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Modelling of the Baumann turbine stage operation Part II. Free and kinetic vibrations

In this paper has been presented a methodology of validation a novel mathematical model dedicated to evaluation and prediction of material degradation and damage of steam turbine elements such as blades, valves, and pipes due to three mechanisms: stress-corrosion, high-temperature creep and low-cyclic fatigue. The validation concept is based on an experimental setup manufactured in the Laboratory of Faculty of Mechanical and Power Engineering, Wrocław UT. The concept of validation by comparison of measured and numerically predicted eigen-frequencies and eigen-modes of different turbine elements within laboratory conditions are presented, and mathematical models of three damage mechanisms have been described. Using the mentioned method of experimental validation based on comparisons of eigen-frequencies, we could calibrate yet unknown coefficients in the turbine damage model. A practical aim is an implementation of a novel life-time module for the BOTT (block of thermal stresses restriction) system. In particular the stress-corrosion factor will be added for the advanced numerical control system, creating in such way a universal, flexible and a complete tool for monitoring degrees of degradation, corrosion and damage of critical points in a steam turbine.

Nomenclature

A, n, R_n	–	constants of the calibrated model
c_α	–	chemical reaction product, where $\alpha =$ H_2Ca, CaO

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D_{ijkl}	–	elements of viscosity tensor
D	–	damage parameter
d_{ij}	–	components of deformation tensor, where $i, j = x, y, z$
E	–	Young's module
F	–	domain elastic response
J_2	–	second Cauchy invariant based on deviator stress
$J_i^{stress}, J_i^{chem}, J_i^{cycle}$	–	diffusion fluxes coming: stress, chemical components and cycle damage, where $i = 1, 2, 3$
R_e	–	yield limit
R_w	–	isotropic strengthening
R_w^{∞}	–	limits of the hardening
s^c	–	source of corrosion formed local chemical reactions, where $c =$H$_2$Ca, CaO
s_{ij}	–	components of stress deviator, $i, j = x, y, z$
s_{α}	–	chemical source product
T, T_{ref}	–	temperature, respectively: actual and referential
α_{ij}	–	kinematic strengthening
β	–	heat transfer coefficient
ε_{ij}	–	components of total strain tensor
ε_{ij}^{el}	–	components of strain tensor, where el, p, θ, c, ch, TP, PT respectively: elastic, plastic, thermal, creep, chemical, plastic strain induced by changes phase and changes phase induced by plastic strain
λ	–	parameter of the plastic flow
μ, λ	–	Lame's elastic constants
ν	–	Poisson's ratio
σ_{HMH}	–	reduced stress, Huber-Mises-Hencky
σ_{ij}	–	components of the stress tensor, $i, j = x, y, z$
ϕ	–	potential of plastic flow

Subscripts

$(\dot{\ })$	–	time derivative
δ_{ij}	–	Kronecker delta

1 Introduction

This study extends the previous works of *Modeling of the Baumann turbine stage* [10]. Here, our intention is to develop a universal research tool based on CSD† (computational solid dynamics), whose purpose is verification of stresses, temperature and dynamics, inaccessible to measurement in the critical points and other places in turbines or valves. The aim is to obtain a complete system for evaluation and online supervision of sensitive parts in the turbogenerator, e.g. blades of first and last stages, a part of rotor located near to the inlet and outlet steam in HP, MP turbine(high and medium pressure), screen in valves, body valves and nozzles and gates in valves. This model should contain construction modules, which are fully verified with measurements performed *in situ*. This monitoring and control module should be capable of estimating the degree of corrosion‡ — first of all the corrosion induced by stresses, high temperature, and low cyclic. More insight to this problem the reader can find in our papers [1–3,12].

Taking into account a highly complex nature of turbine damage, a dedicated mathematical model for damage evolution and estimation of irreversible strain should correctly describe thermal, creep, instantaneous plastics, rate of local chemical reactions and respectively, rate of strain phase transition induced by plasticity and other phenomena [5,6]. Such an extensive diagnostic tool requires, due to a number of generally undefined coefficients, a significant volume of experimental verifications. So, the extensive diagnostic tools require, due to generally undefined coefficients for calibration, more experimental verification.

