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Experimental Verification of Test Blade Loading

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Abstract. The paper describes process of verification of FEM model of turbine rotor blade fatigue test. Design of new turbine blade is tested on experimental stand, where load situation is similar to load of rotor blade of steam turbine during operation. Real load of blade and boundary condition on experimental stand was monitored by means of strain gauges set. Experimental data were compared with the computed results of several FEM models.

Introduction

Rotor turbine blades and their disk mounting are key elements of steam turbines. Rotor blades and blade-root attachment are exposed to variety of mechanical loads such as centrifugal force, loading by mass of steam or dynamic loading by means other sources of excitation as rotating and wakes behind stators blades. Different types of blades attachment together with design methodologies were developed in the past. These traditional methods are very conservative today, and it is difficult to use new materials optimally. New methodology must be developed for new modern types of steam turbines with 3D blades and new types of blades attachments.

Some of the problems mentioned in the previous paragraph are solved in project TA CR Blade attachments. New experimental stand for rotor turbine blade and its attachment loading was designed. The turbine blade can be loaded by combination of axial tension, bending and torque. FEM models of the stand were prepared and for verification of these models were carried out measurements on the stand during experiment.

Measuring on the stand

Loading of experimental turbine blade is combination of axial tension bending a torsional loads. Axial and torsional loads are static during experiments and these are introduced by loading bar. Fig. 1 sketches schema of the loading bar. There was made decision that six axial strain gauges and torsion strain gauges with two measuring grids were installed on the bar. Fig. 1 shows strain gauges position and its designation. Two strain gauges rosettes were installed on opposite side of turbine blade specimen due to monitor stress load on the blade surface during loading. Cyclic bending loading was made by resonant pulsator.

Loading of the bar was determined by strains measured in two cuts I-I and II-II. Strains measured by longitudinal strain gauges are combination of bending and axial parts of strain. Fig. 2 shows the situation in cut I-I. It is possible describe deformation measured by strain gauge T1 by equation (1)

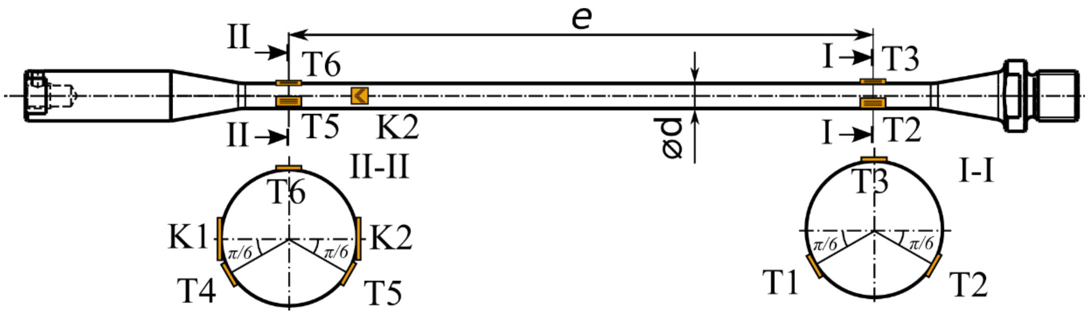

Fig.1 Load bar with installed strain gauges

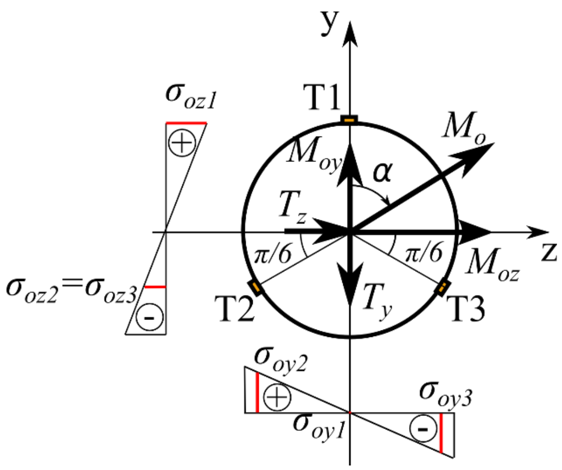

Fig.2 Bending stress in cut I-I

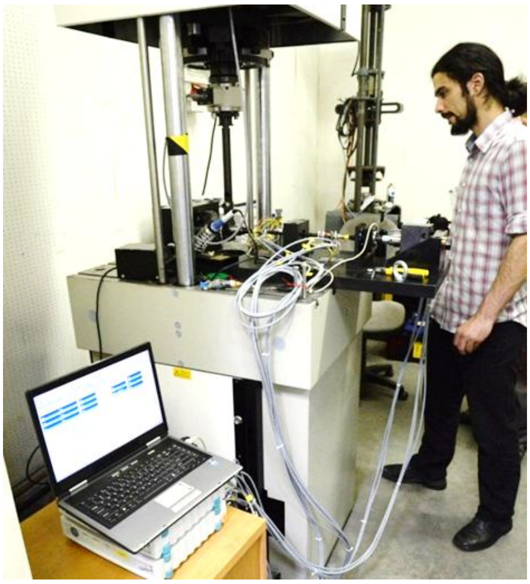

Fig.3 Experiment on testing stand

$$\varepsilon_1 = \frac{\sigma_{oz1}}{E} + \frac{\sigma_t}{E} = \frac{M_{oz}\frac{d}{2}}{JE} + \frac{F}{AE}, \tag{1}$$

