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Low cycle fatigue analysis of a last stage steam turbine blade

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Abstract

The present paper deals with the low cycle fatigue analysis of the low pressure (LP) steam turbine blade. The blade is cyclically loaded by the centrifugal force because of the repeated startups of the turbine. The goal of the research is to develop a technique to assess fatigue life of the blade and to determine the number of startups to the crack initiation. Two approaches were employed. First approach is based on the elastic finite element analysis. Fictive 'elastic' results are recalculated using Neuber's rule and the equivalent energy method. Triaxial state of stress is reduced using von Mises theory. Strain amplitude is calculated employing the cyclic deformation curve. Second approach is based on elastic-plastic FE analysis. Strain amplitude is determined directly from the FE analysis by reducing the triaxial state of strain. Fatigue life was assessed using uniaxial damage parameters. Both approaches are compared and their applicability is discussed. Factors that can influence the fatigue life are introduced. Experimental low cycle fatigue testing is shortly described.

Keywords: low cycle fatigue, uniaxial damage parameters, LP blade, steam turbine, Neuber's rule, equivalent strain amplitude, experimental LCF testing

1. Introduction

More than 90 % of all service failures of machine components are caused by fatigue. Therefore, fatigue must be taken into consideration especially when designing extremely loaded components like steam turbine blades. Since the requirements for turbine efficiency rise, the loading of the blades also increases. Consequently, new methods how to predict working life of the blades have to be implemented into the blading design process. The evidence that it is necessary to cope with the fatigue of turbine blades are previously reported blade failures. See e.g. Rao [4].

This paper deals with the fatigue analysis of a low pressure blade that is used for the last stages of large steam turbines. The blade is equipped with the fir-tree root assembled in axial groove. The last stage blades being the longest and the heaviest are most severely loaded under the centrifugal forces. As the whole turbine is repeatedly started up and shut down, the varying centrifugal force results in alternating stress inside the blade material. This can lead to crack initiation and consequential failure of the whole blade. Such failure would result in the destruction of the turbine and it may have catastrophic consequences. However, there are not only centrifugal loads that strain the low pressure blades. Destructive dynamic forces can occur if the natural frequency of the blade is in resonance with the harmonics of rotational frequency. The resonance with the nozzle passing frequency is typical for high pressure blades and is usually not dangerous for the low pressure blades. However, the blade under question is robust and stiff enough to avoid the resonance with the lowest multiples of the rotational frequency [2]. For this reason, dynamic forces are considered to be harmless and are neglected.

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Bending moment induced by steam pressure is neglected as well because the magnitude of the bending stress represents only $0.15\,\%$ of the stress caused by the centrifugal loads at the critical location of the blade. Creep yielding does not appear because the last stage blades operate at temperatures under $100\,°\mathrm{C}$. Hence, the only loading taken into account is centrifugal loading. The aim of the fatigue assessment is to determine the number of startups to the crack initiation. Numbers of startups correspond with the number of centrifugal force reversals. Crack initiation is considered to be the end of the service life of the blade.

It is not possible to carry out experimental fatigue testing for all types of low pressure blades. Thus, appropriate computational method of life assessment must be developed for this application. In this case the experimental part of the fatigue analysis reduces to material testing in order to obtain information about the fatigue response of the material. These experiments are performed using simple test specimens. Experimental data for many materials are available in databases. Experimental testing of the real component should be performed only in the case when numerical simulation does not exclude the possibility of fatigue failure during the expected service life of the blade.

The durability of the particular low pressure blade is predicted using various methods available for LCF analysis. The results are then compared to each other and the most suitable method for this application is described. The aim of the research is to compare various methods and to find the most appropriate technique to determine durability of the low pressure turbine blade in terms of low cycle fatigue. The second goal is to determine the working life of the particular low pressure (LP) blade. Numerically predicted working life will be afterwards verified by experimental low cycle fatigue testing.

First, theoretical background and the technique of LCF analysis is mentioned. In the second part the described methods are applied to the particular blade. The results of various methods are then compared and discussed. Next part of the paper deals with factors that can influence fatigue life. The last part shortly describes experimental fatigue testing which is about to be carried out later.

2. Theoretical background

Most existing methods for fatigue assessment are based on elastic finite element analysis. However, when dealing with low cycle fatigue the yield strength of the material is often exceeded and the material becomes plastic. Repetitive plastic deformation is the main cause of low cycle fatigue failure. It is the case of the LP blade. The material in the fir-tree root becomes locally plastic during loading. Therefore, *Local Plastic Stress and Strain Analysis* (LPSA) approach was selected for fatigue assessment.