This article presents a concept of the model verification and calibration based on numerical and experimental comparison of dynamics of blades in different state — from a completely new blade to the blades after one, two, three and more years of operation. The first step of our concept is a designation of referential (new, not used) state for the blade under consideration. Next, the differences in work

†CSD (computational solid dynamics) is an authorial name created 25th years ago by prof. Janusz Badur. This methodology is based on the equations of balance mass, momentum and energy (by the analogy to computational fluid dynamics), resolved simultaneously on a one finite volume or finite element method discretization. This suggestion was the basis of the methodology developed of the Energy Conversion Energy Department, 25th years ago.

‡Stresses induced corrosion – it is a case when corrosion is modeled by a function the concentration of components, for instance, hydrogen which is manifested by the so-called hydrogen embrittlement [8]. Thermal induced corrosion, generally due to high temperature chemical reactions, depends mainly on the temperature and chemically aggressive compounds like HCl and H_2S. Low-cyclic induced corrosion – it has been modeled by a electrochemical corrosion progress function. It depends mainly on variable thermomechanical loads for cycle, start up, nominal operation, shut down, and stopping.

conditions in the laboratory, at a turbine and in the numerical model should be explained. Further, based on the measurements of the old blades, by comparison with the progressive simulated, damage, stress-corrosion and low-cycle fatigue, we will be ready for verification of the mathematical model and its implementation to a commercial FEM code [1]–[8].

2 The experimental setup

As the referential blade there has been adopted a penultimate stage blade taken-off from the steam turbine K200-130 [9,13]. This blade is located in the Laboratory of Faculty of Mechanical and Power Engineering, Wrocław University of Technology. All relevant, referential blade data were taken at that location.

A set of vibration measurements were performed by using professional tools Vibxpert [4] on the experimental setup, which enables to impose various boundary conditions. The first one is a rigid fixing of a blade foot and in a second one, a rigid fixing of food blades simultaneously with the rigid fixing of a surface of the shelf for steam flow separation. As the experimental setup cannot simulate stiffness coming from the damping wire, therefore we take the wires only as a possibility in the CSD simulation. In Fig. 1 there has been shown a photo of the referential blade, and its geometry used in the CSD analysis.

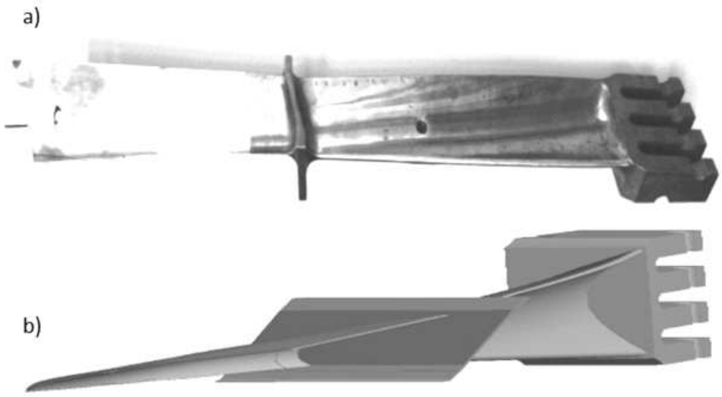

Figure 1. The reference blade for the Baumann stage: a) photo, b) 3D geometry used in CSD analysis.

In the vibration diagram (Fig. 2) there has been shown two peaks identified as a first and second natural frequencies of a free vibration blade. The first one is 106 Hz and it can be identified as a bending mode.