where ε_1 is strain measured by strain gauges T1, σ_{oz1} is bending stress caused by the component M_{oz} of bending moment M_o , F is axial force, E is Young's modulus, J is area moment of inertia of the cross section, A is cross section area and d is diameter of loading bar. Similarly is possible describe strains measured by strain gauges T2 and T3:

$$\varepsilon_2 = -\frac{\sigma_{oz2}}{E} + \frac{\sigma_{oy2}}{E} + \frac{\sigma_t}{E} = -\frac{M_{oz}\frac{d}{2}\sin\frac{\pi}{6}}{J_z E} + \frac{M_{oy}\frac{d}{2}\cos\frac{\pi}{6}}{J_y E} + \frac{F}{AE}, \tag{2}$$

$$\varepsilon_3 = -\frac{\sigma_{oz3}}{E} + \frac{\sigma_{oy3}}{E} + \frac{\sigma_t}{E} = -\frac{M_{oz}\frac{d}{2}\sin\frac{\pi}{6}}{J_z E} - \frac{M_{oy}\frac{d}{2}\cos\frac{\pi}{6}}{J_y E} + \frac{F}{AE}. \tag{3}$$

Axial force F can computed from the sum of equation (1)-(3):

$$\varepsilon_1 + \varepsilon_2 + \varepsilon_3 = \frac{3F}{AE} \Rightarrow F = \frac{\varepsilon_1+\varepsilon_2+\varepsilon_3\ \ AE}{3} = \varepsilon_t AE. \tag{4}$$

If ε_t is strain corresponding to axial loading is possible express components o bending moment in cut I-I:

$$\varepsilon_1 - \varepsilon_t = \frac{\sigma_{oz1}}{E} = \frac{M_{oz}\frac{d}{2}}{J_z E} \Rightarrow M_{oz} = (\varepsilon_1 - \varepsilon_t) W_o E, \tag{5}$$

$$\varepsilon_2 - \varepsilon_3 = \frac{2M_{oy}\frac{d}{2}\cos\frac{\pi}{6}}{J_y E} \Rightarrow M_{oy} = \ \varepsilon_2 - \varepsilon_3\ \ \frac{W_o E}{2\cos\frac{\pi}{6}}. \tag{6}$$

Components of moments in cut II-II are obtained similarly by means of strains measured by strain gauges T4-T6. Components of shear force can be computed by equations (7)

$$M_{oz}^{I} - M_{oz}^{II} = T_y e \Rightarrow T_y = \frac{M_{oz}^{I} - M_{oz}^{II}}{e},\ M_{oy}^{I} - M_{oy}^{II} = T_z e \Rightarrow T_z = \frac{M_{oy}^{I} - M_{oy}^{II}}{e}, \qquad (7)$$

where *e* is distance of cutes I-I and II-II.

Strain data were measured when experimental stand and turbine blade were exposed to various loading conditions. Load bar with connected turbine rotor blade specimen can be seen in Fig 4.

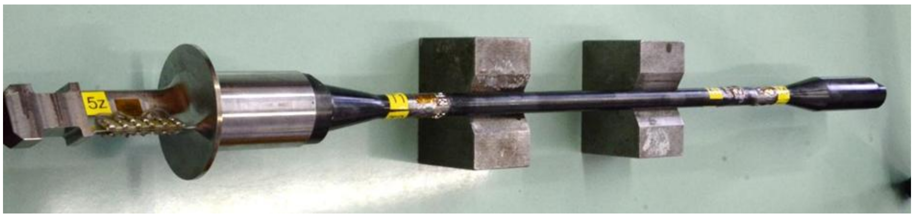

Fig.4 Loading bar and turbine blade assembly with installed strain gauges

Two strain gauge rosette were installed on opposite side of the blade. Data measured by rosettes were used for comparison with various numerical models. Example of measured data can be seen in Fig. 5. Measured strain extremes were chosen to compute extreme internal loads in chosen cuts [1].

