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NUMERICAL SIMULATION OF STEAM FLOW AROUND VIBRATING TURBINE BLADES

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Abstract: *This paper is devoted to the numerical simulations of a two-dimensional turbulent steam flow around vibrating turbine blades in a software ANSYS Fluent. Computational domain contains a passage of three blade profiles which were cut near the tip of long curved turbine blade. The movement of each blade was based on the results of a modal analysis which was accomplished by a software ANSYS Mechanical. The mode shape of the blade was described by User – defined function (UDF) which defines the displacement of a two-dimensional rigid body and the rotation of this body around an axis placed in the center of gravity. Numerical simulations of the steam flow through the passage were carried out for a range of nodal diameters defining a phase lag between each pair of blades on a turbine disk and the results of CFD analysis were used for the computation of aerodynamic work of the steam on the blade surface to assess the blade stability.*

Keywords: Turbine blade, ANSYS Fluent, Steam flow, Mode shape, Stability.

1. Introduction

Unsteady pressure distribution along the turbine blade surface which occurs during a steam turbine operation can cause unstable behavior of the blade. One of the tools which can help by assessing the stability of the blade is numerical simulation of the steam flow around the vibrating blades and evaluation of loading that provides work on the blade surface. CFD analysis of the flow around moving bodies is a complex problem which comprises the preparation of a geometry model, generation of a high - quality computational mesh, definition of the blade motion and of the movement of deforming mesh and the correct setting of transient solver.

2. Computational domain and description of blade motion

First of all, the modal analysis in ANSYS Mechanical was provided for three-dimensional low-pressure turbine blades. Natural frequencies for free-standing blade, presented in this paper, were the same for all nodal diameters. However in cases with shrouded blades, natural frequencies differ for different nodal diameters, because blades are bonded at shroud and in the middle.

On the basis of modal analysis the two – dimensional profile was cut near the blade tip which is obviously the place of maximum deformation as seen in Fig. 1 (left). The mode shape of blade profile consists of the displacement in direction of Cartesian coordinates x, y (bending in 3D) and the rotation around the axis placed in the center of gravity (torsion in 3D). Directional vector of displacement and maximal rotational angle were gotten from the results of modal analysis. The amplitude of bending vibrations was obtained from Blade tip timing measurement. Vibrating motion was considered as a sinusoidal time-dependent function

$$\begin{aligned} x(t) &= x_0 \sin(\omega t) \\ y(t) &= y_0 \sin(\omega t) \ , \\ \alpha(t) &= \alpha_0 \sin(\omega t) \end{aligned} \qquad (1)$$

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where t is time, $\omega = 2\pi f$ is angular frequency, where f is natural frequency, x_0, y_0 are the amplitudes of bending vibrations and α_0 is the amplitude of torsional vibrations.

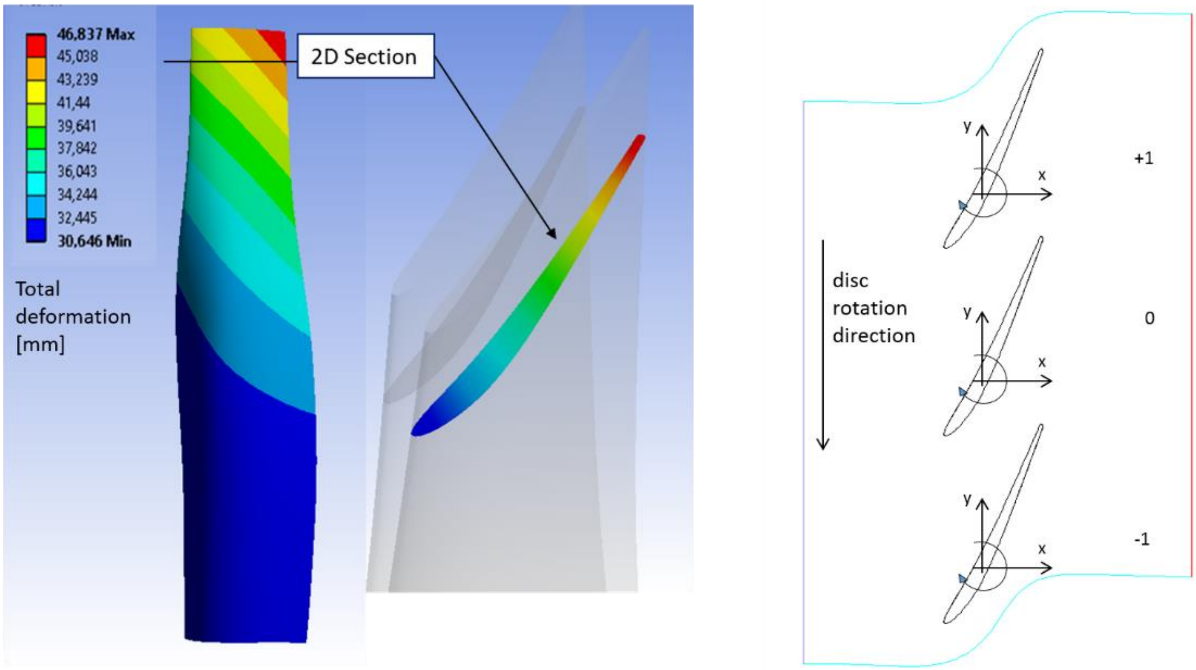

Fig. 1: Total deformation of 3D blade (left). Computational domain (right).