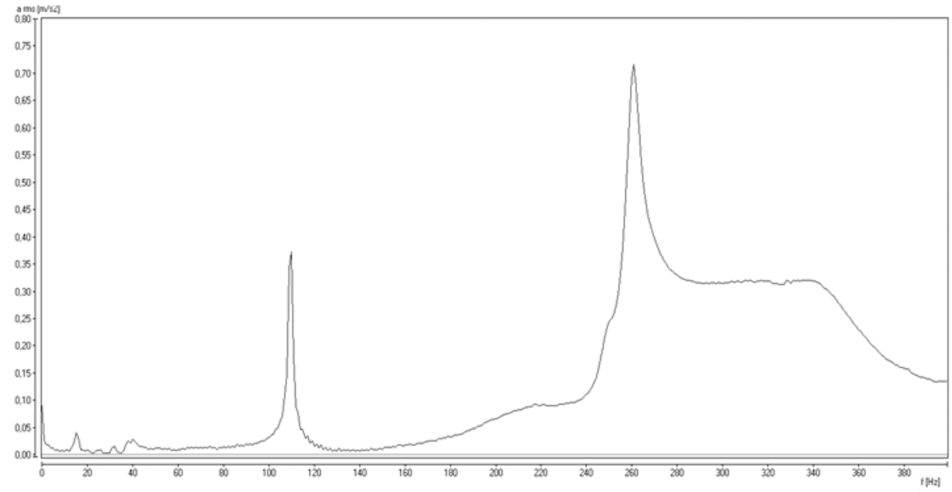

Figure 2. The measured curve obtained from the experimental setup for the blade rigidly fixed in a foot blade and rigidly fixed on a surface of the steam flow separator. The first obtained eigen-frequencies are: f1 = 106 Hz, f2 = 265 Hz.

The second peak is 256 Hz and it is also interpreted to be a bending mode. A specific blade fixing approaching the experimental setup and excitation has been made by a modal hammer, excludes the possibility of torsion modes excitation. It is worth nothing that the range of frequencies obtainable with the used modal hammer does not exceed the maximum of 400 Hz, therefore that restriction may have influenced our measurement and the overall performance of the presented verification method.

3 Numerical results by CSD approach

Numerical analysis of the eigenvalue problem has been performed using a standard finite element method (CSD-equation of balance resolved simultaneously on the finite element discretization region) commercial tools [10], assumed to carry out harmonic computational simulation by using four type of blade fixing. In the first type, the numerical fixing of boundary nodes corresponds to the fixing used at the experimental setup. The second type is due to additional extra fixing in a hole for the damping wire. Third one is similar for the first numerical fixing, modified with adding rotation at 3000 rpm, and the last type, similar to the second numerical fixing with added rotation at 3000 rpm. Results from the second, third and fourth numerical fixing types there are compared to the first numerical fixing type that is verified by the experimental measurement. Such methodology has been applied due to the fact that it was impossible to reflect real conditions of operating blade in the experimental setup. Therefore, we are using this numerical analogy between the first and the other fixing type, for which experimental measurements have been used for verification.

The comparison of the results of numerical and experimental analysis is shown in the Tab.1. In the experimental measurements have been found two natural frequencies: first one is 106 Hz and second 265 Hz, being a third natural frequency. It means that in the experiment there has not been found a second mode. In the second column the frequencies resulting from validation of the numerical CSD model with experimentally obtained results have been shown. Values of the first and third frequencies follow the experimental results very accurately. Note that at the time of experimental analysis the third mode has been treated to be a second mode. Such situation is not found in the experiment, because this mode is a torsional one. It means that the torsional mode, in the first type of fixing, cannot be excited. Value of its this natural frequency is 239 Hz.

Modes of vibration corresponding to the natural frequencies for the first fixing type are presented in Fig. 3. The first mode (Fig. 3a) is a bending mode, whereas the second one (Fig. 3b) being a torsional one, and they cannot by excited using the experimental setup. The last mode (Fig. 3c) is a bending mode of a higher order. In the third column of Tab. 1. a frequency referring to the additional wire fixing with the hole is shown. Value of the frequency increased from 377 Hz to 521 Hz. When the blade rotation was added (rotation speed 3000 rpm) to the second type fixing, frequency decreased from 377 and 421 Hz to 341 and 516 Hz respectively with lower frequency values corresponding to fourth fixing (which is the best approximation to the real operating blade in the turbine stage). This anomaly is caused by a complex geometry of this blade[§] (strong angular torsion and strong profile blade changes).

