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COMPUTATIONAL FATIGUE STRENGTH PREDICTION OF TURBINE BLADE PIN JOINT

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Summary: *Presented paper deals with computational fatigue strength prediction of high pressure level steam turbine blade pin joints. Quazi-static FE analysis of one turbine disk segment was carried out and the most exposed domains were found with respect to the centrifugal load due to nominal revolution 3000 rpm and varying pressure due to stator blades. The possibility of fatigue cracks initiation and following propagation was evaluated on the bases of FE analyses results.*

1. Introduction

Presented analysis was carried out in conjunction with ŠKODA POWER a.s. in Pilsen. This concern deals with design and manufacturing of steam turbines. Typical steam turbines work under nominal revolution of 3000 rpm and high pressure water steam condition.

Complex analysis of high pressure steam rotor blade lock is presented in the paper. The aim was evaluation of the most exposed domains of rotor blade locks with respect to the possibility of fatigue cracks initiation and following propagation. This analysis was based on the stress and strain fields determined using FE computation.

2. Theoretical assumptions of fatigue crack propagation modeling

Initial quarter-circular crack is located in blade pin-lock hole just in point of maximum radial stress. The direction of the crack propagation is assumed to be prependicular on radial direction at which the dominant stress component, caused by centrifugal forces, is acting. Numerical simulation according to analytical (NASA FLAGRO) propagation model was done to evaluate (potential) propagation of the crack:

$$\frac{\mathrm{d}a}{\mathrm{d}N} = A^{*} \cdot \Delta K^{m} \frac{\left(1 - \frac{K_{th}^{*}}{\Delta K}\right)^{p}}{\left(1 - \frac{\Delta K}{K_{c}^{*}}\right)^{q}}, \tag{1}$$

where

$$A^{*} = \frac{A}{(1-R)^{(1-\gamma)m}}, \; K_{th}^{*} = K_{th}(1-R)^{(1-\gamma)m}, \text{ and } K_{c}^{*} = K_{c}.$$

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Further, crack tip shape is assumed with no change. It remains quarter-circular, only radius describes crack propagation. Simulation is based on material parameters A, m, K_{th}, K_c, p, q, γ and loading, described using stress intensity factor (SIF) amplitude, ΔK, and load ratio $R \approx K_{\min} / K_{\max}$.

Analytical model (1) requires SIF-crack length dependence to evaluate crack increment. Both pin-lock geometry and loading is too complex to use existing empiric formulas evaluating SIF at quarter-circular crack tip in hole. K-calibration was used to construct correction function

$$Y_A = \frac{K}{\sigma_A \sqrt{\pi \cdot a}}, \tag{2}$$

of SIF K in given point of crack tip, stress σ_A in chosen nominal point A and crack length a. If loading conditions (σ_A) are similar to conditions at which correction was computed, I allows to evaluate SIF as

$$K = Y_A \cdot \sigma_A \sqrt{\pi\left(a + r_p^*\right)}, \tag{3}$$

where $r_p^* = \dfrac{K^2}{6\pi \cdot R_e^2}$ is plastic zone at crack tip dimension, and R_e is yield point of material.

3. Finite Element Model

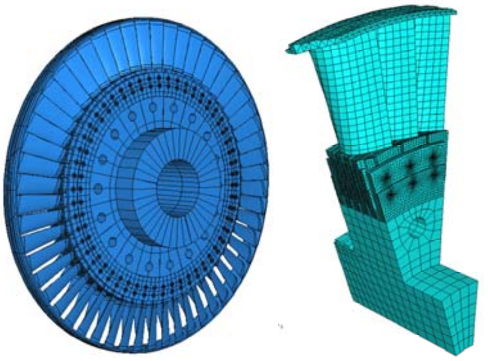

Fig1: Actiing wheel and single three-blades bundle segment .

