FORCED RESPONSE COMPUTATION FOR BLADED DISKS INDUSTRIAL PRACTICES AND ADVANCED METHODS

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1. introduction

Turbomachinery designers have a constant preoccupation: improve performance while reducing masses. The new aerodynamic computation tools (CFD) are leading to new types of blade geometry, very different from the eighties flat plates.

Indeed, 3D and viscosity effects, now taken into account in the CFD design, generates complex and thinner blade shapes. One main consequence, often underestimated, is the effect on the blade stress design and dynamic behaviour. Static loads due to the rotation speed, the temperature and the steady aerodynamic forces are now so important that the ability of the material to accept extra dynamic stresses is dramatically reduced. Moreover, the intensity of the sources of excitation in turbomachines has increased: smaller axial gaps between stators and rotors, smaller tip gaps inducing rotor/casing contact risks, etc...

A significant illustration of that is the analysis of failure problems under operating condition: Two decades ago, the main engine problems under operation were related to life assessment, due to fatigue or creep. Static stresses were not perfectly known so the design was performed with significant security margins. These security margins were generally sufficient to accept medium intensity vibrations without any failure. Problems were due to unexpected ageing or damage on blades. Nowadays, the majority of problems encountered is connected to High Cycle Fatigue (HCF). These problems are often very sudden because a huge number of stress cycles can be performed in a very short time (during one flight for instance). A 1 kHz vibration during 20 minutes corresponds to one million oscillations. The margins for dynamic loads being smaller and smaller, the tolerance to vibration have become a key point.

The engine manufacturer has two main approaches to mitigate HCF risks:

- The first approach is the standard design practice, which consists in avoiding dangerous resonance in the operating range. The resonance dangerousness is very empiric and based on older engine experience. I general resonances must be avoided on the first modes (firsts bending and torsion modes). But it is impossible to remove all the resonance and consequently an engine testing is required in order to assess the dynamic levels on the remaining ones. These tests are very expensive and are performed at the very end of the development process. The late discovery of vibratory problems can impact dramatically the engine development schedule and costs.
- The second approach is to accept resonance in the operating range but to estimate early in the design process the associated response level. This requires specific methods for forced response computation. Moreover, in order to improve the accuracy of the analysis, many secondary phenomenons must be accounted for, such as mistuning or friction damping. In this case, advanced methods are necessary.

2. Mechanical particularities of bladed disks

Bladed disks are rotating structures submitted to high load levels (static, dynamic and impact). The way to account for steady load will be presented first.
But these structures are also cyclically symmetric. The mechanical couplings between the blade and the disk are important: while few years ago the blade alone was designed, it is now necessary to study the complete assembly. The computational cost of a 3D model of a 80 blades bladed disks is very important (few millions degrees of freedom). The structure symmetry properties are used to reduce the cost of the analysis by modelling one blade and one repetitive disk sector. The techniques used are presented in this chapter.

2.1. Rotation effects - Static analysis

Only one sector of the structure is meshed. On each "cut" of the disk, specific symmetry conditions are used. For the static analysis, these conditions are simple: the displacements of the two disk cut faces must be similar in order to guaranty the structure continuity.

The problem to solve is a simple static linear problem: $Fs = [K].X$
The unknown is X, the static displacement vector of the structure, $[K]$ is the stiffness matrix and Fs the steady forces applied to the structure (thermal, rotation, steady aerodynamic forces …).

The application of the rotation forces generates a deformation of the structure. This deformation modifies the rotation forces field, which is dependant of each point location in space and the associated mass. Two solutions are possible to deal with this difficulty:

- In case of significant displacement, typically for fan stages, a non linear analysis is required. The non linearity is due to the significant deformation of the structure that modifies the stiffness matrix and the force field. In such a case, at each iteration of the computation, the deformation is determined and the associated stiffness matrix and forces are updated. The convergence is obtained when the evolution of deformations is small between two steps. At the end of the process, the deformation is identified as well as the structure updated stiffness matrix.
- In case of small displacements, only two iterations are generally performed. The deformation is computed and two additive stiffness terms are identified. A $[Ks]$ term, due to the structure deformation than modifies its stiffness. A $[Kr]$ term due to the force field evolution (spin softening). The stiffness matrix is updated ($[K1]=[K]+[Kr]+[Ks]$) and the problem is solved again but with the basic force field and without geometry update (which are already accounted for in the updated stiffness matrix).

It can be seen that one single static analysis does not give an accurate results. The rotation effects are requiring at least two successive linear analyses (not expensive and generally sufficient) or one non linear analysis (for fan for instance).

If the rotation effect is not properly taken into account, the analysis can lead to a wrong operating geometry, not consistent with the one determined by the CFD analysis.