Vibration Analysis of Low Pressure Stages of Large Steam Turbines with ANSYS

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Abstract:

High cycle fatigue (HCF) plays a significant role in a bulk of turbine blade failures. During operation, periodic fluctuations in the steam force occur at frequencies corresponding to the operating speed and harmonics and cause vibration of the bladed disks. Digital prototyping and optimization become the most practical and economical means for turbine stages design and for identification and solving HCF-caused failures associated with resonant conditions.

During fabrication, 2-3% variation of blades thickness is considered acceptable. Such fluctuations can lead to deviation of the blades vibration from expected values within the range of 5%. More drastic difference of blades' dynamic properties during turbine operation could occur as a result of blades erosion. Significant amount of research efforts went into developing strategies of grouping blades such that minimize adversary effect of blade variations. These techniques are illustrated with analysis of gas turbine working wheel with 34 buckets. Mistuning was modeled by blade material density variation. It was presumed that a half of the blades ware 5% heavier than another half. This leads to 2.5% difference in natural frequencies. Two simplest blading methods were considered: blades of the same type were arranged in either 4 interlaced sectors, or in 8 sectors.

Introduction

Usually, computational methods that include cyclic symmetry of design are used for dynamic analysis of bladed discs. The fact that computational objectives of cyclic symmetry problem correspond to the objectives of 2-sectors problem allows not to worry too much about a number of finite elements and concentrate efforts on the fidelity of finite element model to the blade geometry. This approach assumes that all sectors are identical in geometrical and physical characteristics. In practice, the identity of the geometrical parameters could be attained just within the technological tolerances of particular technological processes. Besides, the blades' wear is non-uniform during operation. During last decade significant efforts were devoted to taking into account the individual properties of each blade in vibration calculations of turbine wheels, determining results of the wheel's cyclic asymmetry and issuing recommendations for optimal spacing of mistuned blades in the wheel. Due to the complexity of the object, reduced order modeling [1- 5] became a main method for vibration computations. At the same time, this approach is not supported directly by commercial FE packages and is not practical engineering method.

This work uses parameterization of blade, disc, shank and shroud for direct FE mesh generation. With modest requirements to computational resources, this makes possible to consider a complete problem of turbine working wheel vibration with mistuned blades without involving any additional simplifying assumptions. **ANSYS** provides engineers with capabilities to carry out all dynamic calculations for various laws of mistuned blades spacing in reasonable time and obtain recommendations for blades arrangement.

Parameterization of bladed disc

An array of blade profiles serves as the initial data for bladed disc vibration computations. The profiles are the output data of axial turbine flow path design and optimization obtained with ***AxSTREAM™*** and ***AxPLAN™*** software tools [6]. Preliminary computations of the blades' structural and dynamic characteristics were carried out within ***AxSTREAM™*** on the basis of 1D rod theory. Therefore, verification of the structural and dynamic characteristics of designed blade and fully bladed wheels needs to be performed with detailed 3D FEA. **ANSYS** was selected for this computational work.

At this phase of the turbine design, the working wheel's final parameters are not determined yet and, hence, 3D solid model is not available. For modeling purposes it is much more economical not to create a solid model, but rather use parameterized FE model of shroud, airfoil base rim, shank, and disc for direct mesh generation. Bucket parameters are transferred from ***AxSTREAM™*** as the arrays of points at cross-sections profiles. ***AxSTREAM™*** also computes pressure fields at the blade surface that can be used for harmonic analysis. ***MinuteMesh-Turbo*™** software tool [6]**,** a parameterized mesh generator specifically developed for turbomachinery applications, was used for direct mesh generation. Assignment of shroud and shank parameters in ***MinuteMesh-Turbo*™** GUI is shown in Fig. 1and 2.