The main purpose of numerical stress analysis was to determine SIF K in blade pin-joint model crack needed by previously mentioned NASA FLAGRO propagation model. All FE analyses presented in the paper utilized program ABAQUS. To analyze SIF in modeled cracks effectively, submodeling technique was employed. The idea of submodeling is, simply said, to analyze a small (usually critically loaded) domain with much better interpolation than the rest of the body. The domain, represented as separate FE model, is embedded into global model (separate, relatively coarse meshed FE model of whole body) such a way, that coordinate frames of global model and submodel are coincident. To represent reality, submodel should be loaded with given body, surface and nodal forces acting in it's domain, except the interface, where displacements transferred from global model should be prescribed.

Submodeling technique with two levels of modeling was used:

1. Global model including three blades bundle welded with blading cover and corresponding segment of rotor shaft was designed to evaluate stress and displacement in whole action

wheel of the turbine under centrifugal loads and jettube excitation. Resulting displacements were used to load submodel

2. Submodel - relatively small part cut off blade part of pin-joint includes two bodies - pin and hole - at which quarter-circle crack is modeled. Submodel is loaded with centrifugal loads (“inside”), and with displacements determined from global model (on interface). Several submodels were used to evaluate SIF K as a function of crack dimension.

Global model represents single rotor segment (Fig. 1) consisting of three blades bundle welded together with blading cover and corresponding shaft segment. Blades bundle is fitted in rotor shaft with pin-joint. Pin-joint includes shaft and blade slots with holes and six pins coming through the holes. Attention was focused namely on the pin-joints holes at which cracks were modeled. Geometric imperfections of pins and holes were not implemented in the model. Global model was analyzed with linear elastic, as well as with elastic-plastic material.

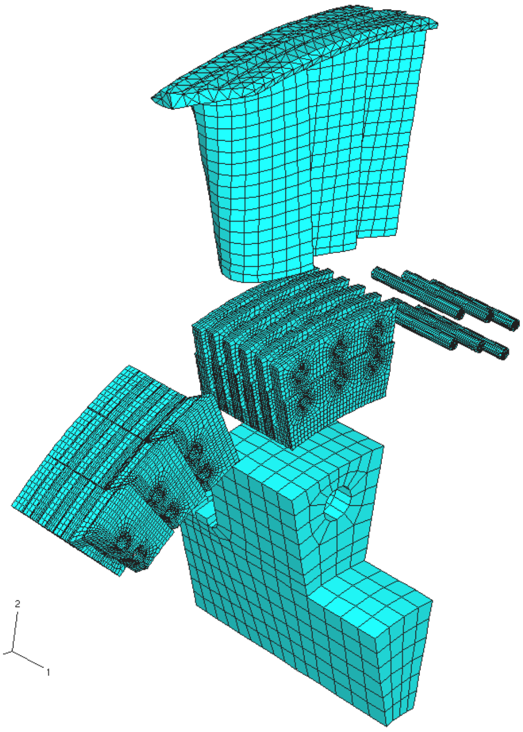

Fig.2: FE - mesh of global model.

Segment was loaded with centrifugal loads from steady state rotation (50 rpm) and with harmonic (frequency 160 Hz) excitation forces caused by flowing steam (so called jettube excitation), shown in Fig. 3. Used harmonic excitation forces were based on CFD analysis (Fluent) and steady state vibrations analysis on cyclically symmetric model (PMD). Internal forces in blades roots during one period were used as excitation load.

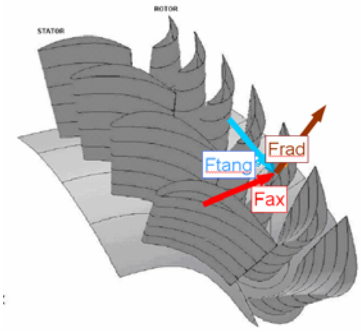

Fig.3: Jettube excitation forces.

Shaft segment was supported using axisymmetric boundary conditions. It was acceptable because only quasistatic loading regimes were solved, based on appropriate internal loads corresponding with harmonic excitation, as mentioned above.