This method takes into account only the local volume of material around the critical location that goes plastic. The real state of stress and strain at this critical volume is a fundamental input into LPSA method. These values can be obtained either from elastic-plastic FE analysis or from simple elastic FE analysis. However, in the case of elastic FE analysis the elastic results must be recalculated and adapted to the real plastic (and therefore nonlinear) material behavior. The 'plastic adaptation' of linear FEA results is discussed in subsection 2.2. The second essential input into LPSA method are coefficients defining fatigue behavior of the material: σ'_f — fatigue strength coefficient, b — fatigue strength exponent, ε'_f — fatigue ductility coefficient and c — fatigue ductility exponent. The values of these coefficients must be determined experimentally or taken from the database. Material properties are discussed in section 3 in more detail.

The principle of the LPSA method for uniaxial damage parameters is depicted in fig. 1. First, material testing have to be carried out in order to identify the cyclic and fatigue properties of the material: E — Young's modulus of elasticity, υ — Poisson's ratio, K' — cyclic stress hardening coefficient, n' — cyclic stress hardening exponent and four coefficients defining fatigue properties of the material (explained above). Then, FE analysis is performed. The results of the FE analysis are stress and strain components σ_{ij} and ε_{ij} at the critical location of the blade. In the case of elastic-plastic FE analysis, resulting values of stresses and strains are real which is designated by subscript υ. Results of the elastic FE analysis are just fictive (on account of the simplifying assumption of linear material behavior). Afterwards, equivalent stress and strain is calculated. Fictive elastic equivalent upper stress $\sigma_{h,fic}$ and mean stress $\sigma_{m,fic}$ ($\sigma_{m,fic} = \sigma_{h,fic}/2$) have to be recalculated and adapted to the true plastic state in the notch. Strain amplitude is then calculated using cyclic deformation curve. By contrast, true strain amplitude $\varepsilon_{a,\upsilon}$ is the direct output of the elastic-plastic FE analysis. Number of startups to crack initiation is determined using uniaxial damage parameters and linear Palmgren-Miner approach of damage cumulation. Detailed description of every single step of the analysis is introduced below.

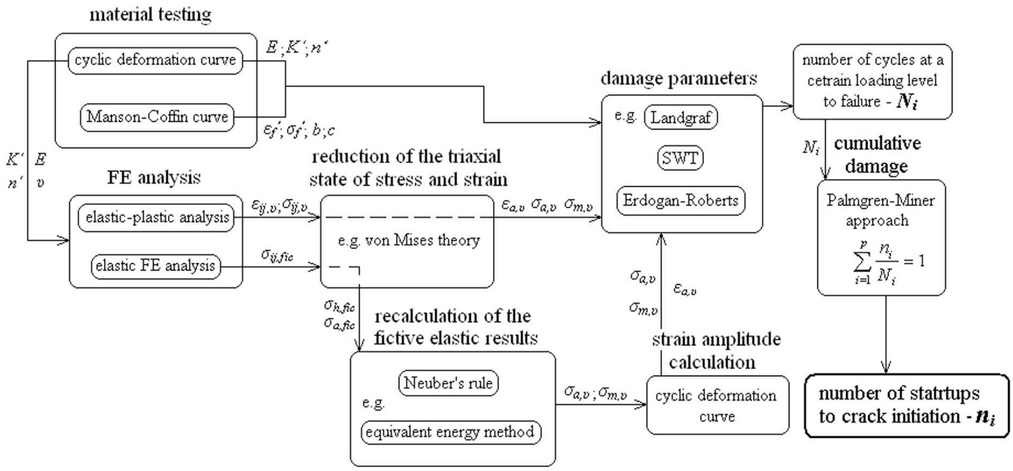


Fig. 1. Schematic diagram of LPSA method

2.1. Reduction of the triaxial state of stress and strain

Uniaxial damage parameters were used to assess fatigue life of the turbine blade. Thus, there was a need to reduce the triaxial state of stress and strain at the critical location of the blade. The fictive uniaxial equivalent stress and strain was then used as an input to the uniaxial damage parameters. Equivalent stress σ_{eq} was calculated using von Mises rule [3]

$$\sigma_{eq} = \frac{1}{\sqrt{2}}\sqrt{(\sigma_{xx}-\sigma_{yy})^2+(\sigma_{yy}-\sigma_{zz})^2+(\sigma_{zz}-\sigma_{xx})^2+6(\sigma_{xy}^2+\sigma_{yz}^2+\sigma_{xz}^2)}, \tag{1}$$

where σ_{ij} is a component of stress acting in direction of i axis in the plane whose normal is j axis. Equivalent strain ε_{eq} was calculated using formula mentioned in [7]