Fig. 1 (right) shows computational domain which contains three two – dimensional blade profiles. The blade numbered zero is referential and this blade is the main subject of this research. The neighboring blades are numbered ±1. These two blades are included in computational domain because they have the greatest influence on origin of unsteady pressures at the surface of referential blade in comparison to other blades of the bladed disk which was proved in paper (Panovsky et al., 2000).

The blades were considered to vibrate according to harmonic function (1), where each pair of blades vibrate with particular constant phase angle ± φ - so called inter-blade phase angle (IBPA). The blade number +1 moves with phase angle + φ with respect to referential blade and similarly the blade -1 vibrate with phase angle –φ according to equations

$$\begin{array}{lll} x(t) = x_0 \sin(\omega t) & x(t) = x_0 \sin(\omega t + \varphi) & x(t) = x_0 \sin(\omega t - \varphi) \\ y(t) = y_0 \sin(\omega t) & y(t) = y_0 \sin(\omega t + \varphi) & y(t) = y_0 \sin(\omega t - \varphi)\,. \\ \alpha(t) = \alpha_0 \sin(\omega t) & \alpha(t) = \alpha_0 \sin(\omega t + \varphi) & \alpha(t) = \alpha_0 \sin(\omega t - \varphi) \end{array} \qquad (2)$$

Numerical simulations were accomplished for IBPAs from interval (-180 °, 180 °). The so called traveling wave propagates along bladed disk for each phase angle. The number of these waves is denoted as a nodal diameter *nd* which could be computed from IBPA as

$$nd = \frac{IPBA \cdot N}{360} \qquad (3)$$

where N is the number of blades at the disk.

3. Mesh motion method

Quadrilateral computational mesh with boundary layer at the blade surface was created using software GridPro, see Fig. 2. The mesh quality was checked by the value of parameter y+ in the first cell of boundary layer. In this case lies the value of y+ in interval (0.1; 4) and average value is around 1.

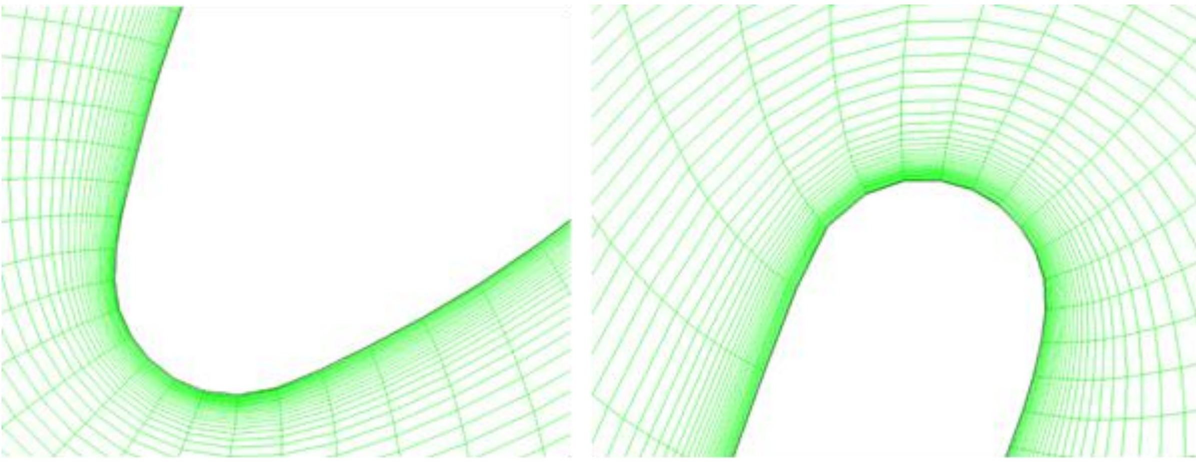

Fig. 2: Details of computational mesh – leading edge (left), trailing edge (right).