Mesh consisting of linear (8-node) continuum elements was significantly denser in pins and near joint holes. Remaining areas were meshed rather coarsely with respect to computational costs. Model utilizes incompatible meshes connected with constraints, as shown in Fig. 2. Interaction between pins and holes was modeled as contact with friction coefficient 0.1.

Submodel. Single hole-pin submodel (Fig. 4) was analyzed to evaluate initiation and/or propagation of fatigue cracks in the most loaded lock. Halph-circle shaped cracks with varying radius were assumed (and modeled) in the hole in submodels. K-calibration of analytical crack propagation model was based on stress intensity factors computed with the submodels.

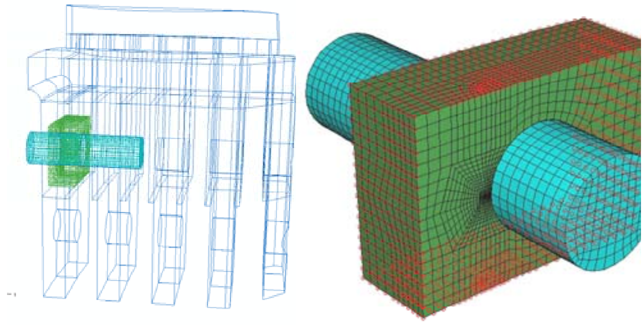

Fig. 4: Submmodel – location.

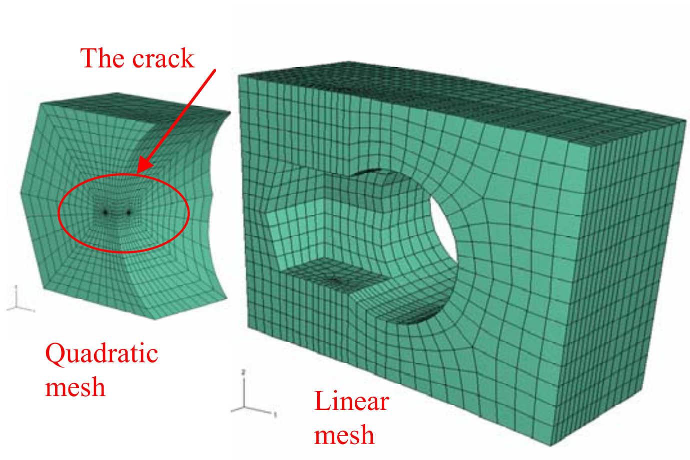

Fig.5: Submodel with crack domain.

Submodel slot mesh is subdivided into two domains with incompatible interface (see Fig. 5): the domain including crack is densely meshed with quadratic (20-node) elements, while in the rest of slot linear (8-node) continuum elements are used.

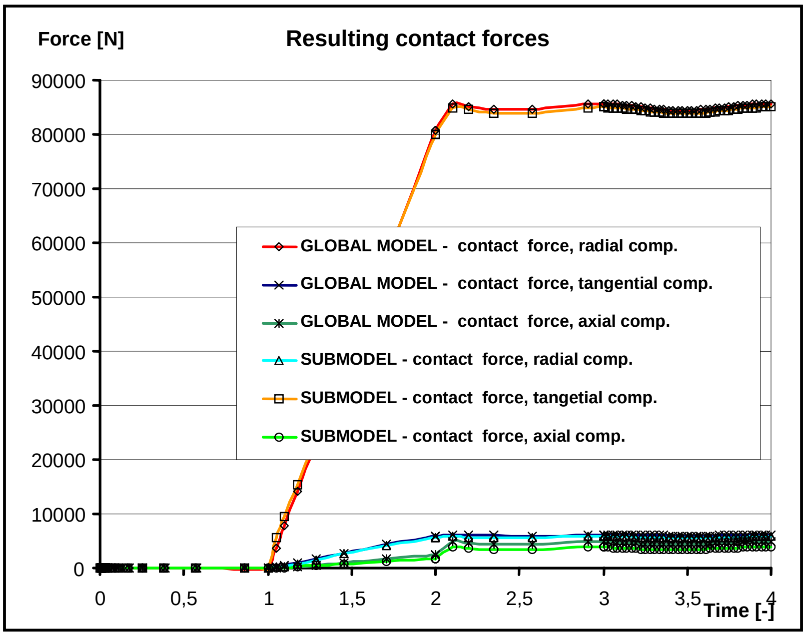

Fig.6: Contact forces in global model and submodel comparison.

