A and B are also suitable to attenuate the susceptibility of the maximum forced response to random mistuning.

Novel strategies target on increasing the aerodynamic damping by employing IM in order to decrease the maximum forced response even below that of the reference design with identical blades. According to the publications of Petrov [7] and Schoenenborn et al. [8] favorable conditions are required with respect to the dependence of aerodynamic damping on the inter blade phase angle and the engine order (EO) to be considered. Martel and Sánchez-Álvarez [9] have presented a successful forced response reduction by means of IM superimposed with random mistuning for different academic blisks. Furthermore, they introduced a theoretical limit of the achievable forced response reduction depending on the exciting EO, the dedicated aerodynamic damping and the mean value of aerodynamic damping. The practical implementation of an optimized IM pattern is addressed in [10] for a turbine blisk by means of varying fillet radii, which affects a 70% reduction of the maximum forced response caused by low engine order excitation of the fundamental bending mode. Figaschewsky et al. [11] demonstrated another practical application of IM for a blisk fan. Both forced response caused by low engine order excitation and flutter susceptibility of the first blade mode could be reduced crucially.

Among others, Miyakozawa [1] addressed the increasing effect of IM for stabilizing aerodynamically unstable rotors. The same applies to the work of Kaza and Kielb [12] who carried out aeroelastic analyses for a high-aspect ratio turbofan, which demonstrated that alternate mistuning has the potential to alleviate flutter problems. Equally, Srinivasan [13] exemplarily showed that an originally tuned but unstable rotor in terms of torsional flutter is stabilized due to the presence of mistuning. The case study of Kielb et al. [14] also demonstrates the positive effect of mistuning on flutter suppression, which becomes more effective with increasing mistuning magnitude. Martel et al. [15] employed their well-established asymptotic mistuning model to enhance the stability of two aerodynamically unstable low-pressure turbine rotors by means of optimized intentional mistuning patterns. Considering a tip-shrouded bladed disk, Corral et al. [16] presented the experimental evidence of how alternate mistuning is applicable to mitigate or even completely remove flutter instabilities.

The present paper focuses on stability analyses of the last stage low-pressure steam turbine wheel highlighted in Fig. 1 with respect to flutter suppression aided by IM. The

academic case study makes use of a modally reduced structural model introduced by Yang and Griffin [17], which is known as subset of nominal system modes (SNM). The SNM easily allows for considering both frequency mistuning and aeroelastic interaction. Alternate IM is chosen first with the objective to stabilize the originally unstable wheel for the first bending mode (1B) at nominal speed and part-load conditions. The stability is assessed based on occurring negative aerodynamic damping ratios and indeed, it becomes apparent that the flutter susceptibility could be enhanced in terms of a satisfying and robust solution from the engineering point of view, i.e. even the least aerodynamic damping ratio takes a positive value. Basically, further improvements are achievable by employing an IM pattern optimized by means of genetic algorithms even if still only two different blade designs are admitted in order to keep the manufacturing efforts small. However, the solution found turns out to be less robust compared to alternate IM. Further analyses are addressing the impact of mistuning magnitude. Beyond that, complementary analyses prove that IM superimposed with random mistuning design has a positive impact on the maximum forced response compared to the reference design with identical blades.

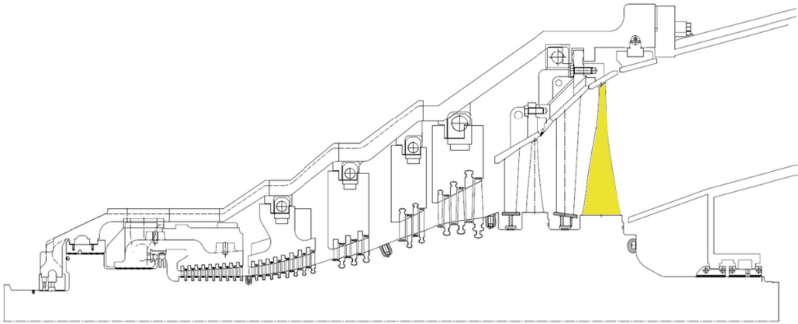


Fig. 1. Sectional drawing of industrial steam turbine (last stage highlighted)