Smoothing method was used for defining of mesh motion. This algorithm is convenient for cases with body motion which has a small amplitude of vibrations because it doesn't change whole mesh structure, just repositions each mesh node according to movement of the body. This method, which is described for example in contribution (Řidký et al., 2015), moves mesh nodes in response to displacement of boundaries by calculating a mesh velocity by an equation

$$\nabla \cdot (\gamma \, \nabla \vec{u}) = 0 \tag{4}$$

where the velocity $\vec{u}$ at the moving boundary is used as a Dirichlet boundary condition. The diffusion coefficient γ is computed from boundary distance *d* by formula

$$\gamma = \frac{1}{d^{\beta}}, \tag{5}$$

where parameter β could be from interval (0, 2) and controls how far from moving boundary the diffusion affects the mesh. The diffusion parameter in our simulations was 2. Higher values of β preserve the shape of mesh cells near the moving boundary and spread the movement further from body. In opposite for $\beta = 0$ the equation (5) gives the diffusion coefficient $\gamma = 1$, which means a uniform diffusion of the mesh motion.

4. Numerical simulation and results

Numerical simulations of a two-dimensional steam flow through the computational domain were accomplished by software ANSYS Fluent for eighteen different nodal diameters covering a whole range of bladed disk mode shapes.

Transient computation was initialized by steady simulation. Density based solver was chosen which is more convenient for compressible fluid flow. The steam was substituted by a compressible ideal gas with particular material properties which correctly describe the steam behavior. Two – equation SST k-omega turbulence model was used to set the turbulent viscosity. The disk rotation was included using a single moving reference frame approach which assigns constant velocity to whole domain. Vibrating motion of blades was defined by UDF *Define-cg-motion* which assigns the blade the translational and rotational velocity in form of a time derivative of equations (2).

Numerical results of CFD analysis were used to assess the option of unstable behavior of the referential blade. Aerodynamic work of fluid on the blade was computed from pressure distribution along the blade surface. Negative value of aerodynamic work means that the fluid absorbs blade energy and the blade motion would be damped. On the other hand positive value of aerodynamic work could lead to unstable blade behavior. Aerodynamic work of fluid on the referential blade is computed as the time integral of power as described in thesis (Panovsky, 1997)

$$W = \int_{t}^{t+T} \int_{A} p \, \vec{u} \, \vec{n} \, dA dt, \tag{6}$$

where p is unsteady pressure on the surface of the referential blade, $\vec{u}$ is the local velocity vector, $\vec{n}$ is the outer unit normal vector, A is blade surface area and T is the time period of oscillation. Dependence of aerodynamic work on nodal diameter is shown in Fig. 3.

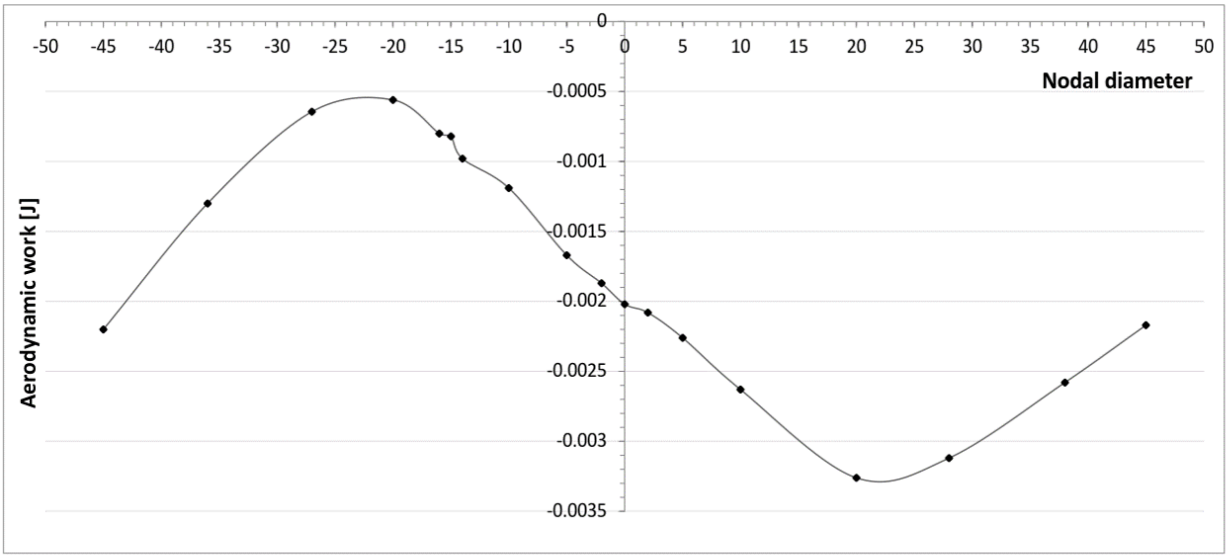

Fig. 3: Aerodynamic work for nodal diameters from interval (-45, 45).

5. Conclusions

This contribution describes the numerical simulation of steam flow around vibrating 2D – blade profiles and stability analysis for the turbine blade. Aerodynamic work of fluid at the blade surface was evaluated. Positive value of aerodynamic work would mean that the blade absorbs energy from the flow which would lead to unstable blade behavior. The tested blade would be stable according to the results depicted in graph in Fig. 3 because of the negative value of aerodynamic work which results in the damped motion of referential blade.

Acknowledgement

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