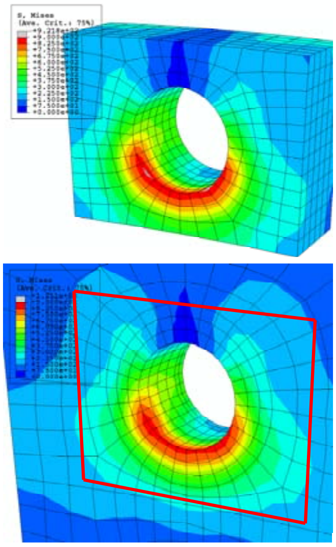

Fig. 7: Von Mises stress

Submodel /global model

Submodeling verification was based namely on comparison of radial component of contact force between the slot and pin expressed in both global model and submodel. Correspondence between (computational) time-contact force dependencies during loading process plotted in Fig. 6 is acceptable. Also visual comparison of stress fields in domain of interest between global model and submodel (Fig. 7) is satisfactory, and therefore submodel may be accepted.

4. The results.

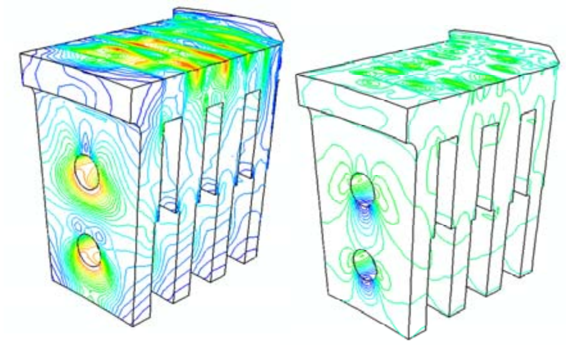

Fig.8: Radial (left) and Von Mises (right) stress disribution.

Damage and fatigue crack initiation. Crack initiation resistance evaluation was based on stress field in slots holes domains determined in global model with elastic plastic material.