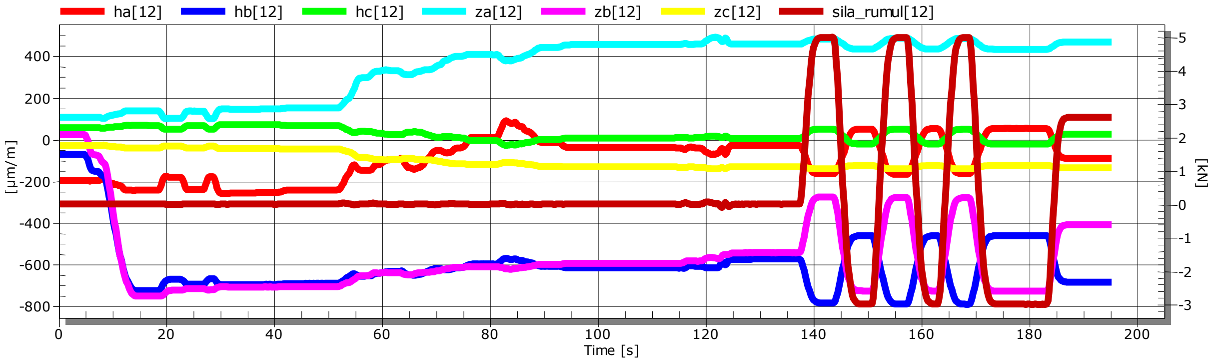

Fig.5 Example of strains measured on surface of test turbine blade

FEM computing

Number of cycles to failure and load magnitude are primal outputs of complex testing of turbine blades on designed experimental stand. The fatigue criteria (to be through comprehensive testing of this product verified) mostly incorporate time-varying components of the tensor of stress or strain. The course determination of these components is very difficult due to the complicated geometry of the blades and complex loading, therefore this course is determined by numerical simulation. Numerical model of the stand must be designed with respect to efficiency and speed of computation for different configurations as well as to results accuracy. For the purpose of verification of the calculations were created several models differing in ways both load and attachments of testing turbine blades [2]. Firstly complex FEM model of testing stand was create but comparison with experimental results was not satisfactory. The production inaccuracies resulting in internal loads are the supposed reasons for the difference.

Next model was simplified and loading bar was replaced by inner load determined by experiment such as components of bending moment M and shear stress computed by methodology described in previous section. Results of experiment and computation were compared by strains measured on the turbine blade specimen. Fig. 8 shows relative comparing of complex and simple FEM models with experimental results.

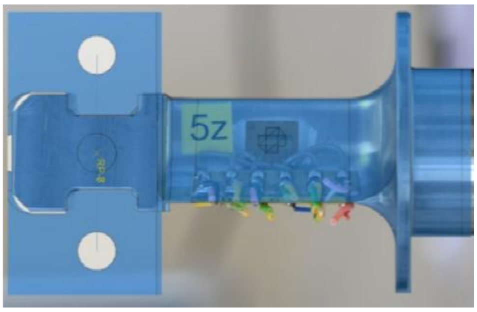

Fig.6 Position of numerical and real rosette

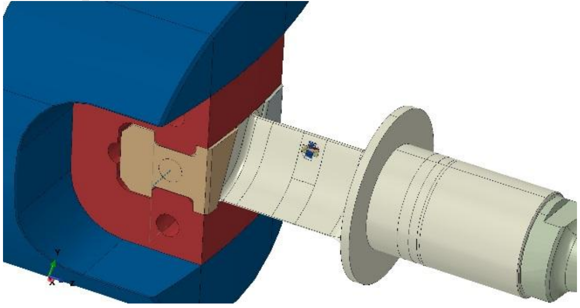

Fig.7 FEM model with rosette

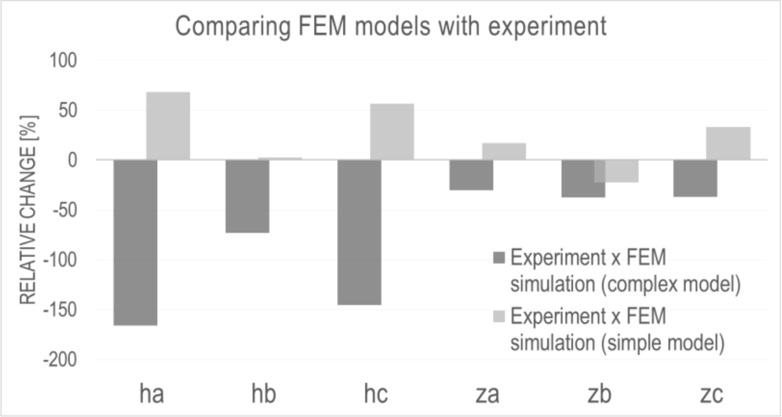

Fig.8 Relative comparing of FEM models with experimental data

Summary

Real loading of rotor turbine blade tested on experimental stand was monitored by number of strain gauges. Experimental data were used for verification of various types of FEA models. The model with the best ratio of accuracy versus computational requirements was selected. This model will be used for determination of real loading of turbine blade specimen. Input data will be computed by means of strains measured on the loading bar. Experimental data together with chosen FEA model will be used for new design methodology development.

Acknowledgment

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