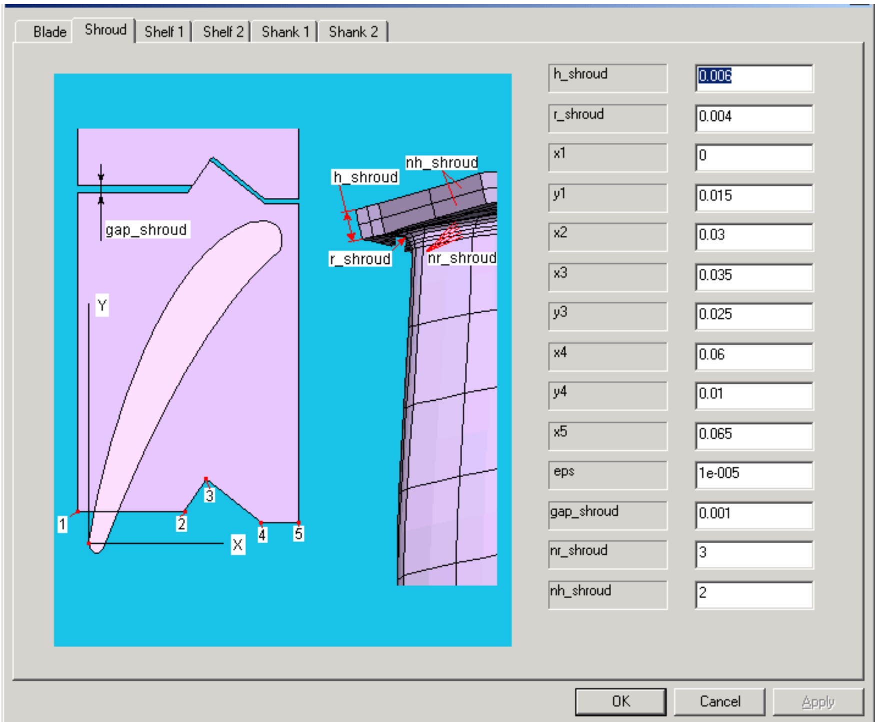


Figure 1. Dialog for assigning shroud parameters

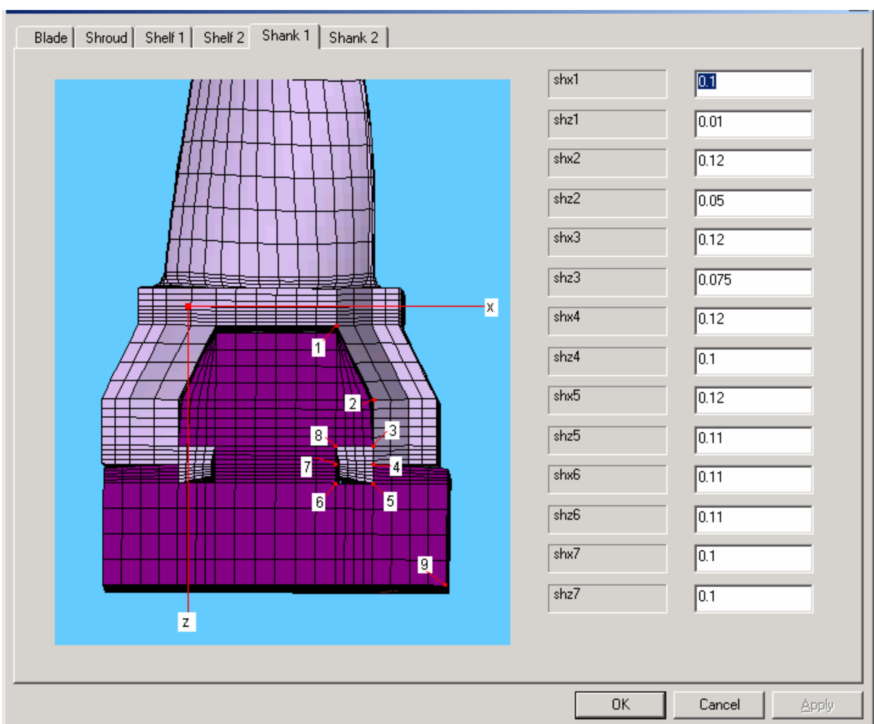


Figure 2. Dialog for assigning shank parameters

FE mesh generation

Linear or quadratic hexahedral Solid 45 and Solid 95 finite elements were used for the bladed disc computation. In the areas with intricate geometry, the degeneration of these elements into wedge elements was allowed. Generation of structured mesh led to minimization of total degree of freedom number, and made possible creation of high fidelity finite element models of reasonable size. Analysis was performed on typical PC.