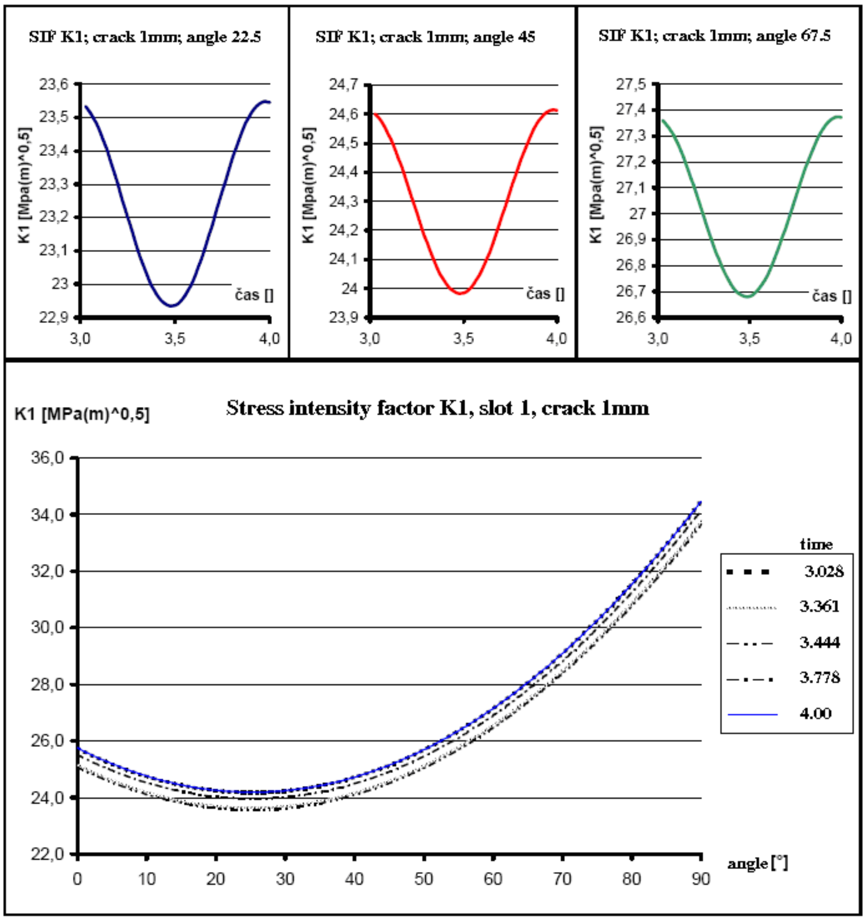

Fig. 10: SIF distribution slony 1 mm crack tip.

Fig.9:The specimen

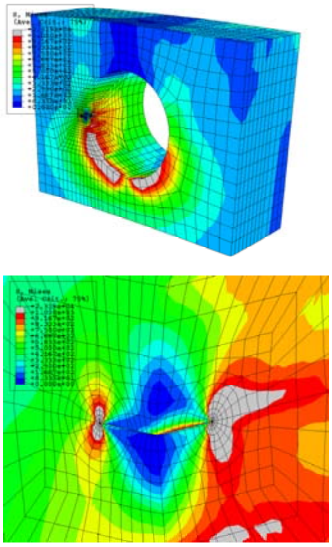

Fig.11: Von Mises stress near 1 mm crack tip.

Damage evaluation was done with special software PragTic (www.pragtic.com) developed in FME CTU in Prague. According to experimental research (see specimen in Fig. 9) the most appropriate multi-axial criteria - integral criterion and critical plane criterion – according to Papuga were used. Maximum damage was calculated as 0.8 (1.0 crresponds with fatigue crack initiation), while manufacturing effects were neglected. It proves, that, under ideal conditions, no fatigue cracks initiations occurs during operational life of turbine

Fatigue crack propagation. SIF factors K were determined for quarter-circle craks with radius of 0.1; 0.3; 1 a 5 mm using appropriate submodels. SIF along the 1 mm crack tip distribution is plotted in Fig. 10, where computational time between 3. and 4. represents one period of jettube excitation under centrifugal preloading. Stress in 1mm crack domain is presented in Fig. XX 11.

Calibration function Y (2) was determined using polynomial approximation between SIF values from FE calculations in dependence on crack length (radius) a and nominal stress σ_A in net slot cross-section done through hole axis. The function is plotted in Fig. 12.

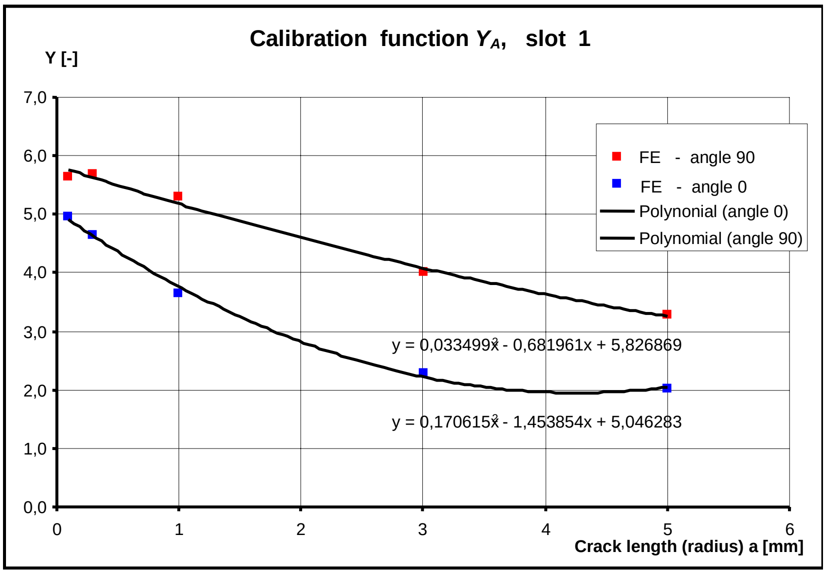

Fig.12: Calibration functions .