2 Preliminary Investigations

2.1 Structural Analyses of the Reference Wheel with Identical Blades

Both identifying modal characteristics of the turbine wheel with 51 blades and preparing input data for the following reduced order modelling require to set up a finite element (FE) sector model, which features 43.000 tetrahedral elements. As long as mode shapes are dominated by blade motion and the strain energy is concentrated in the blades, the modes are grouped in blade mode families. The first three of them (Fig. 2, right) are named conformable to their appearance with first bending mode (1B), second bending mode (2B) and first torsional mode (1T). Focusing on the first bending mode, the sector mode shapes of the 1B-family are used to set up SNM-models with only 51 degrees of freedom, which are consequently only valid for the frequency range of the 1B. Three different models are created due to the speed dependence of natural frequencies revealed

in the Campbell plot of Fig. 2, namely for 100% speed, where self-excited vibration may occur at part-load conditions, and for 30% and 50% speed, where resonance crossings may cause large forced response.

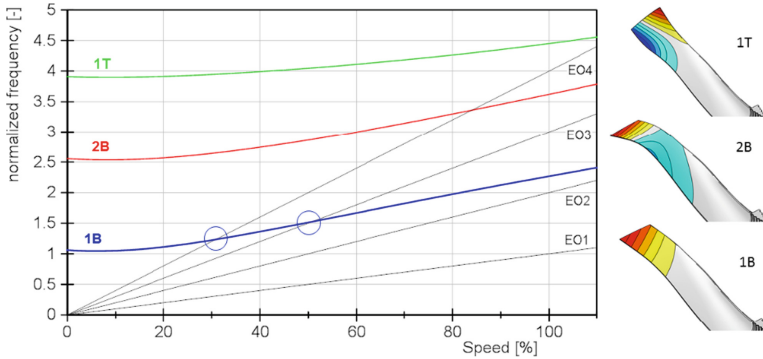


Fig. 2. Campbell diagram involving the first three blade modes

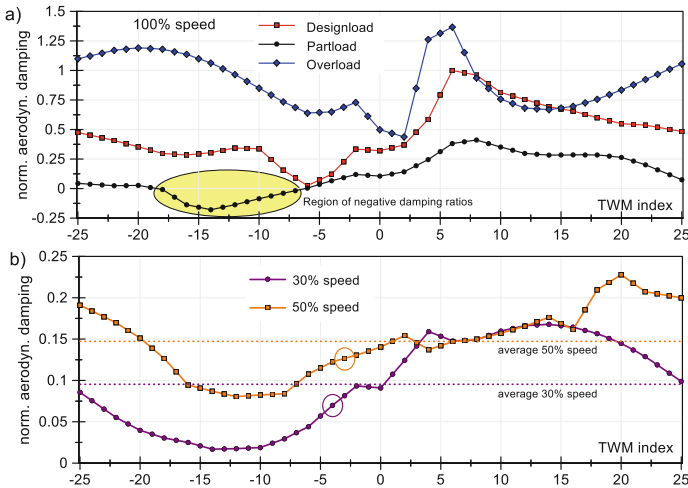


Fig. 3. Travelling wave mode dependent aerodynamic damping a) 100% speed (nominal), and b) part-speed (30% and 50%)

2.2 Aeroelastic Simulation

Starting with 2D computations of the whole steam turbine, steady-state CFD analyses are carried out to identify the flow boundary conditions for the second step, 3D computations of the last two turbine stages only. Circumferentially periodic boundary conditions and the two-equation $k-\omega$ -SST turbulence model [18] are chosen for last-named analyses. Fillets and tip gaps are neglected to keep the simulation as simple as possible. Results

are calculated at nominal speed (100%) for three different pairs of condenser pressure and exhaust velocity: (i) part load, (ii) design load, and (iii) over load. In contrast, only one relevant load condition each is considered for 30% and 50% part-speed conditions.

Aeroelastic eigenvalues of the last stage are determined by means of single passage computations using the results of steady state computations to define both initial conditions and boundary conditions. Structural natural frequencies and mode shapes are carried over from FE-analyses. A time-linearized flow solver is used to provide aerodynamic damping values and aeroelastic natural frequencies. Since natural frequencies of the 1B blade mode family hardly change due to the impact of aeroelastic interaction, the following discussions are focused on travelling wave mode (TWM) dependent aerodynamic damping ratios. According to the yellow highlighted region in Fig. 3a, negative damping ratios are occurring at part load conditions and 100% speed for TWM between -18 and -7 , so that unstable conditions may take place for this operating point. Increasing condenser pressure and exhaust velocity take a positive effect, nevertheless the least damping ratio at design load is approaching the stability margin whereas overload conditions prove to be always stable. As previously announced IM is employed to ensure safe operation at any load condition and time.