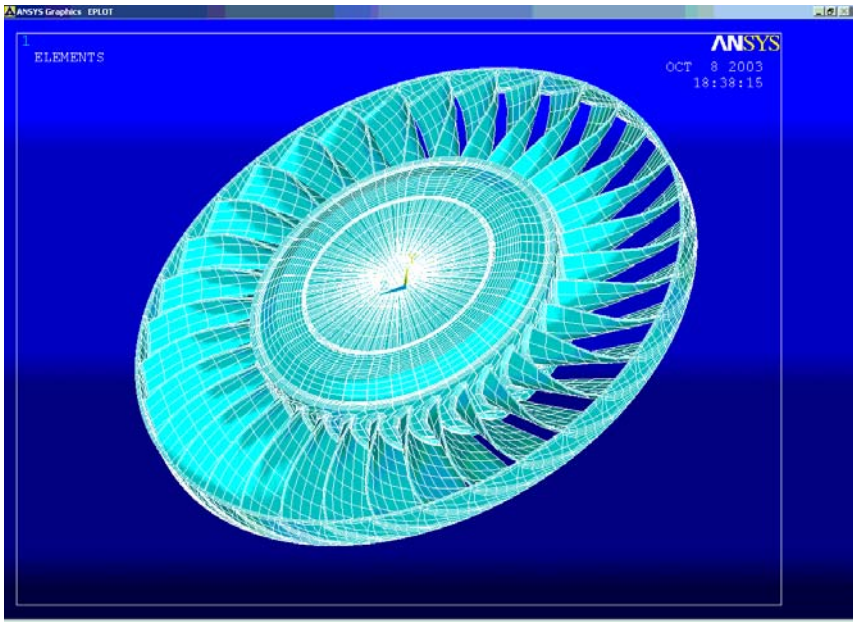


Figure 3. FE mesh for bladed wheel

Blade's surfaces were generated by constructing a B-splines that interpolate between points describing profiles. Engineers have control on partitioning of the blade's surface on leading/trailing edges and pressure/suction sides via GUI of ***MinuteMesh-Turbo*™** . The finite elements are constructed by uniform partitioning of pressure and suction sides in parametrical space and equal partitioning across blade's thickness.

Main approach for shroud and shank FE mesh generation is extrusion of 2D meshes along the line, as well as along two and three lines with linear transformation of the 2D mesh. Mesh transformation with nodes transposition onto selected surface is used for near-fillet surfaces mesh generation.

Modeling of individual properties of the blades was achieved by varying the blade material specification and morphing of meshes. Typical FE model of working wheel generated by ***MinuteMesh-Turbo*™** and imported into **ANSYS** is shown in Fig. 3.

Computation of natural oscillations of mistuned bladed disc

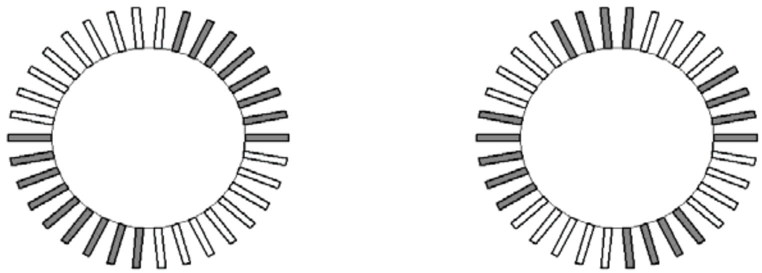
During fabrication, 2-3% variation of blades thickness is considered acceptable. Such fluctuations can lead to deviation of the blades vibration from the nominal within the range of 5%. More drastic difference of blades dynamic properties during turbine operation could occur as a result of blades erosion.

Analysis of gas turbine working wheel with 34 buckets can serve as illustration of the approach offered for studying the mistuned bladed discs dynamics. Mistuning was modeled by blade material density variation. It was presumed that a half of the blade were 5% heavier than another half. This leads to 2.5% difference in natural frequencies. Two blading methods were considered: blades of the same type were arranged in either 4 interlaced sectors, or in 8 sectors, Fig. 4. These two methods are the simplest ones. In principle heavy and light blades can be also arranged in 16 groups. Also, blades can be separated in more than 2 groups based on their weight, leading to even more possible arrangements, providing more flexibility for tuning.

Finite element model of working wheel contained 135,000 elements and had 500,000 degrees of freedom.

Fig. 5 and 6 depict results of the modal analysis. Shapes of axial displacements for 7th and 12th modes are shown. Careful analysis of the pictures reveals lack of harmonic node's lines that normally obtained via analysis with cyclic BCs. This is, of course, expected. The whole point of used here approach is to model realistic situation when all blades are slightly different either because of manufacturing tolerances, or as a consequence of "tear and wear".

Table 1 lists the computed frequencies. A comparison shows that modifying the order of disc blading can change some frequencies by 0.8% leaving other frequencies unchanged. Moreover, frequencies of neighboring 5th and 6th



A B

Figure 4. Two possible blading patterns (darker blades are heavier): A – 4-sector blading; B – 8-sector blading

modes get apart, while frequencies of neighboring 6th and 7th modes get closer. In certain cases, such behavior of frequencies allows to tune the blades out of resonance without redesign them, i.e. solely based on selecting a certain order of blading.