The Campbell vibration diagram for the analyzed blade with all types of fixings is presented in Fig. 4. Note, that full dots represent the measurement data, one star (*) (see f1*, f2*, f3* in the diagram) refer to the results of simulation with the first type fixing applied (for 0 rpm) and the third type fixing (for 3000 rpm). Double stars (**) (f1**, f2**, f3** in the diagram) refer to results of simulation used second fixing type (for 0 rpm) and fourth fixing type (for 3000 rpm). Double-starred lines (**) show analogical results to those achieved during the experiment.

[§]A strong angular torsion, causing by turbine blade and strong blade profile changes, causes introduction (apart from the tension force, acting always perpendicular for rotating axis) bending force (component of vector force), causing stronger amplitude of vibration in the high fields blade (see: Badur *et al.* [4]).

Table 1. Results of experimental and numerical (computational solid dynamics) analysis.

No.	Experiment [Hz]	First fixing [Hz]	Second fixing [Hz]	Third fixing [Hz]	Fourth fixing [Hz]
1	106	95	377	149	341
2	–	239	505	271	499
3	265	267	521	293	516

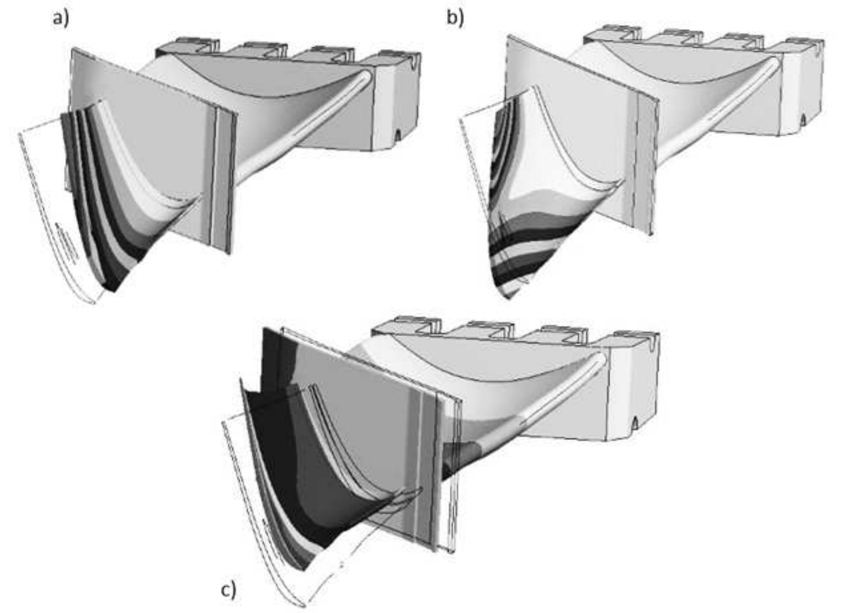

Figure 3. Mode of natural frequencies for the first fixing type.

Natural modes resulting from the use of the second fixing type can be inspected in Fig. 5. It is worth noting that the second mode is changing from a torsional to a bending one. Natural frequency of this (second) mode is 505 Hz. As this value exceeds the maximum frequency value obtainable with modal hammer excitation, great care must be taken while constructing analogy as intended. Furthermore, the diagram in Fig. 5 cannot be treated as a precise Campbell diagram of the Baumann blade for design purposes, since the working conditions of a 200 MW steam turbine are quite different. It must be remembered that the laboratory facility never reproduces the real conditions in which the blade

