SERVICE LIFE ESTIMATION FOR RUNNER'S BLADE OF AN AXIAL TURBINE

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The present paper analyse the stress, deformations and service life on runner's blade of an axial turbine, using the finite element software Cosmos Design Star, for different versions of the stress relieve groove and number of blades.

1. INTRODUCTION

The purpose of the study is to calculate the service life for a blade runner of a Kaplan turbine based on stress and deformations resulted from finite element analyse. The blade has a complex geometry and was created as 3D solid geometry using Autodesk Inventor software [1]. The blade's number is imposed by constructive and functional criteria. The following steps are required for this analyse:

- generate the 3D model of the blade as solid model;
- determine the blade loads from hydrodynamic conditions;
- the blade linear static analyse, for different versions of the stress relieve groove and number of blades; analyse was made with Cosmos Design Star software [2], using the methodology detailed in [3];
- the service life estimation.

2. THE BLADE GEOMETRY

The CAD modelling process of the blade was detailed in [4]. To be used in Cosmos Design Star software, the blade geometry must be modelled as 3D solid geometry, figure 1. The regions A and B are the critical for stress values; the blade is designed with stress relieve groove to decrease the maximal value of the stress in the regions A and B. There was analysed two geometry of the stress relieve groove:

- version 1 – a channel with constant radius R_1 (Fig. 2);
- version 2 – a channel with constant radius R_2 ($R_2 > R_1$) and with conical surface (Fig. 3).

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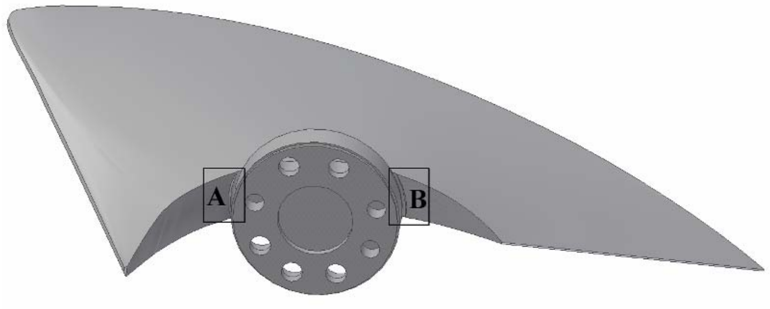

Fig. 1 – The 3D geometry of the blade.

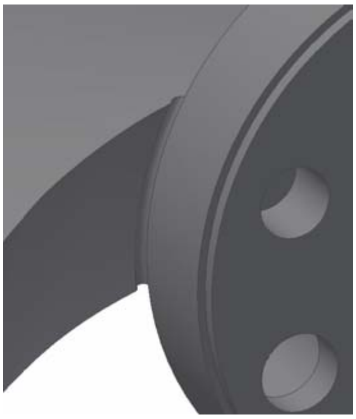

Fig. 2 – The geometry of the stress relieve groove – version 1.

Fig. 3 – The geometry of the stress relieve groove – version 2.

3. THE BLADE LOADS

The hydrodynamic loads applied to the runner blade are presented in Figs. 4 and 5 [5]:

- the gravity load G result from the blade mass;
- the centrifugal load F_c result from the blade rotation;
- axial load $P_n = F_{ax}$; the axial load value is obtained from force and moments measurements on the turbine model;
- the tangential load T, resulted from the machine couple.

The axial and tangential components are calculated for 4 and 6 blades number of the runner. In the Fig. 4, R is the resultant hydrodynamic load and e is the distance to the blade axis.

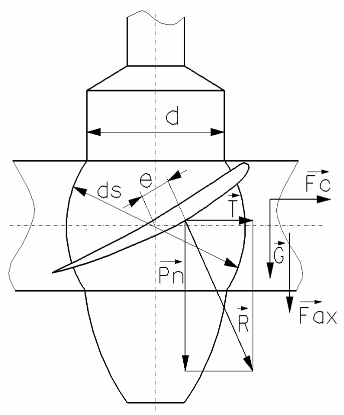

Fig. 4 – The theoretical blade loads

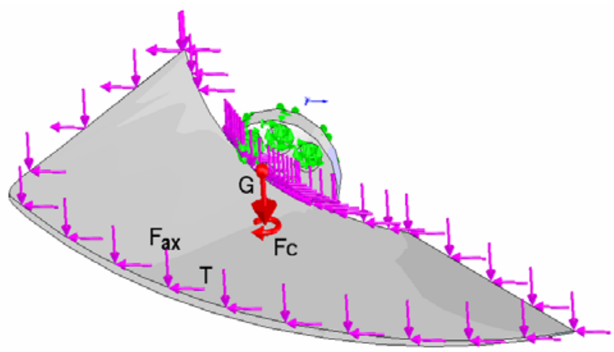

Fig. 5 – The blade loads applied in Design Star.

