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COMPUTER MODELING OF THE STRESS STATE AND RELIABILITY ASSESSMENT OF BOLTED CONNECTIONS OF THE ROTOR AND RUNNER OF KAPLAN TURBINES

This study presents a comprehensive theoretical assessment of the deformed state and stresses in the bolted joints of the runner and shaft of the hydro-turbine. The investigation is motivated by the need to understand the structural behavior of these critical components under varying operational conditions. The primary purpose of this research is to evaluate the theoretical aspects of the bolted joints hydro-turbine, emphasizing the impact of tightening technology on the deformed state and stress distribution. A comprehensive review of literature, drawings, material properties, and acting loads of the bolted connection for turbines is conducted, estimating loads during normal operation and load rejection. A detailed nonlinear finite element model of the bolted joint is built, considering a 3D bolt model and runner fragments. The model incorporates nonlinear effects such as contact interaction and plasticity of the material during deformation.

A methodology for modeling bolted joint tightening technology is developed, allowing estimation of bolt pre-operation stress and deformed states. Error estimation and model validation are performed through a comparative analysis of models with different meshes. Stresses arising at different tightening levels are determined, showcasing the non-linear effect on bolt stresses.

The study reveals that the maximum numerical error in stress estimation does not exceed 5.2%. Stresses at different tightening levels demonstrate a non-linear response, with significant reductions and increases corresponding to tightening deviations. Estimated stresses in the main operating loading cycle provide insights into the amplitude and average stress values during normal operation and load rejection. The study estimates a significant number of cycles before the initiation of a fatigue crack, equivalent to approximately 142 years of operation.

Keywords: Bolted joints, Hydro-turbine, Finite element model, Stress analysis, Tightening technology.

У цьому дослідженні представлено комплексну теоретичну оцінку деформованого стану та напружень у болтових з’єднаннях робочого колеса та валу гідротурбіни. Розслідування мотивовано необхідністю зрозуміти структурну поведінку цих критичних компонентів за різних умов експлуатації. Основною метою цього дослідження є оцінка теоретичних аспектів болтових з’єднань у гідротурбіні, підкреслюючи вплив технології затягування на деформований стан і розподіл напруги. Проведено комплексний огляд літератури, креслень, властивостей матеріалів та діючих навантажень болтового з’єднання для турбін, оцінюючи навантаження під час нормальної роботи та відхилення навантаження. Побудовано детальну нелінійну скінчено-елементну модель болтового з’єднання з урахуванням тривимірної моделі болта та фрагментів робочого колеса. Модель включає нелінійні ефекти, такі як контактна взаємодія та пластичність матеріалу під час деформації.

Розроблено методику моделювання технології затягування болтового з’єднання, що дозволяє оцінити передексплуатаційні напруження та деформований стан болта. Оцінка похибок і перевірка моделі виконуються шляхом порівняльного аналізу моделей з різними сітками. Визначаються напруження, що виникають при різних рівнях затягування, демонструючи нелінійний вплив на напруги болтів.

Дослідження показує, що максимальна числова похибка в оцінці напруги не перевищує 5,2%. Напруги на різних рівнях затягування демонструють нелінійну реакцію зі значними зменшеннями та збільшеннями, що відповідають відхиленням затягування. Розрахункові напруги в основному робочому циклі навантаження дають уявлення про амплітуду та середні значення напружень під час нормальної роботи та скидання навантаження. Дослідження оцінює значну кількість циклів до появи втомної тріщини, що еквівалентно приблизно 142 рокам експлуатації.

Ключові слова: Болтові з'єднання, Гідротурбіна, Модель скінченних елементів, Аналіз напружень, Технологія затягування.

Introduction. Bolted connections are widely used in engineering and are one of the most common ways of collapsible connection of machine parts. A wide range of applications of bolted joints raises the problem of rational choice of their sizes and designs. The bibliographic paper [1] provides references to more than 700 papers (published in the period from 1990 to 2002), which are devoted to the problems of calculation, design, and use of threaded connections. Another review paper [2] summarizes five optimization techniques available in the literature, including bolt layout, tightening strategies, tightening sequences, bolt size, and stresses. In [3] is reviewed state-of-the-art research on bolt tightening force measurement and loosening detection, including fundamental theories, algorithms, experimental set-ups, and practical applications.

Along with static and dynamic strength, fatigue strength is also an important characteristic of the elements of power machines, and its support is an important practical task. In the last 20 years alone, many books have been published describing the history of the occurrence, detection, and study of the causes of fatigue failure of various machines and mechanisms [4-7].

A method for fatigue design by comparing the load on the bolt in joints with the fatigue strength of the bolt-nut joints is proposed in [8]. Another paper [9] describes the failure modes of bolted joints in composite materials, the influence of the bolt clamping torque, the clearance between the bolt and the hole, and aging on the performance of the joint. In [10] discusses improving tightening reliability on bolted joints for the calibrated wrench method. And [11] shows the design and reliability influences on self-loosening of multi-bolted joints.