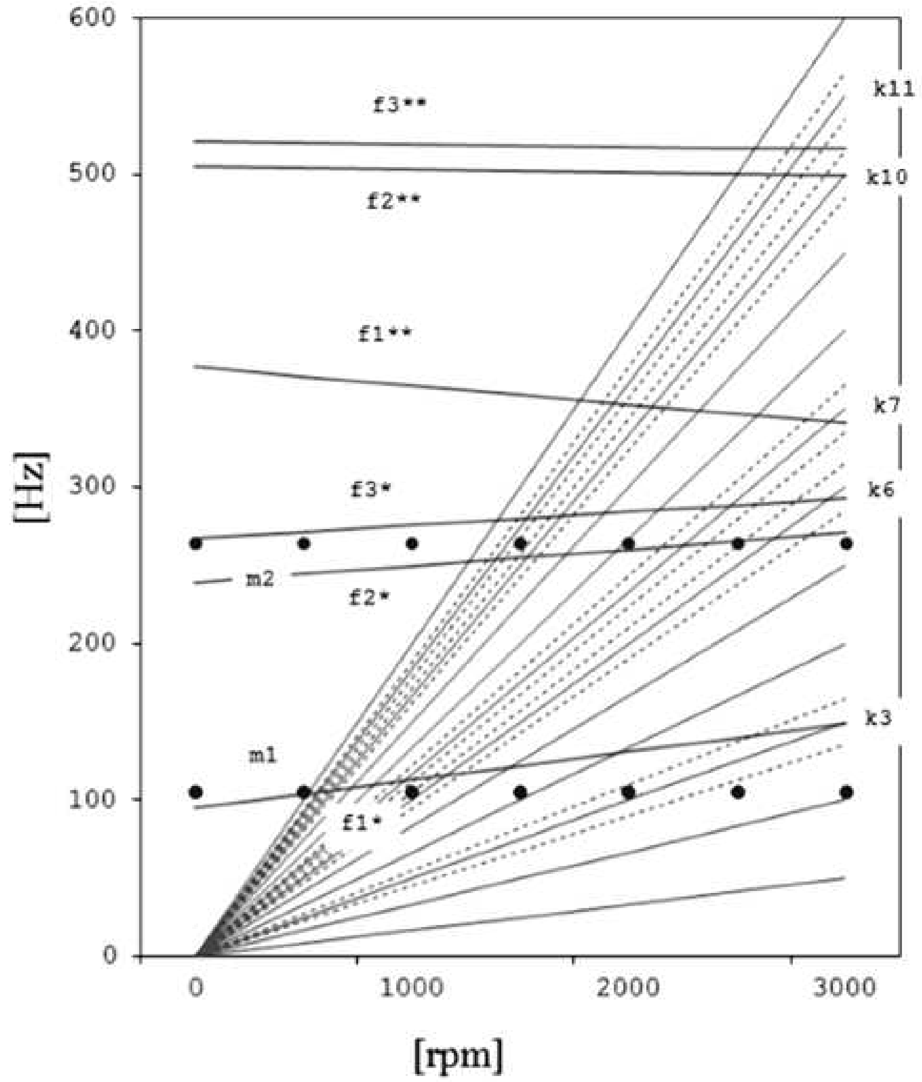

Figure 4. The Campbell vibrational diagram of the Baumann blade: m1 and m2 (full dots in the diagram) denote experimental measurements, f1*, f2*, f3* are results for first fixing type and added rotation 300 rpm (validation with the experiment), f1**, f2**, f3** are second fixing type (added fixing at the hole of wire damping) and added rotation 3000 rpm, k3–11 is a harmonic number.