The inputs data for loads calculus are:

- Z – number of the runner blades; this analyse was made for 4 and 6 runner blades;
- D – the runner diameter;
- n – the runner speed;
- H, Q, P – the turbine head, discharge and power at operating point; the values were selected from the prototype hill chart;
- $F_{ax\ 11}$ – the unit axial force at operating point (n_{11}, Q_{11}); the values were selected from experimental measurement on the turbine model, for n_{11} and Q_{11} calculated values.

The following formulas are applied for loads calculus:

$$n_{11} = \frac{n \cdot D}{\sqrt{H}}, \tag{1}$$

$$Q_{11} = Q \cdot D^2 \sqrt{H} \tag{2}$$

$$F_{ax} = \frac{F_{ax11} \cdot H \cdot D^2}{Z} \tag{3}$$

$$T = \frac{P}{\frac{\pi \cdot n}{30} \cdot b \cdot Z} \tag{4}$$

where: n_{11}, Q_{11} – the unit speed and the unit discharge respectively; b – the distance from tangential force to the runner axis.

There were selected many operating points to calculate loads, points which correspond to the whole turbine operating range.

4. THE BLADE LINEAR STATIC ANALYSE

Linear static analysis calculates displacements, strains, stresses and reaction forces under the effect of applied loads. Required input for linear static analysis are: material properties, meshed model, adequate restraints to prevent the body from rigid body motion and at least one of the following types of loading: concentrated forces, pressure, prescribed nonzero displacements, body forces (gravitational and/or centrifugal). Body forces (gravity load and centrifugal load) are applied to the whole model. The density of the material must be defined for the software to calculate gravity forces and the angular velocity (radians or rpm) must be defined for the software to calculate centrifugal forces. The material of the blade is alloy steel with properties presented in Table 1.

Table 1

The material properties

Elastic modulus	2.1e+011	N/m^2	Mass density	7700	kg/m^3
Poisson's ratio	0.28	–	Tensile strength	7.2383e+008	N/m^2
Shear modulus	7.9e+010	N/m^2	Yield strength	6.2042e+008	N/m^2

The mesh is presented in Fig. 6. The mesh in the stress relieve groove regions is much fine than the rest of the blade (Fig. 7).

Fig. 6 – The blade mesh.

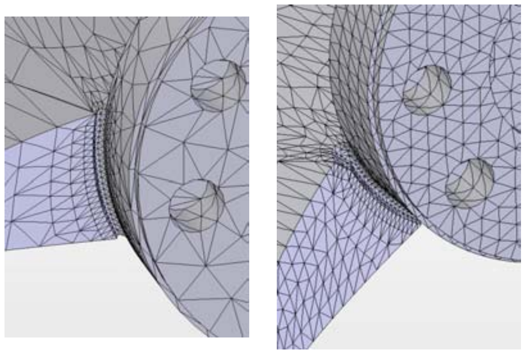

Fig. 7 – The blade mesh in the stress relieve groove area version 1 / version 2.

5. THE LINEAR STATIC ANALYSE RESULTS

The static analyse was performed for multiple operating points and the following variants:

- the blade with stress relieve groove version 1 with loads calculated for 4 blades number of the runner;
- the blade with stress relieve groove version 1 with loads calculated for 6 blades number of the runner;
- the blade with stress relieve groove version 2 with loads calculated for 4 blades number of the runner.

Output of static analysis are: displacement components (translation in *X*, *Z*, *Y* direction and resultant displacement), reaction force (in *X*, *Z*, *Y* direction and resultant reaction force), normal stress σ (in *X*, *Z*, *Y* direction and von Mises stress value), shear stress τ (in *XZ*, *XZ*, *YZ* direction). The Von Mises stress is computed from the six stress components as follows:

$$\sigma_{VM} = \sqrt{\frac{1}{2}\left[(\sigma_X - \sigma_Y)^2 + (\sigma_X - \sigma_Z)^2 + (\sigma_Y - \sigma_Z)^2\right] + 3\left(\tau_{XY}{}^2 + \tau_{XZ}{}^2 + \tau_{YZ}{}^2\right)}. \quad (5)$$

Figs. 8 and 9 exemplify the displacement and von Mises plots calculated for one operating point. The maximum value of deformation and von Mises stress are localised at the periphery respectively at the stress relieve groove on the trailing edge side.