$$\varepsilon_{eq} = \left(\frac{2}{3}e_{ij}e_{ij}\right)^{\frac{1}{2}} \tag{2}$$

that is inscribed using the Einstein summation convention for $i, j = x, y, z$. Strain deviator e_{ij} is defined by equation (3):

$$e_{ij} = \varepsilon_{ij} - \frac{1}{3}\varepsilon_{kk}\delta_{ij}, \tag{3}$$

where ε_{ij} represents the strain component in direction of the i axis acting in the plane having j axis as a normal. δ_{ij} denotes Kronecker delta and ε_{kk} expresses the first invariant of the strain tensor. Equation (2) can be rewritten into following formula:

$$\varepsilon_{eq} = \frac{\sqrt{2}}{3}\sqrt{(\varepsilon_{xx} - \varepsilon_{yy})^2 + (\varepsilon_{yy} - \varepsilon_{zz})^2 + (\varepsilon_{zz} - \varepsilon_{xx})^2 + \frac{3}{2}(\varepsilon_{xy}^2 + \varepsilon_{yz}^2 + \varepsilon_{xz}^2)}. \tag{4}$$

This expression corresponds with the expression for strain intensity ε_i if $\upsilon = 0.5$ is substituted [3]. Poisson's ratio becomes 0.5 when fully plastic conditions develop at the critical location.

2.2. Recalculation of the fictive elastic stresses (elastic FE analysis)

Since the stress at the critical location of the turbine blade exceeds the yield strength of the material it is not possible to presume the linear relation between stress and strain to obtain true results. However, in some cases when the loading history is complicated it is not possible to perform advanced nonlinear analysis for all loading levels. For this reason, only simple linear FE analysis can be performed and fictive 'elastic' results can be recalculated with respect to the nonlinear behavior behind the yield strength of the material. This approach was also used when assessing fatigue of the blade. Two techniques how to recalculate fictive 'elastic' stresses are introduced. The simplest and most commonly used is the Neuber's rule. This rule is based on the assumption that the true strain concentration factor times the true stress concentration factor in the notch equals the theoretical 'elastic' stress concentration factor squared. Second technique that can be used is the equivalent energy method (Glinka's rule). This theory postulates that the strain energy density of the fictive elastic state of stress equals the strain energy density of the real elastic-plastic state. The basic idea of both methods is depicted in fig. 2. Applied methods are explained in [2] in more detail. However, the output of both methods is true stress amplitude in the notch $\sigma_{a,\upsilon}$ and true mean stress in the notch — $\sigma_{m,\upsilon}$. It has been discovered that the results obtained by the equivalent energy method correspond better with the results acquired by the elastic-plastic analysis.

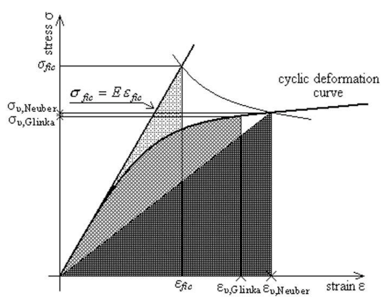


Fig. 2. Basic idea of Neuber's rule and Glinka's rule (equivalent energy method)

2.3. Strain amplitude calculation

In the case of elastic FE analysis true stress amplitude has to be calculated first. Fictive elastic equivalent upper stress $\sigma_{h,fic}$ and mean stress $\sigma_{m,fic}$ have to be recalculated in accordance with subsection 2.2. Then the true stress amplitude can be calculated as the difference between true upper stress and true mean stress. The strain amplitude can be calculated using the cyclic deformation curve which defines the relationship between stress and strain during cyclic loading.

In the case of elastic-plastic analysis, the stress amplitude calculation is slightly more complicated. The elastic-plastic FE analysis must be performed in two steps. First, the state of loading must be analyzed. Then; in contrast to the elastic FE analysis, the state of unloading have be analyzed as well. Since the yield strength of the material is exceeded during loading, residual stress occurs at the fir-tree root of the blade. Hence, the result of the elastic-plastic FE analysis is the set of strain components in the state of loading and in the state of unloading. A method how to determine the equivalent uniaxial strain amplitude had to be developed. Technical standard for pressure vessels [8] was used. This standard, originally developed for assessing stresses, was modified for strains in accordance with equation (4). The idea of this method can be expressed by following equations:

$$\Delta\Sigma_{ij} = \left(\Sigma_{ij}\right)_a - \left(\Sigma_{ij}\right)_b \,, \tag{5}$$