Macro crack propagation was simulated based on NASA Flagro model (1), (2), using the correction function. This approximate, simplified analysis should proof if, and under what conditions, the macro crack can propagate. Two operational regimes were in focus of interest.

1) The regime of turbine startup and breakdown cycles representing loading with load ratio $R = 0$ at which maximum load corresponds with operational frequency 50 rpm and minimum, zero load corresponds with frequency 0 rpm.

2) The regime of steady state harmonic jettube excitation during

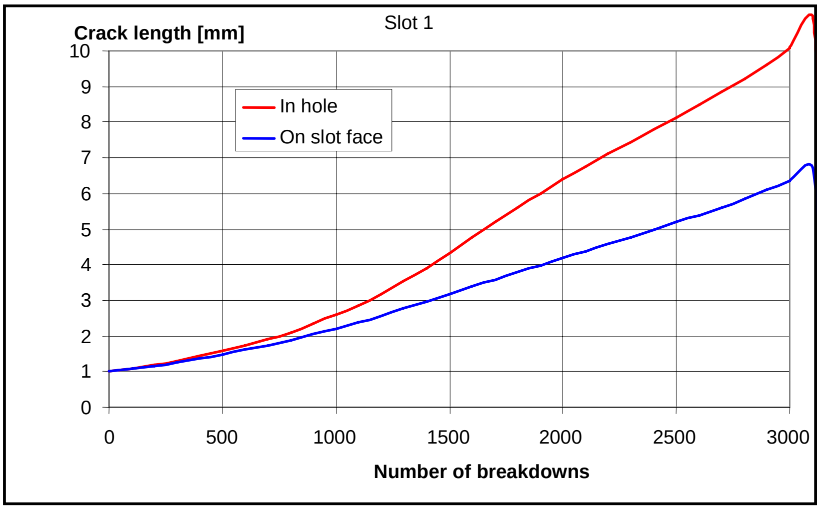

Fig.13 Propagation curves.

operational rotation under $R \geq 0.85$.

Since regimes differ, material parameters were accommodated to each other via crack propagation tests with high load ratios, to determine threshold value of SIF.

Simulations have shown, that in both focused cases cracks propagation occurs in large number of cycles significantly exceeding lifetime of blade joints. For illustration, dependency of crack length on number of cycles associated with regime 1) is plotted in fig 13

5. Conclusion

- Steady state vibrations (1600 Hz) occur during turbine operation due to the jettube excitation. Excitation forces were determined via CFD and utilized to calculate corresponding quasi-static stress.
- Stress in joints has large average value and amplitude 5 -10 MPa in high-cycle fatigue domain.
- Detailed non-linear FE analysis of model specimen was used to tune parameters of fatigue damage models in program PragTic.
- Fatigue analysis of blade joint model proved, that no crack initiation is expected
- Quarter-circle model crack was utilized to evaluate influence of blade joint holes surface damage during manufacturing.
- Initial estimations of material parameters were proved experimentally.
- Crack propagation rate (initial size from 0.1 to 5 mm) according to analytic model is very slow. Limit state would be reached after design service-life.

Acknowledgments

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Literature

Papuga, J. (2005) Mapping of Fatigue Damages - Program Shell of FE-Calculation, *PhD Thesis*, CTU in Prague, (http://www.pragtic.com).

Broek, D (1982) *Elementary Engineering Fracture Mechanics*, Martin Nijhoff Publ.