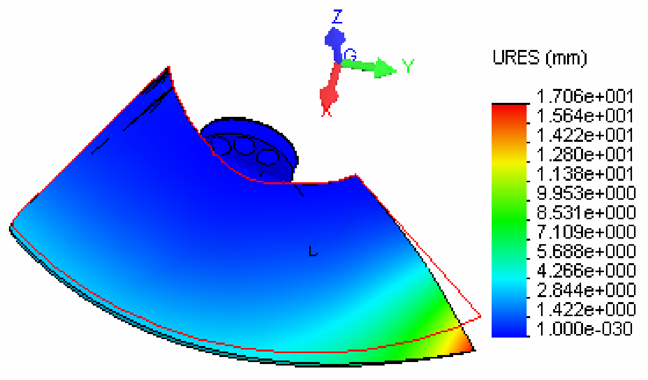

Fig. 8 – The resultant displacement plot.

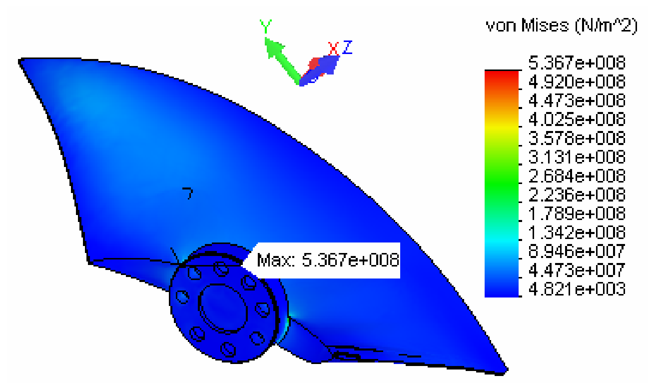

Fig. 9 – The von Mises plot.

From finite element analyze results two stress concentrators on the blade (Fig. 1) one concentrator placed in area *A* on leading edge side and the second placed in area *B* on trailing edge area, the von Mises values from the second concentrators are bigger than the corresponding values of the first concentrator. The Figs. 10, 11 and 12 present the von Mises stress values for the operating points and the three variants analysed, with power *P* as parameter. From this diagrams result the following conclusions:

- the blade with stress relieve groove version 1 and 4 blades has the highest values of the von Mises stress (Fig. 10); the minimal, medium and maximal values are 432, 545 and 623 MPa respectively;
- the von Mises stress values are slightly reduced for the blade with stress relieve groove version 2 and 4 blades (Fig. 11); the minimal, medium and maximal values are 438, 495 and 579 MPa respectively;
- the blade with stress relieve groove version 1 and 6 blades has the lowest values of the von Mises stress (Fig. 12); the minimal, medium and maximal values are 311, 388 and 441 MPa respectively.

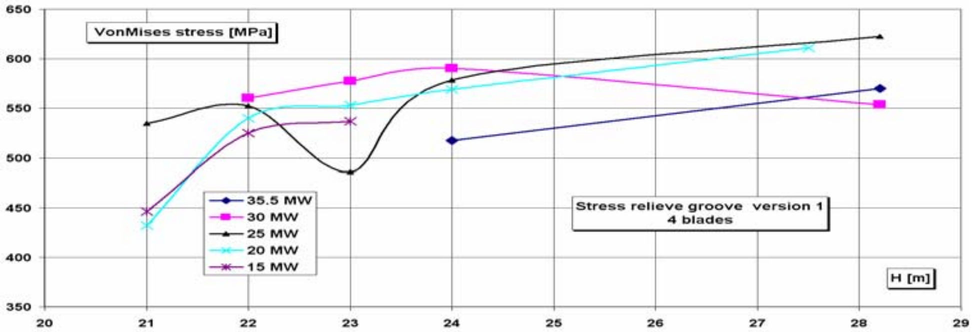

Fig. 10 – The Von Mises stress for stress relieve groove version 1, 4 blades.