$$\begin{aligned} \Delta\varepsilon_{eq} \;=\; & \frac{2}{3}\sqrt{\Delta\Sigma_{xx}^2 + \Delta\Sigma_{yy}^2 + \Delta\Sigma_{zz}^2 - \Delta\Sigma_{xx}\Delta\Sigma_{yy} - \ldots} \\ & \overline{\ldots - \Delta\Sigma_{yy}\Delta\Sigma_{zz} - \Delta\Sigma_{zz}\Delta\Sigma_{xx} + \frac{3}{4}\left(\Delta\Sigma_{xy}^2 + \Delta\Sigma_{yz}^2 + \Delta\Sigma_{zx}^2\right)} \end{aligned} \tag{6}$$

and

$$\begin{aligned} \Delta\sigma_{eq} \;=\; & \sqrt{\Delta\Sigma_{xx}^2 + \Delta\Sigma_{yy}^2 + \Delta\Sigma_{zz}^2 - \Delta\Sigma_{xx}\Delta\Sigma_{yy} - \ldots} \\ & \overline{\ldots - \Delta\Sigma_{yy}\Delta\Sigma_{zz} - \Delta\Sigma_{zz}\Delta\Sigma_{xx} + 3\left(\Delta\Sigma_{xy}^2 + \Delta\Sigma_{yz}^2 + \Delta\Sigma_{zx}^2\right)}. \end{aligned} \tag{7}$$

Symbol Δ designates double amplitude, Σ physical quantity (stress or strain), $i, j = x, y, z$ and the subscript eq designates equivalent quantity. Subscripts a and b mean state of loading, unloading respectively.

2.4. Damage parameters

The number of reversals of the centrifugal force to the crack initiation at a certain (i^{th}) loading level (certain operational speed of the turbine) is determined using uniaxial damage parameters. True equivalent strain amplitude, true equivalent mean stress and true equivalent stress amplitude (scalar quantities) are inputs into these damage parameters. Three damage parameters were used. Their detailed description is available in the literature [2, 6]. The damage parameter for *Landgraf's method* — P_L — is strain amplitude ε_a. The fatigue life equation (right side of the equation (8)) for this method is based on the Manson-Coffin curve. Number of cycles to crack initiation N_i can be determined using equation

$$P_L = \varepsilon_a = \frac{\sigma_f' - \sigma_m}{E}(2N_i)^b + \varepsilon_f'(2N_i)^c. \tag{8}$$

Another method used is *Smith, Topper, Wetzel (SWT) method*. The damage parameter for this method — P_{SWT} — is equivalent strain amplitude (left side of the equation (9)). The fatigue life equation is based on Manson-Coffin curve again. Number of force reversals to crack initiation N_i according to the SWT method can be calculated using equation

$$P_{SWT} = \sqrt{(\sigma_a + \sigma_m)\varepsilon_a E} = \left[{\sigma'_f}^2 (2N_i)^{2b} + E\sigma'_f \varepsilon'_f (2N_i)^{b+c} \right]^{0.5}. \tag{9}$$

The third damage parameter used is *Erdogan-Roberts's parameter*. The damage parameter for this method – $P_{E,R}$ — is the left side of the equation (10). The fatigue life equation stays the same as in the case of SWT parameter. Number of cycles to crack initiation N_i according to the E-R method can be calculated using equation

$$P_{E,R} = \sqrt{\sigma_a^\gamma (\sigma_a + \sigma_m)^{1-\gamma} \varepsilon_a E} = \left[{\sigma'_f}^2 (2N_i)^{2b} + E\sigma'_f \varepsilon'_f (2N_i)^{b+c} \right]^{0.5}. \tag{10}$$

2.5. Cumulative damage

The loading spectrum of the turbine consists of one balancing cell over-speed test (typically 3 600 rpm), app. 40 over-speed tests (typically 3 300 rpm) and unknown number of startups (operating speed typically 3 000 rpm). The aim of the research was to determine the number of startups to the crack initiation. However, additional cycles of balancing cell over-speed test and over-speed tests have to be incorporated to the damage cumulation. This was performed employing the linear Palmgren-Miner rule that is described by the equation

$$\sum_{i=1}^{p} \frac{n_i}{N_i} = 1, \tag{11}$$

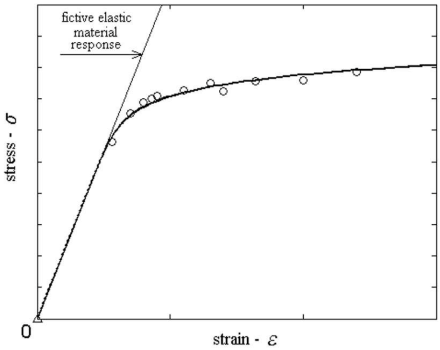
where n_i is the number of cycles at the i^{th} loading level, N_i designates the number of cycles to crack initiation at the i^{th} loading level and p is the number of loading levels (in this case $p = 3$).