Since IM will affect the forced response at part-speed conditions as well, additional simulations are necessary for speeds close to resonance crossings highlighted in Figs. 2 and 3b (1B excitation in EO 3 and 4). Comparatively low aerodynamic damping ratios occur for negative or backwards travelling waves, respectively. Consequently, the damping ratios dedicated to regularly excited TWMs -3 and -4 are located beneath the average damping lines so that IM offers the opportunity to increase the resulting aerodynamic damping by coupling in higher damped TWMs [9]. Hence, the commonly response magnifying impact of mistuning can be stemmed or even inverted [10].

2.3 Speed Dependence of Structural Mistuning

The manufacturer has chosen alternate IM first since it has proved to be effective for flutter suppression in the past. That is why two different sets of blades have been produced, which are arranged in IM patterns denoted as A and B. Foregoing bonk tests have shown, that these patterns feature small deviations from the design intention (Fig. 4). However, the blade-to-blade frequency differences change due to the speed dependence of natural frequencies. In consequence, a speed-dependent correction factor $c(\Omega)$ is multiplied to relative mistuning patterns $\Delta f_i(0)$ determined at rest:

$$\Delta f_i(\Omega) = c(\Omega) \cdot \Delta f_i(0) \quad (1)$$

The determination of $c(\Omega)$ requires FE modal analyses of the wheel with alternate IM taking into account speed and stiffening effects. Next, SNM models are employed to adopt the mistuning magnitude as long natural frequencies of FEM and SNM are matching. This procedure is repeated for every speed of interest. Figure 5 reveals the tremendous impact of speed on alternate IM yielding a reduction of mistuning magnitude up to 63% at 100% speed compared to the resting wheel. As a side note, the popular correction suggested by Feiner [19] and by Nipkau [20] later on overrates the speed impact on mistuning in this particular case.

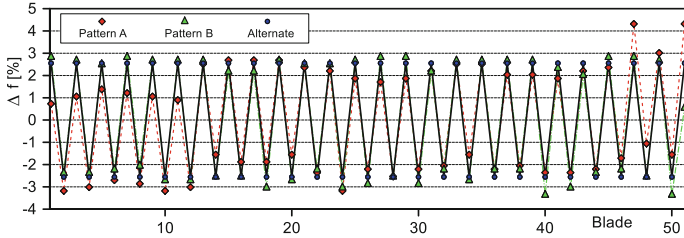


Fig. 4. Alternate frequency mistuning at rest: As manufactured (A and B) and original

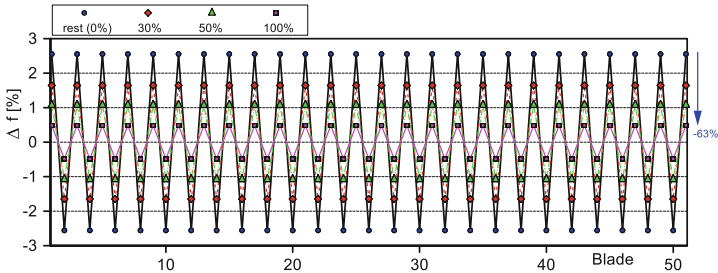


Fig. 5. Alternate frequency mistuning: Impact of speed

3 Stability Analyses

3.1 Alternate Intentional Mistuning

Previously shown aerodynamic damping curves (Fig. 3a) reveal that negative damping ratios become apparent at 100% speed under part load conditions. First, alternate IM is employed with the objective to stabilize the rotor at any load condition. The success of this measure is evaluated based on the least aerodynamic damping ratios, which are computed together with aeroelastic natural frequencies by solving the aeroelastic SNM eigenvalue problem. Starting with design load conditions (Fig. 6), a number of particularities become apparent:

- (a) Pure alternate mistuning causes a splitting of the tuned frequency group (squared symbols) into two groups, one located at greater frequencies and one located at lower frequencies (triangles).
- (b) Additional random mistuning as in case of patterns A and B affects first, a widening of the frequency range of both groups, and second, that the co-domain of aerodynamic damping ratios is significantly narrowed.
- (c) In general, mistuning causes an increase of the least aerodynamic damping ratio, which is highlighted by circle marks.
- (d) Alternate IM superimposed with random mistuning due to manufacturing (patterns A and B) yields an insignificantly greater increase of the least aerodynamic damping ratio compared to pure alternate mistuning.