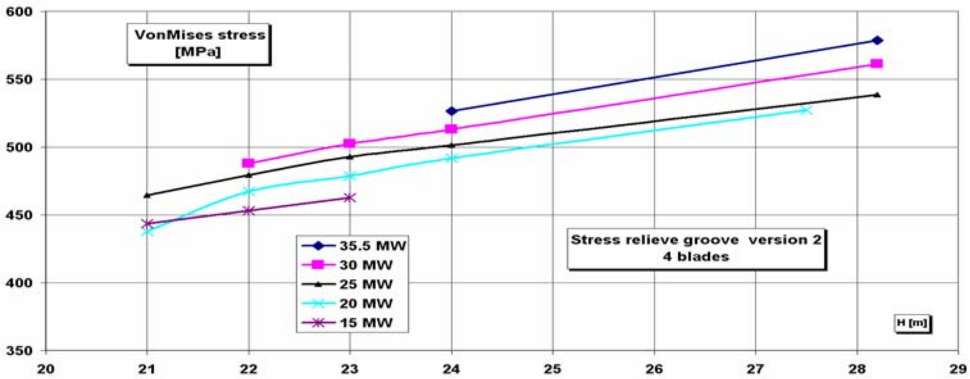

Fig. 11 – The Von Mises stress for stress relieve groove version 2, 4 blades.

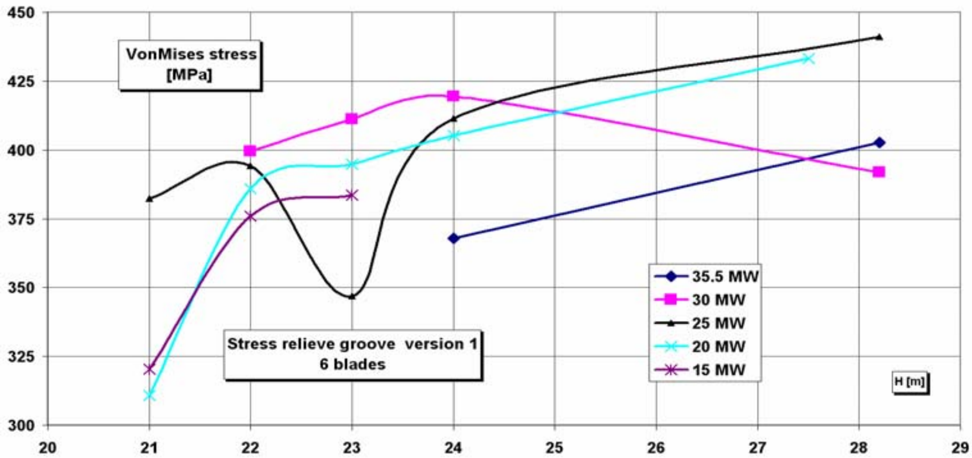

Fig. 12 – The Von Mises stress for stress relieve groove version 1, 6 blades.

6. THE SERVICE LIFE ESTIMATION

For service life estimation of the turbine blades two parameters were calculated [6]:

- the frequency *f* resulted from the neuniformity velocity field of the wicked gates;
- the fatigue cycle number n_c.

$$f = \frac{n \cdot Z_0}{60} \tag{6}$$

$$n_c = n \times 60 \times z_0 \times h \tag{7}$$

where Z_o is the blades number of the wicked gates and *h* the number of operating hours.

From finite element analyze were selected the regimes with maximal values of the von Mises stress. The oscillations amplitude for the blades of the hydraulic turbines is experimental obtained at value 20 MPa [6]. The minimal yield strength of the blade material is $RP_{0.2\,min}$ = 550 MPa.

The calculated von Mises stress values for 4 blades were introduced in Haigh diagrams (Figs. 13 and 14). For stress relieve groove version 1 result the conclusion that the blade fissure should be produced between $10^8 \div 10^{10}$ fatigue cycles and the fissure dangerous appear at a very small number of operating hours (around 1000 hours).