3. Application: turbine blade fatigue assessment

3.1. Material properties

The LP blade is made of high strength steel X12CrNiMo12. The turbine rotor is made of 27NiCrMoV15 – 6 stainless steel. In order to obtain cyclic-fatigue properties of both blading and rotor material, experimental testing must have been carried out. It was performed using simple cylindrical specimens in the regime of controlled deformation. Temperature during the test was $100\,^\circ$C which corresponds with the operational temperature of a turbine last stage. The test specimens were loaded at the frequency range of 0.3–1.75 Hz. Stress, strain, tensile force and the number of cycles to failure were recorded during the test. By the interpolation of the test results the cyclic deformation curve and the Manson-Coffin curve of both materials was identified. Both curves for the blading material are depicted in fig. 3. The cyclic deformation curve can be defined by coefficients K' and n' and defines the relation between stress and strain during cyclic loading. Therefore, it characterizes the cyclic behavior of the material. The Manson-Coffin curve is described by parameters σ'_f, b, ε'_f and c. This curve gives the relationship between the number of reversals to failure and the strain amplitude. Thus, it characterizes the fatigue behavior of the material.

Since the data record that contains hysteresis loops of the loading during material testing is not available it was not possible to establish absolutely truthful material hardening model. However, the kinematic hardening material model was used with respect to principles of the Bauschinger effect that takes place during saturated cyclic loading.



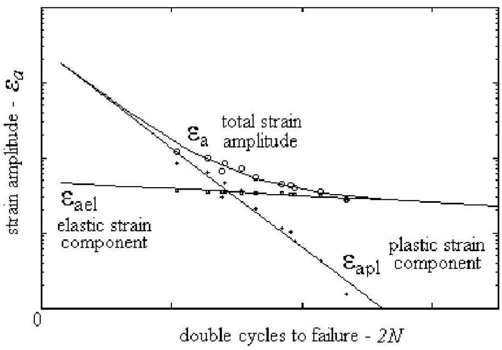


Fig. 3. Cyclic deformation curve (left); Manson-Coffin curve (right). note: Numerical values are confidential; hence, they have been deleted

3.2. FE analysis

The FE analysis was performed using the finite element computational software ANSYS 11.0. The intention of the FE analysis was to determine the stress and strain components at the critical location of the blade. It has been found out that the critical location of the blade is situated at the fir-tree root of the blade. More precisely, at the convex side of the dovetail underneath the entering vane edge. However, it is required to assess the fatigue life of the whole blade-disc connection. Therefore, the critical spot at the disc groove has to be located as well and the fatigue life of the rotor disc has to be evaluated separately (disc is made of different material). Hence, the whole blade-disc system has to be analyzed.

Bladed turbine disc is a cyclic symmetrical structure. For this reason, the dimension of the analysis can be reduced. Modeling only one blade and the belonging disc section is representative for the whole structure. Nevertheless, the effects of the surplus disc section that is not modeled must be compensated by the boundary condition of cyclic symmetry. The effect of remaining parts of the rotor is replaced by the axial displacement boundary condition. The situation is depicted in fig. 4.

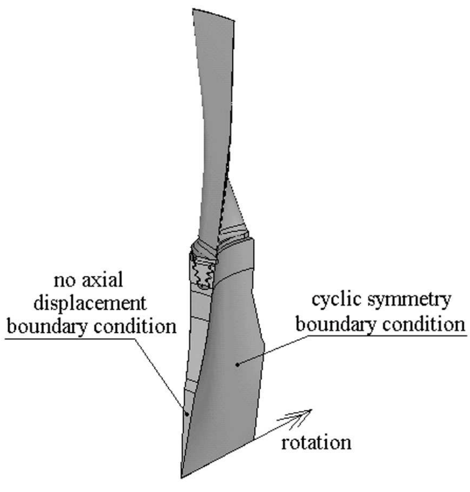


Fig. 4. Assembly of LP turbine blade and disc section; boundary conditions

Surface-to-surface contact was defined at the contact surfaces between the blade and the groove in the disc. For this reason, the analysis becomes nonlinear. Linear elastic isotropic material model was used for the elastic FE analysis. For the elastic-plastic FE analysis nonlinear inelastic rate independent isotropic model was employed. Kinematic hardening rule was chosen for this model. Nonlinear material behavior was defined by the multilinear cyclic deformation curve. The whole structure was loaded with rotation of $314.16\,\mathrm{rads}^{-1}$. This corresponds with the operational speed of 3 000 rpm.