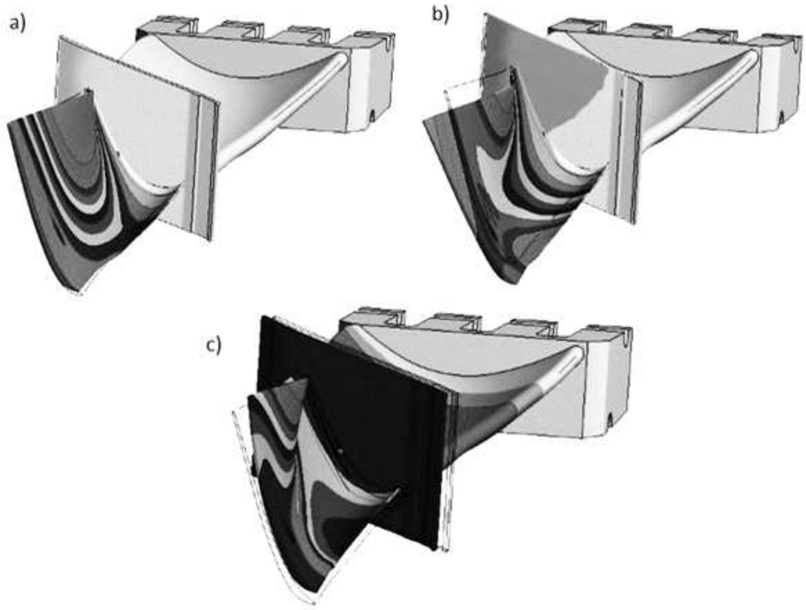

Figure 5. Mode of natural frequencies for the second fixing type.

is working in the turbine stage. During experimental measurement, we can only determine the frequency of the blade corresponding to the fixing found in the Laboratory. For such case, we can verify the numerical model. Next, we can add rotation and the boundary conditions reflecting the damping wire.

4 Mathematical model of the operational degradation

Having a set of different blades that have spent a different time in a real turbine conditions, we can identify numerous sources of damage and the decrease of the life-time. The most important are irreversible (permanent) contributions to the deformation state, coming mainly from inelastic contributions to the deformation tensor.

4.1 Decomposition of the rate of strain

Mathematical model of material degradation, partially developed in papers by Bielecki, Dudda, Kucharski, *et al.* [1–8], is now scrutinised. Our aim is to use it for developing of the advanced BOTT modules. Bearing in mind, that elements of steam turbines undergo only small deformations we shell assume an additive decomposition rate of the strain tensor into elastic and inelastic parts [5,6]

$$d_{ij} = \dot{\varepsilon}_{ij} = \dot{\varepsilon}^{el}_{ij} + \dot{\varepsilon}^{p}_{ij} + \dot{\varepsilon}^{\theta}_{ij} + \dot{\varepsilon}^{c}_{ij} + \dot{\varepsilon}^{ch}_{ij} + \dot{\varepsilon}^{TP}_{ij} + \dot{\varepsilon}^{PT}_{ij} + \dots \quad i,j = x,y,z \ , \tag{1}$$

where:
$\dot{\varepsilon}_{ij}$ – rate of total strain, analogous to the deformation tensor d_{ij};
$\dot{\varepsilon}^{el}_{ij}$ – rate of elastic strain tensor, compatible with the Hooke law, Eq. (‘2);
$\dot{\varepsilon}^{p}_{ij} = \dot{\lambda}\partial\phi/\partial s_{ij}$ – rate of incompressible plastic flow, where $\dot{\lambda}$ is a parameter of plastic flow;
$\dot{\varepsilon}^{\theta}_{ij} = \beta(\dot{T} - \dot{T}_{ref})\delta_{ij}$ – rate of spherical thermal strain tensor;
$\dot{\varepsilon}^{c}_{ij} = \dot{\varepsilon}^{cr} s_{ij}$ – rate of creep strain, compatible with the Norton law for $\dot{\varepsilon}^{cr}$;
$\dot{\varepsilon}^{ch}_{ij} = \dot{\varepsilon}^{ch}(c_\alpha\delta_{ij}, \partial_i c_\alpha \partial_j c_\alpha, s_\alpha)$ – rate of spherical chemical strain tensor induced by a local deformation of chemical reaction products c_α, $\alpha =H_2$,Ca,CaO, etc.;
$\dot{\varepsilon}^{TP,PT}_{ij}$ – rate of plastic strain tensor induced by changes phase and changes phase induced by plastic strain.
These two strains describe degradation of alloy steel undergoing cyclic changes phase. Details of this model are presented in papers by Bielecki *et al.* [1], Kucharski and Badur [2], Dudda and Badur [6].

Adducted summing by single formulas, does not exclude all contributions of individual phenomena, which may be coupled by evolution equation, and not by momentum balance equation. For example the Mroz model cannot be applied for a creep of material having plastic strain, as this type of coupling needs to describe rate of creep strain through a separate evolution equation. Other type of coupling have been developed and described in [2,5,6].