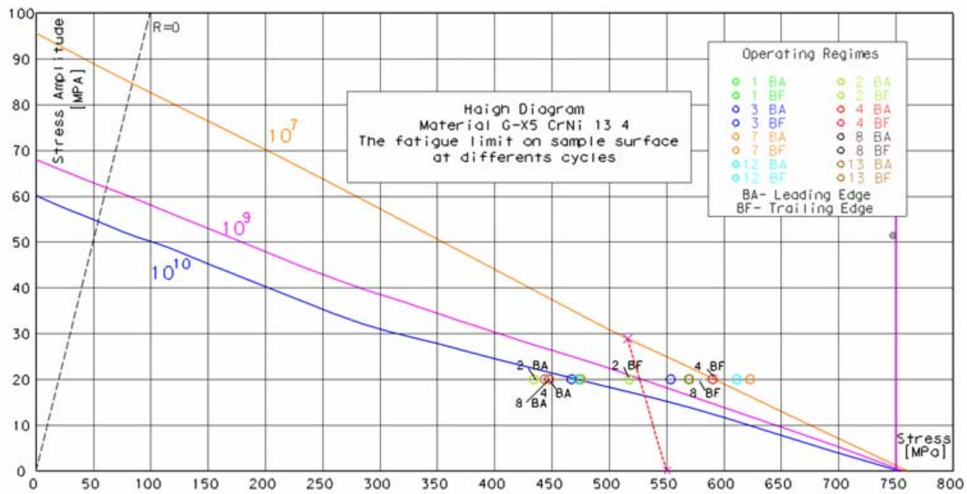

Fig. 13 – The Haigh diagram for stress relieve groove version 1 and 4 blade.

For different version of stress relieve groove geometry the finite elements analyse was recomputed to reduce the von Mises stress values, the version 2 being selected. From Fig. 14 result the conclusion that, for trailing edge, the fissure dangerous appear after 10^{10} fatigue cycles (60 000 number of operating hours) and for leading edge, the fissure dangerous appear after 10^{9} fatigue cycles (10 000 number of operating hours).

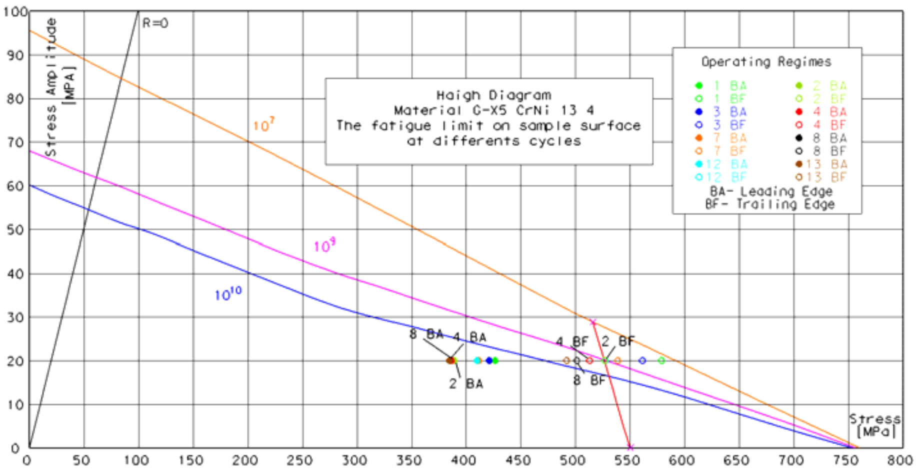

Fig. 14 – The Haigh diagram for stress relieve groove version 2 and 4 blade.

The diagrams for Figs. 13 and 14 are ideal, because the fatigue curves obtained in laboratory correspond to homogenous samples, with no fissures and surfaces defects. From these reasons in the fatigue practice of hydraulic turbines is recommended that the maximal values of the stress concentrators will be maximum half value from yield strength of the blade material, in our case 275 MPa. So, the final conclusions is that this stress relieve groove are not sufficient to reduce the fissure dangerous for a service life of 30 years.

7. CONCLUSIONS

The main reason of the blades fissures for the axial runner turbines is the fatigue corrosion under the loads action. The real fissures appear and developed from the stress concentrator placed to the leading edge side of the blade.

To decrease the maximal stress value it is necessary to reduce the hydrodinamic loads on the blade. For imposed operating conditions (discharge, head, speed, power) the stress decreasing is possible only by increasing the blade number of the runner. The present study demonstrates this idea, with calculus of the same blade geometry for a runner with 6 blades. The results show a reduced stress with maxim 200 MPa, which associated with a optimal geometry of the stress relieve groove, has the effect of stress decreasing below the admissible limits.

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