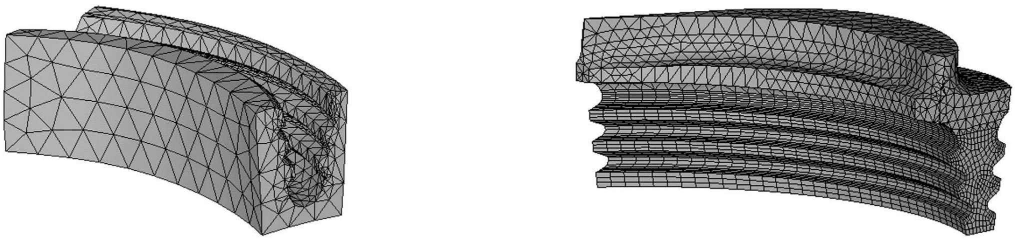


Fig. 5. Detailed depiction of the finite element mesh at the critical location

The finite element mesh was built as a combination of free and mapped mesh. 3D 20-node quadratic elements SOLID95 were used. See figure 5 for detailed depiction of the finite element mesh of the blade disc connection.

Since the analysis is nonlinear, the number of degrees of freedom of the model is limited to app. 300 000 because of extremely long computational times. Therefore, the technique of submodeling must have been introduced in order to obtain accurate results at the locations of stress concentration. Two submodels were used to calculate stresses and strains at the critical locations of both blade and disc. Number of degrees of freedom of submodels was limited to app. 300 000 as well. Elastic-plastic analysis was performed in two steps as described above: first step for the state of loading, second step for the state of unloading. Only one loading-unloading cycle was performed because of long computational times. The more cycles is performed, the better results; however, the saturated state is modeled and the results should be representative even if only one cycle is carried out [9].

3.3. Fatigue life assessment

The results of FE analysis at the critical location of both blade and disc were postprocessed in accordance with the methods described in section 2 and substituted into uniaxial damage parameters. See equations (8)–(10). Although, the loading spectrum of the turbine contains three loading levels (see subsection 2.5) in fatigue life assessment only startups to the operational speed of 3 000 rpm will be considered. The fatigue life assessment for every additional loading level requires performance of six extra FE analyses (elastic and elastic-plastic analyses including submodeling). However, approximate estimation can be made using elastic results and neglecting nonlinearity caused by contact between the blade and the disc. According to this guess the number of cycles would decrease by 15 % if one balancing cell over-speed test and forty over-speed tests are included into the process of damage cumulation.

Table 1 summarizes the predicted number of startups to failure for the blade and for the disc — N — calculated according to the technique described above using different damage parameters. In the case of elastic FE analysis, Glinka's method was used to recalculate fictive 'elastic' results.

Table 1. Number of startups to operational speed of 3 000 rpm to crack initiation

		elastic FE analysis + Glinka's rule	elastic-plastic FE analysis
		N	N
blade	Landgraf	598	4 027
	SWT	**527**	**4 436**
	Erdogan-Roberts	604	9 587
disc	Landgraf	2 869	7 298
	SWT	**2 517**	**6 742**
	Erdogan-Roberts	2 925	10 819

4. Discussion

It can be observed that the fatigue life prediction based on linear elastic FE analysis is significantly more conservative in comparison with the prediction based on elastic-plastic FE analysis. This may have several reasons. Whilst the equivalent strain amplitude — which is the decisive variable for low cycle fatigue failure — is determined directly from the elastic-plastic FE analysis, the equivalent strain amplitude in the case of elastic analysis is calculated using the cyclic strain curve and the equivalent stress amplitude. The reduction of the triaxial loading state is done in stresses in contrast to the elastic-plastic analysis where this reduction is done directly in strains. This can lead to overestimation of shear stresses in the critical location and incorrect determination of the equivalent strain amplitude in the notch in the case of elastic FE analysis. Since the Manson-Coffin curve is relatively flat in the considered section, even a little change in strain amplitude can lead to dramatic change in the number of cycles to failure. In the case of the prediction based on elastic-plastic FE analysis true strain amplitude is the direct output of the FEA. For this reason, the fatigue assessment based on the elastic-plastic FE analysis is considered to correspond better with reality.

Another piece of knowledge resulting from the research is that the predicted number of cycles to crack initiation is very sensitive to the input data. The reason is the shape of the cyclic deformation curve. Small inaccuracy in stress determination leads to the significant difference in strain. Hence, the precise determination of the true state of stress and strain at the critical location is crucial. Another fact that can be deducted is that the results of various damage parameters differ from each other. In some cases even by more then 100 %. Hence, the appropriate damage parameter has to be chosen or the most conservative one should be considered.