4.2 Definition of elastic deformations

Typical materials used in power plants exhibit a linear isotropic relationship of strain and stress tensor represented by the Hooke law for isotropic materials [11]

$$\sigma_{ij} = D_{ijkl}\varepsilon^{el}_{kl} = 2\mu^{el}_{ij} + \lambda^{el}_{kk}\delta_{ij} \ , \tag{2}$$

where σ_{ij} is the elastic stress tensor, D_{ijkl} is the element of the elastic tensor $(i,j,k,l = 1,2,3,)$, the Lamè coefficients μ, λ are coupled with the Young module, E, and Poisson constant, ν, by the formulae [10]

$$\lambda = \nu E/(1+\nu)(1-2\nu), \quad \mu = E/2(1+\nu) \ . \tag{3}$$

Traditionally, modules E, ν depends on temperature.

4.3 Definition of plastic strain

An essential element of the Prandtl-Reuss plastic-flow theory is the possibility of extending the 0D experimental relationship for the rate of incompressible plastic flow, $\dot{\varepsilon}^p = f(\sigma)$, to the 3D case. This can be described by the plastic flow model [9]

$$\dot{\varepsilon}_{ij}^p = \dot{\varepsilon}^p \partial F/\partial s_{ij} \ , \tag{4}$$

where the stress deviator is defined as: $s_{ij} = \sigma_{ij} - (1/3)\sigma_{kk}\delta_{ij}$, and F describes a domain of elastic response. This theory assumes a direction of the plastic flow, namely $\dot{\varepsilon}_{ij}^p$, should be perpendicular to the surface of plasticity, which can be written by Huber-Mises-Hencky (HMH) formulas:

$$F(\sigma_{ij}, \alpha_{ij}, R_w) = \sqrt{3/2J_2} - R_w - R_e = 0 \ , \tag{5}$$

$$J_2 = 1/2(s_{ij} - \alpha_{ij})(s_{ji} - \alpha_{ji}) \ , \quad i, j, k = x, y, z \ , \tag{6}$$

where J_2 is the second Cauchy invariant based on the difference of the deviator stress and α_{ij} is the kinematic strengthening tensor, R_w is the isotropic strengthening, and R_e is the plastic yield, resulting from one axis, static tensile experiment.

Plasticity writing by the Hubber-Mises-Hencky condition, determines the minimal value of shape deformation energy, evolving by kinematic and isotropic strengthening. Kinematic and isotropic strengthening evolution equation has been written by the Prager-Ziegler formula [11]

$$\dot{\alpha}_{ij} = 2/3R_w^\infty \dot{\varepsilon}_{ij}^p - a\dot{\varepsilon}^p \alpha_{ij}, \quad \dot{R}_w = \dot{\varepsilon}^p b(1 - R_w/(R_w^\infty) \ , \tag{7}$$

where R_w^∞ is a natural limit of hardening, and a, b are dimensional coefficients, $\dot{\alpha}_{ij}, \dot{R}_w$ is the rate of evolution, respectively kinematic and isotropic strengthening.

4.4 Evolution of the damage parameter

4.4.1 Damage in the third creep phase by the Gurson-Bielecki model

Parameter of damage rate $\dot{D}$ occurring in nonlinear strain models, was conceived as a supplement law of material degradation in the locations of maximal stresses. This parameter, adapted to description of the third phase creep from the damage takes the form [1,7]:

$$\dot{D} = A \left[\frac{\sigma_{HMH}(1 + \varepsilon^c)}{R_n(1 - D)} \right]^n \ , \tag{8}$$

where A, n, R_n are constants of the model calibrated by Bielecki for P91 steel [1] and σ_{HMH} is the so-called Huber reduced stresses whereas ε^c is the scalar of creep deformation.

For the Kachanov damage model, the parameter of damage does not participate in additional decomposition of the rate of strains, but affects the rate of strain obtained by the changes in the Young module and the Poisson constant [8].

$$E = E_0(1 - D), \quad \nu = \nu_0(1 - D) \, , \tag{9}$$

where E_0, ν_0 are values for the material at the beginning.