Table 1. Comparison of blade row prime ten frequencies for two variants of mistuned blades arrangement

Number of frequency	1	2	3	4	5	6	7	8	9	10
1st method	308.69	527.30	535.10	544.82	571.09	572.00	614.01	617.24	681.20	681.73
2nd method	308.69	530.68	531.56	545.90	566.53	575.43	615.07	616.34	681.19	681.67
% difference	0	0.641	-0.661	0.198	-0.798	0.6	0.173	-0.146	-0.002	-0.009

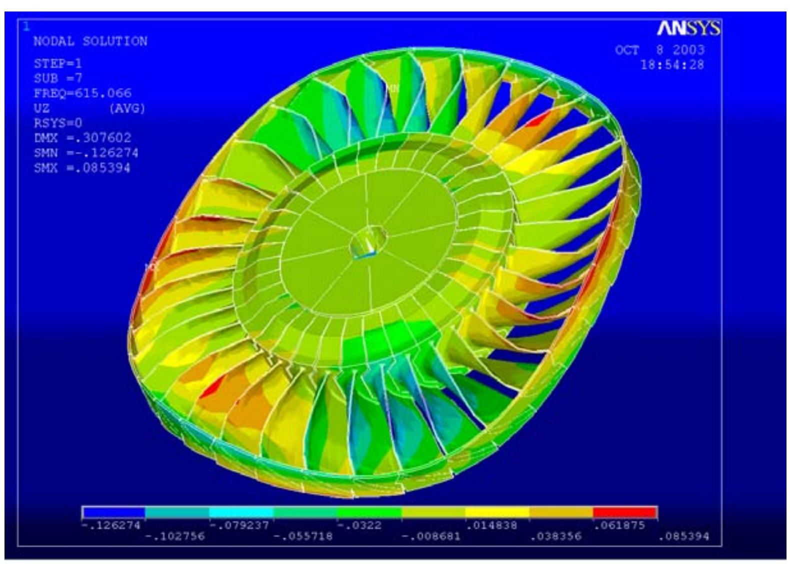


Figure 5. 7th mode shape of bladed wheel

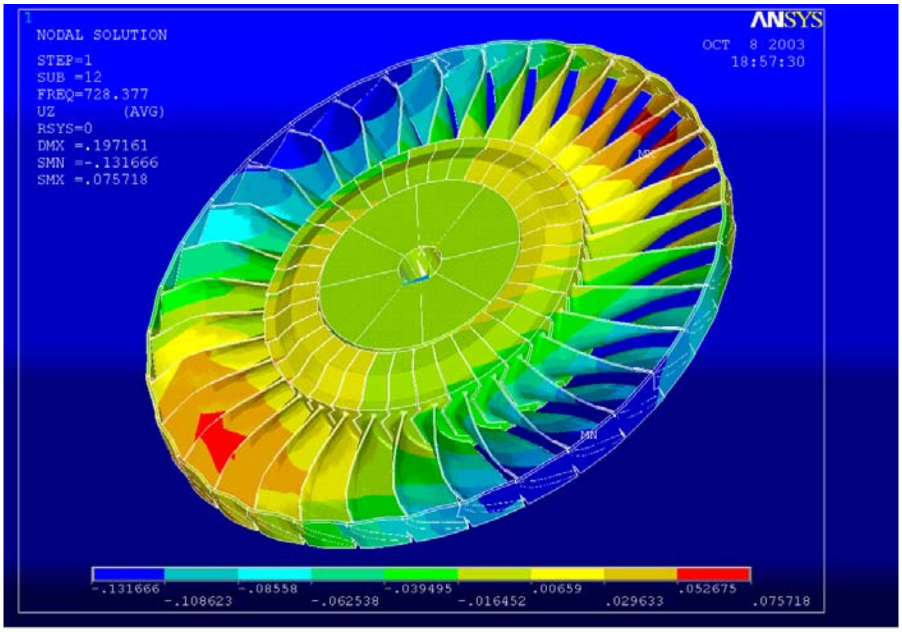


Figure 6. 12th mode shape of bladed wheel

Conclusion

Proposed method for direct computation of mistuned bladed disc with **ANSYS** based on parameterization of blades, shroud, shank, and disc geometry and direct mesh generation with hexahedral finite elements. The method allows carrying out structural and vibration analysis on high fidelity models without using model size reduction techniques.

References

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