5. Factors that influence the fatigue damage cumulation process

Most factors introduced in literature e.g. [5] like stress concentration factor, notch sensitivity etc. need not be considered just because of the principle of the LPSA method. The LPSA method considers only the local volume of the material at the critical location of the real component and the true stress and strain conditions within this volume are then compared to the material experimental data. However, it is necessary to take into account some factors when analyzing a machine component in terms of fatigue.

5.1. Temperature and environment

The LP turbine blade operates usually at the temperatures below $100\,^{\circ}$C. Thus, creep yielding should not occur. But, as described in literature [5] even the temperatures below $400\,^{\circ}$C can shorten the predicted working life of the blade. Nevertheless, material testing was carried out at the temperature of 100 °C. Thus, the influence of the temperature has already been included into prediction.

The real LP blade works usually under the hostile environment of flowing steam. This may worsen the fatigue response of the blade in comparison to the numerical prediction which is based on atmospheric material testing.

5.2. Loading frequency

Test specimens were loaded at the frequency range of 0.3–1.75 Hz. Whereas, one reversal of the centrifugal force takes weeks or months. This can reduce the number of cycles to blade failure if set against the theoretical prediction. As the lower the frequency the worse the fatigue response. The reason for this phenomenon is that the plastic state can be fully developed in case of very low frequencies. In addition, long delay at the stage of full loading worsens fatigue response even more. However, it is very complicated to quantify the influence of the loading frequency.

5.3. Size of the exposed volume

The critical location at the turbine blade root is the spot of the steep stress gradient. On the other hand, the test specimen has no location of meaningful stress concentration. Hence, the exposed volume at the critical location of the turbine blade is significantly smaller than the exposed volume of the test specimen. As the literature [1] introduces the specimens with bigger exposed volume show worse fatigue response. The probable reason is bigger probability of occurrence of inclusions and structural imperfections of the material. Thus, the numerically predicted fatigue life assessment may underestimate the number of cycles to failure due to the small exposed volume at the critical location of the blade.

5.4. Mean stress effect

The mean stress effect is incorporated in the damage parameters in the form of correction of the Manson-Coffin curve (see e.g. equation (8)). However, the influence of the mean stress in the case of low cycle fatigue is lower than the one in case of high cycle fatigue. This fact can be substantiated by the mean stress relaxation. Therefore, in the case of the turbine blade where tensile mean stress occurs, the real fatigue response may be better then the one predicted due to the mean stress relaxation.

5.5. The influence of the balancing cell over-speed test on further cyclic behavior

During the balancing cell over-speed test the bladed rotor is rotated at the speed of 3 600 rpm while the operational speed of the turbine is 3 000 rpm. Since the balancing cell over-speed test represents always the first cycle of the loading history it can influence the cyclic response of all following cycles. Extensive plastic deformation develops at the critical location during this test. After unloading a significant residual stress around the volume that went plastic originates. This phenomenon is similar to autofrettage.

The material response during the very first loading is characterized by the tensile curve obtained by the tensile test. First unloading follows the principle of isotropic hardening. Hence, the first cycle causes the decline of the mean stress. The resulting mean stress is the compressive mean stress in contrast to the tensile mean stress in case of loading by the operational centrifugal forces only. With respect to this fact, the predicted fatigue life may increase by app. 100 %.

However, if the stress relaxation effect is considered the improvement of fatigue response is not so significant.

6. Low cycle fatigue testing of the turbine blade root

It has been found that the number of cycles to the crack initiation is very sensitive to the input data preciseness and the method of fatigue assessment. The selection of the damage parameter is important as well. Many factors discussed in section 5 can influence the fatigue damage cumulation process. In addition, many uncertainties such as inaccurate experimental material testing or boundary condition simplification during the FE analysis may be introduced into the fatigue assessment. For this reason, the only reliable way how to determine the true fatigue life of the blade-disc connection is to perform experimental fatigue testing.

It is technically not possible to repeatedly start up and shut down the whole bladed rotor. Hence, the simplified fatigue testing was designed. It can be performed using standard testing machine. The test specimen was designed in order to have same stress distribution as the real blade root. In addition, the jaws were designed to transmit the loading from the pin of the testing machine to the test specimen. Both specimen and jaws are depicted in figure 6.