Moving on to part load conditions (Fig. 7), the observations made before prove to be true. Moreover, alternating mistuning increases the least aerodynamic damping ratio such that completely stable conditions are achieved i.e. the system does not feature negative damping ratios beneath the stability margin anymore. Additional random mistuning affects further stabilization as indicated by patterns A and B.

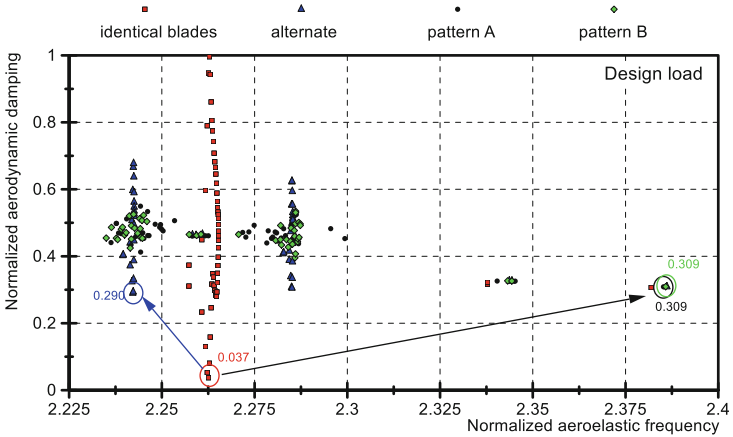


Fig. 6. Aeroelastic eigenvalues at design load conditions and 100% speed (Alternate IM)

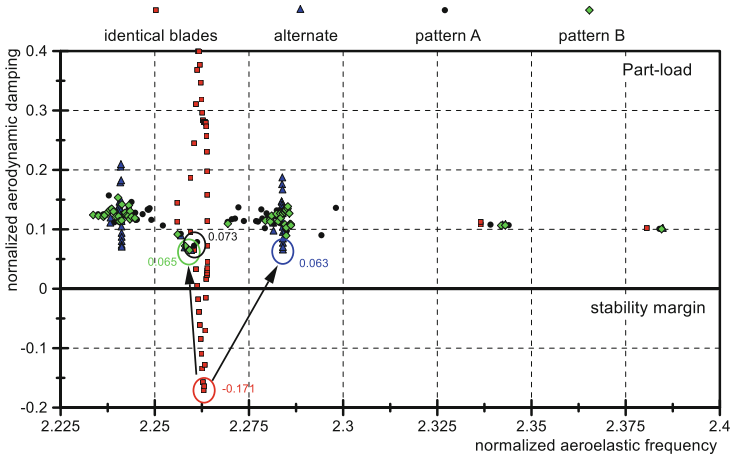


Fig. 7. Aeroelastic eigenvalues at part load conditions and 100% speed (Alternate IM)

3.2 Mistuning Magnitude, Optimized Mistuning and Robustness

The impact of both, IM magnitude and additional random mistuning on flutter suppression is analyzed focusing on part-load conditions. Starting with alternate IM, the solid

red line in Fig. 8a reveals that positive damping ratios or stable conditions, respectively, are obtained for frequency mistuning standard deviations ranging from $\Delta f = 0.18 \dots 5\%$. If additional random mistuning of $\Delta f = \pm 1\%$ is superimposed in terms of Monte Carlo analyses (40000 samples per standard deviation) the solutions remain always stable with respect to the 1% percentile between 0.42 and 5% standard deviations. If the standard deviation is greater than 1% a positive impact of additional random mistuning becomes apparent. The choice of block by block IM with two, three or four blades per package do not improve flutter susceptibility compared to alternate IM. The three (AAABBB...) or four blades per package solutions even yield unstable conditions in terms of negative least aerodynamic damping ratios at any time. Figure 8b shows, that a pure optimized IM pattern (solid red line) could further increase the least aerodynamic damping ratio compared to pure alternate IM. However, again superimposing 1% random mistuning reveals that the 1% percentiles are clearly less than those of alternate IM. Nevertheless, the least aerodynamic damping ratio never falls below the stability margin if the basic standard deviation of optimized IM is greater than 0.43%.