4.4.2 Low-cycle damage by the Dudda model

Other degradation and damage mechanism is found by the Dudda formulae [5,6]. This mechanism has been applied for the low-cyclic damages, induced by thermal gradients and stress corrosion, reattaching and solely local plastic strengthening. As the criterion determining damage in the material was adopted the maximal energy dissipation value, progressive in the every load working cycle. When this energy (we mean energy dissipation) reaches a critical value, construction in this field should be running to degradation. This model has been adapted to numerical CSD simulation writing damage construction by working cycle [5,6]: start up – nominal work – shut down – stopping.

4.4.3 Stress-corrosion damage by the Kucharski model

Universal Kucharski model (i.e., diffusion-chemical model) is described in [2,7,8]. In this model have been adopted modules having the following effects:

- environmental impact through concentration participation of the chemical components,
- environmental impact through pH and electrical potentials,
- environmental impact through corrosion induced by the stop time working turbine,
- impact of the products of chemical reaction which accompany stress corrosion.

Progression of damage parameter in Kucharski model has been written by formula [2,8]:

$$\dot{D} = \frac{\partial}{\partial x_i}\left(J_i^{stress} + J_i^{chem} + J_i^{cycle}\right) + s^c \, , i = 1, 2, 3 \, , \tag{10}$$

where D is the damage parameter, $J_i^{stress} + J_i^{chem} + J_i^{cycle}$ are diffusion fluxes coming from stresses, chemical components of corrosion and a cycle damage contribution, and s^c is the source of corrosion formed by local chemical reactions.

5 Summary and conclusions

In the paper have been presented some results of validation of the numerical model, which has been used by the Advanced Numerical Control System [1–9]) implemented to BOTT, developed for the on-line controls and monitoring of a degree of degradation and damage of critical turbine elements. All experimental measurements have been performed on the setup manufactured at the Laboratory in Faculty of Mechanical and Power Engineering, Wrocław University of Technology.

The comparison of experimental measurement and results of CSD simulations enables validation of a basic model for referential blade, taken off from the Baumann stage turbine of K200-130 type. The differences caused by the blade fixing conditions, as well as for the setup experiment, and fixing conditions in a turbine during real time working have been shown. Although the damage model validation is not a simple task, as for instance a validation by direct comparison of numerical and measurement data, it has been possible to (by using a proposed analogy for numerically fixed blade) simulate appropriate conditions for real working blade in turbine.

Finally, mathematical models that are the base for the ANCS modules have been described, however these modules need an experimental validation.

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Modelowanie pracy turbinowego stopnia Baumanna, część II – drgania swobodne i wymuszone kinetycznie

S t r e s z c z e n i e

Niniejszy artykuł jest kolejnym z cyklu, którego celem jest opracowanie numerycznego narzędzia badawczego opartego na CSD (Computational Solid Dynamics). Zadaniem jego byłaby weryfikacja stanu wytrzymałościowego i dynamicznego trudnodostępnych dla pomiarów urządzeń w turbinie takich jak: rurociągi pary świeżej, zawory, wirniki czy łopatki. Narzędzie obserwowałoby rozwijające się mechanizmy: korozji, pełzania wysokotemperaturowego i niskocyklicznego zmęczenia. Ostateczną walidacje modeli numerycznych z eksperymentem zaplanowano wykonać w Laboratorium Wydziału Mechaniczno-Energetycznego Politechniki Wrocławskiej. W tej części cyklu artykułów pokazano m.in. walidację modelu numerycznego z pomiarami drgań, jakie wykonano na stanowisku badawczym oraz w skrócie opisano poszczególne moduły narzędzia. Po implementacji wszystkich modeli do systemu nadzorującego pracę turbozespołu o nazwie BOTT (Blok Ograniczeń Termicznych), powstanie zupełnie nowy, elastyczny i kompletny system monitorujący stopień degradowania się elementów turbozespołu o nazwie ANCS (Advance Numerical Control Systems) działający w sposób on-line.