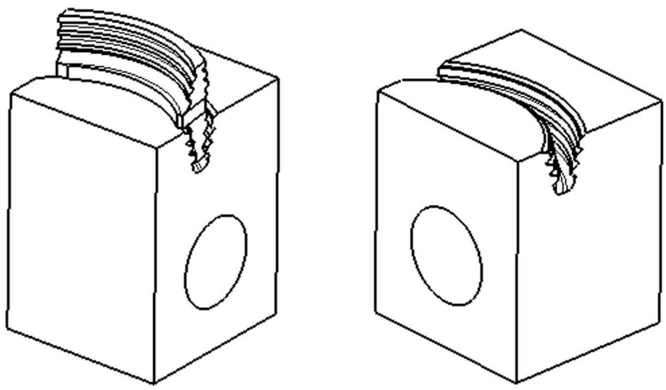


Fig. 6. Low cycle fatigue test specimen and jaws

The main goal of the experimental testing is to determine true number of centrifugal force reversals to the crack initiation. Therefore, the moment of crack initiation has to be revealed using defectoscopy. The test specimen will be equipped with the strain gauge measurement in order to verify the predicted stress distribution. The second goal of the experimental testing is to obtain data which can be used for further debugging of the computational methods.

7. Conclusion

The present paper has attempted to investigate the fatigue response of the turbine blade in terms of low cycle fatigue. The goal of the research was to establish the technique of the low cycle fatigue assessment of the LP turbine blade equipped with the fir-tree root and to determine the number of startups to the crack initiation of the particular LP blade. The analysis has shown excessive sensitivity of the number of cycles to failure to the input data and the method chosen. The fatigue analysis was performed using the uniaxial damage parameters. First, the technique based on linear FE analysis including the way of recalculation of the fictive 'elastic' results was described. However, it has been concluded that this approach is not appropriate for this particular application for the reasons stated in section 4. Therefore, the fatigue assessment based on the elastic FE analysis should not be used. It is necessary to perform nonlinear FE analysis to obtain true state of stress and strain at the critical location of both blade and disc groove. The approach to calculate the equivalent strain amplitude is shown in subsection 2.3. Three uniaxial damage parameters were used to assess fatigue life of the blade-disc connection and the results were compared. It has been found out that the results differ significantly. Hence, the SWT damage parameter is recommended as being the most conservative in most cases.

In addition, a strong sensitivity of the cycles to failure to the input data has been observed. For this reason, precise values of stress and strain in the notch have to be determined. Thus, submodeling is required when performing FE analysis.

Furthermore, there are many factors that can influence the process of damage cumulation that are not included in the prediction. Mostly, it is very complicated or impossible to quantify the influences. Thus, an adequate safety factor has to be used.

Since the maximal number of startups of the turbine is 3800 in a lifetime, according to the technical standard ČSN EN 60045-1, 'Steam turbines, Part 1: Specification', even the progressive fatigue assessment method based on elastic-plastic FE analysis does not exclude the possibility of the blade failure before the end of the expected lifetime. For this reason, the experimental low cycle fatigue testing of the blade-disc connection has to be performed. The result of the experimental testing should be the true number of startups to crack initiation. Experimental testing may be used for the further debugging of the computational methods. If the experimental testing confirms bad numerical prediction some design changes of the blade have to be implemented in order to increase the durability of the blade-disc connection.

In the future, the fatigue analysis using multiaxial damage criteria will be performed. However, an advanced fatigue postprocessor and additional material testing is required to carry out such an analysis.

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References

[1] G. F. Harrison, P. H. Tranter, D. P. Shepherd, T. Ward, Application of multi-scale modelling in aeroengine component life assessment, Materials Science and Engineering A365 (2004) 247–256.

[2] P. Měšťánek, Určení životnosti lopatek závěsů parních turbin, MSc thesis, University of West Bohemia, Pilsen, 2008.

[3] F. Plánička, Z. Kuliš, Základy teorie plasticity, Czech Technical University in Prag, Prag, 2004.

[4] J. S. Rao, Application of Fracture Mechanics in the Failure Analysis of a Last Stage Steam Turbine Blade, Mech. Mach. Theory Vol. 33 (1998) pp. 599–609.

[5] M. Růžička, M. Hanke, M. Rost, Dynamická pevnost a životnost, Czech Technical University in Prag, Prag, 1989.

[6] M. Růžička, J. Papuga, Metody pro hodnocení únavové životnosti, Czech Technical University in Prag Report No. 2051/99/32, Prague, 1999.

[7] A. Seweryn, A. Buczynski, J. Szusta, Damage accumulation model for low cycle fatigue, International Journal of Fatigue 30 (2008) 756–765.

[8] Czech technical standard ČSN-EN 13445-3, Nonheating pressure vessels, 2003.

[9] J. Jurenka, J. Papuga, M. Růžička, Numerická simulace vzniku únavového poškození čepového spoje, Czech Technical University in Prag, Report for Škoda Power, a. s., Prague, 2008.