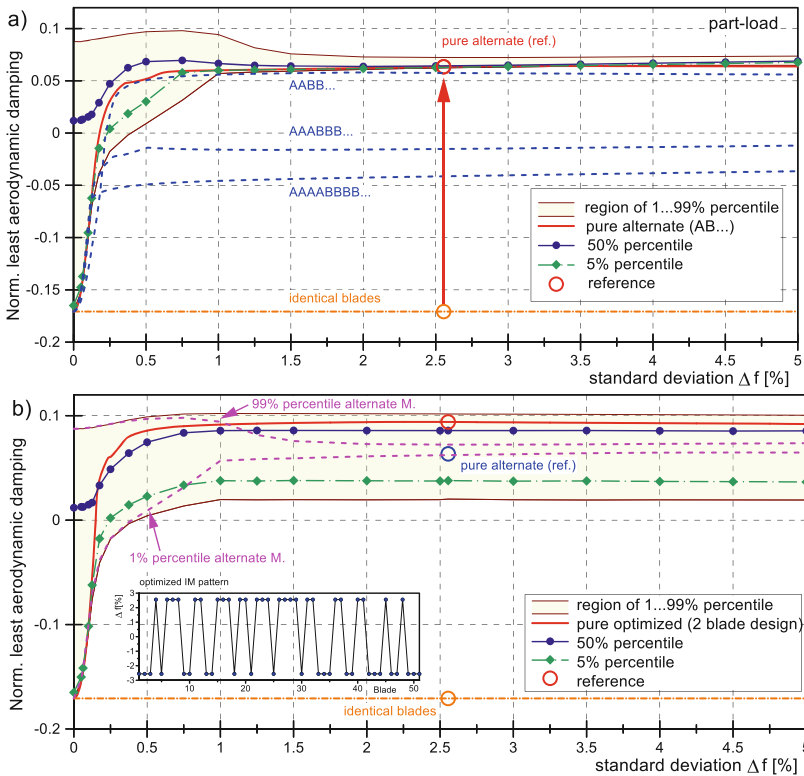


Fig. 8. Impact of mistuning magnitude superimposed by $\Delta f = \pm 1\%$ random mistuning on least aerodynamic damping, a) alternate IM, and b) optimized IM.

4 Forced Response

Since two resonance crossings occur at part speed conditions (Fig. 2, engine orders 3 and 4) forced response analyses are carried out considering the various IM patterns. Again, the impact of random mistuning is taken into account in terms of probabilistic analyses. The results given in Table 1 illustrate first that the tuned reference with nominally identical blades suffers a maximum displacement amplification (99% percentile) of up to 115.9% (EO 3) due to the impact of small random mistuning. IM patterns superimposed with random mistuning exhibit smaller magnification factors compared to the tuned design intention. In particular, alternate IM give favorable results with tremendously lesser magnification factors (47.6% at EO3). Hence, the IM approach has positive effect on both, flutter susceptibility and maximum forced response.

Table 1. 50% and 99% percentiles of maximum displacement magnification superimposed with $\Delta f = 0.5\%$ random mistuning (10000 samples)

Case	EO 3 ($\gamma_{50\%}$)	EO 3 ($\gamma_{99\%}$)	EO 4 ($\gamma_{50\%}$)	EO 4 ($\gamma_{99\%}$)
Identical blades	1.631	2.159	1.276	1.869
Alternate IM	1.212	1.476	1.001	1.210
Optimized IM	1.332	1.644	1.061	1.322

5 Conclusions

The case study could show that intentional mistuning alleviates both flutter susceptibility and maximum forced response of a last stage turbine wheel. The SNM-based reduced order models used for this purpose allow for considering aeroelastic interaction and speed dependence of frequency mistuning in a simple manner. Focusing on the fundamental bending mode and IM patterns composed of only two different blade designs, the impact of superimposed and unpreventable random mistuning has been taken into account, whereby the robustness of the approach has been proved. Both, an alternate IM pattern and an IM pattern designed by means of optimization based on genetic algorithms effectuate the most satisfying suppression of flutter from the engineering point of view. Stable conditions are defined by completely positive aerodynamic damping ratios within the eigenvalue solutions. In this regard, always stable conditions are computed by applying these IM patterns. Simultaneously, the impact of random mistuning on the maximum forced response at part speed proves to be less negative compared to the tuned counterpart with nominally identical blades. In consequence, the danger of getting severe magnifications of the maximum forced response is reduced as well, in particular alternate IM yields the most promising results in this regard.

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