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Maximum mistuning amplification of the forced response vibration of turbomachinery rotors in the presence of aerodynamic damping

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ABSTRACT

Mistuning can dangerously increase the vibration amplitude of the forced response of a turbomachinery rotor. In the case of damping coming from aerodynamic effects the situation is more complicated because the magnitude of the damping changes for the different travelling wave modes of the system. This damping variability modifies the effect of mistuning, and it can even result in a reduction of the mistuned forced response amplitude below that of the tuned case (this is not possible in the usual case of constant material damping). In this paper the Asymptotic Mistuning Model (AMM) methodology is used to analyze this situation. The AMM is a reduced order model that is systematically derived from the mistuned bladed disk full model using a perturbative procedure based on the small size of the mistuning and the damping. The AMM allows to derive a very simple expression for an upper bound of the maximum amplification factor of the vibration amplitude that the system can experience (an extension of the well known Whitehead 1966 result to include the effect of non-uniform aerodamping). This new upper bound gives information on the mechanisms involved in the amplification/reduction of the mistuned response: (i) the number of modes participating in the response, and (ii) the ratio between the aerodamping of the directly forced mode and that of the rest of the modes. A FEM of a mistuned bladed disk is also used to verify the AMM predictions for several different forcing configurations, and both results show a very good quantitative agreement.

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1. Introduction

The term mistuning refers to the small differences among the (theoretically perfectly identical) blades in turbomachinery bladed disks. These small undesirable random imperfections result from manufacturing, assembling and material tolerances, as well as in-service wear; and are essentially unavoidable.

Despite of the fact that the magnitude of the mistuning is typically very small, it has a strong influence on the dynamical response of the bladed disk. There has been a lot of research since the 1970s on the effect of mistuning (details can be found in the review [1]), and, roughly speaking, it can be said that in the case of flutter it has a stabilizing effect, while for the

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forced response excitation it typically leads to a considerable amplification of the vibration amplitudes of individual blades. It has been reported in many previous experiments and simulations that the mistuned response can exceed the corresponding vibration amplitude of the tuned structure (i.e., the structure without imperfections) by a factor that is frequently in the range from 1.5 to 2 (see, e.g., [2]). These increased blade vibration levels have a dramatic effect in terms of high cycle fatigue (HCF), which has been identified as the largest single cause of major component failures in aircraft gas turbine engines [3–5]. Mistuning therefore has a considerable negative impact on the safety, operability and readiness of aircraft gas turbine engines, and many research efforts have been directed to improve the understanding of this phenomenon and to try to predict and quantify its undesirable effects.

Fifty years ago Whitehead [6] introduced the following elegant and simple expression for the maximum amplification factor that the vibration amplitude of a forced mistuned bladed disk can experience

$$\text{Amplification}_{\max} = \frac{1 + \sqrt{N}}{2},$$

with N being the number of blades. The result above applies to a situation where the mode that is being forced belongs to a modal family with very similar modal frequencies (blade dominated), and the damping and the mistuning are small. This amplification factor is an upper bound and, according to the review of results presented in [2], it can be quite conservative for large values of N . But, despite of its accuracy, this kind of expressions are very useful from a preliminary analysis point of view because they give a quick idea of how bad things can get because of mistuning, and what are its basic dependencies and scalings.

More recently, Whitehead's result was updated using a reduced order model methodology called Asymptotic Mistuning Model (AMM) to obtain

$$\text{Amplification}_{\max} = \frac{1 + \sqrt{N_a}}{2},$$

where N_a is now the number of active modes, that is, the number of modes that have frequencies close to the frequency of the forced mode (see [7,8]). The key idea from the AMM is that the important thing is not the total number of blades but the number of modes with frequencies similar to that of the forced mode; these are the only modes that the small mistuning can couple and combine to produce an increase of the forced response. This can be clearly seen for example in [9] where the number of blades is $N=64$ but the maximum mistuned amplification that is obtained is only about 1.2 because the number of active modes appears to be just $N_a = 2$.

All the considerations above assume uniform material damping. If we now take into account aerodynamic effects, then the resulting aerodamping typically dominates over the material damping, being at least one order of magnitude larger [10]. This situation is particularly relevant for the case of BLISKS (integrally bladed rotors with no blade-disk interfaces), where there is no friction damping and nearly all the damping comes from the gas flow.

The effect of mistuning on the forced response of a rotor with aerodynamic damping is more complicated, and changes substantially with respect to the uniform material damping case. There is even the possibility of having a reduction (by a factor of 2 or more) of the forced response vibration amplitude by the effect of mistuning, as it has been reported in [11,12] and found in many configurations [13–16]. The reason for this change in the effect of mistuning relies in the fact that the aerodynamic damping has different values for the different travelling wave (TW) modes of the tuned structure. If the TW that is being forced has small aerodynamic damping, then mistuning will couple it with other TWs with higher damping and the mistuned response will have a higher effective damping, which will translate directly into a reduction of the vibration amplitude.

The analyses available in the literature of the forced response of a mistuned rotor with aerodynamic damping typically perform statistical explorations and optimizations on full FEM models and reduced order models [11,12,14,16]. The purpose of this paper is to analyze the effect of mistuning using the AMM methodology [17,7,8]. The resulting AMM is a very reduced model that gives information about the basic mechanisms that play a role in the mistuned response. One interesting outcome of the AMM analysis is the following expression for the maximum response amplification:

$$\text{Amplification}_{\max} = \frac{1 + \sqrt{\xi_r \sum_a \frac{1}{\xi_a}}}{2},$$

where ξ_r is the aerodamping of the TW that is being directly forced, ξ_a is the aerodamping of an active TW modes, and the sum goes over all N_a active modes. This upper bound can be regarded as an extension of Whitehead's but now including the variability of the aerodamping from TW to TW (if the aerodamping is set to constant then Whitehead's result is recovered). It is very simple, but it gives information on the two mechanisms that have an effect on the amplification of the mistuned response: (i) the number of active TW modes involved in the response (as in Whitehead's expression), and (ii) the ratio between the aerodamping of the forced mode, ξ_r , and that of the rest of the active modes (this was not present in the case of uniform damping). This expression with N_a set to N (number of blades) resembles also the one obtained by Chan and Ewins [18] for the maximum blade vibration of a single-DOF-per-sector system with variable blade structural damping; but there is an important difference: in their case what appears in the expression for the maximum response is the damping

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