Among the considered cases of failures that occupy a significant place is fatigue failure of structural elements, including the problem of fatigue strength of the rotor and the flowing part of the Kaplan’s and other types of turbines. Depending on the design features of the rotor and the flowing part of one or another of its elements is more prone to fatigue failure.

A large number of works on the fatigue strength of various components of the turbine indicates the use in the design of insufficiently accurate models for predicting the reliability of such elements, apparently due to the considerable complexity of processes in the turbine for a very long life. In rotary blades, turbines are most prone to fatigue elements of the rotary mechanism of the blade [12-13], the blade itself [14], fragments of the drive [15], and the rotor as a whole [16]. A significant contribution to the development of methods for calculating the dynamic characteristics of turbine blades and fatigue strength was made by the researchers of the Institute of Mechanical Engineering named after A.M. Pidgorny National Academy of Sciences of Ukraine. In a series of works mathematical models have been developed to study the strength, dynamics, and life of the blades. Based on theoretical research numerical values of natural frequencies of a blade in the air and water are received. Another series of works is devoted to the construction of nonlinear mathematical models describing the oscillations of the blades in a liquid. A significant contribution to the development of mathematical modeling of the dynamics, strength, and crack resistance of turbine rotors has been made by M. Shulzhenko.

A separate problem is the fatigue strength of bolted joints because a failure of the bolted connection can lead to the failure of the entire structure. This is evidenced by the increase in unplanned downtime of turbomachines due to breakdowns of bolted joints that occur during the operation of hydropower plants and pumped storage stations in the CIS (post-soviet) countries. Among them, it is necessary to note breaks: blades of runner, runner, fastenings of thrust bearings, hairpins of fastening of covers of hydroturbines, etc. All these unplanned stops, as well as the accident at the Sayano-Shushenskaya HPP, raised the problem of the need to clarify the calculation methods used in the design of bolted joints.

The operation of bolted connections of hydroturbines is connected with many features: the existence of the corrosive environment (water); significant tightening; significant stresses due to torque and dynamic hydraulic force on the turbine shaft. Failures of bolted joints are often gradual and caused by corrosion and fatigue processes. Another feature of these bolts is their size, as Fig. 1 (photo in the laboratory of the Department of Dynamics and Strength of Machines NTU "KhPI") shows the bolt M110x4 after 30 years of operation at the Dnieper-II HPP.

Fig. 1 – Bolts for mounting the runner of the turbine (M110x4) after 30 years of operation

Motivation and Problem Statement. From the analysis of literary and other sources of information, the following conclusions can be made:

Table 1 – Chemical composition and mechanical properties of steels

Component	Steel 40X	Steel 25X1MF	Steel 15X12VNMF
Vanadium(V)		0.15−0.30	0.15−0.30
Titanium (T)			not more 0.20
Silicon (Si)	0.17−0.37	0.17−0.37	not more 0.40
Manganese (Mn)	0.50−0.80	0.40−0.70	0.50-0.90
Copper (Cu)	no more 0.30	not more 0.020	not more 0.020
Molybdenum (Mo)		0.25−0.35	0.50−0.70
Nickel (Ni)	not more 0.30	not more 0.30	0.40-0.80
Wolframium (W)			0.70-1.10
Sulfur (S)	not more 0.035	not more 0.025	not more 0.025
Carbon (C)	0.36−0.44	0.22−0.29	0.12−0.18
Phosphorous (P)	not more 0.035	not more 0.030	not more 0.030
Chromium (Cr)	0.80−1.10	1.50−1.80	11.0−13.0
Yield stress, MPa	490	735	700
Ultimate stress, MPa	522	880	830

– Bolted connections of the shaft and runner of turbines operate under conditions similar to those considered in the literature;

– The service life of a bolted connection in the Argentina and Uruguay HPP turbines is ~43 years, which corresponds to the known periods for failures of bolted connections at the HPP of Ukraine. During this period, fatigue and corrosion processes in the material can develop;

– The technology of tightening such bolted connections is common for Ukrainian, Argentina and Uruguay HPP;

– The materials from which the bolts are made in Ukraine (40X, 25X1MF) differ from the material of the Argentina and Uruguay HPP runner bolts (15X12VNMF), but they have similar mechanical characteristics - high yield strength and tensile strength (Table 1).

– The main loads acting on the bolted connection are tightening, as well as axial force and moment on the shaft.

Based on the analysis of the problem, runners' bolted connections operate under similar conditions, and the same problems are expected. To assess the possibility of potential problems for bolted connections, the problem statement can be formulated:

1. To study information about the design, material properties, and loads of the bolted connection of the shaft and the runner;

2. To build a detailed 3D geometric and finite element model of a bolted connection, consisting of a bolt and fragments of the shaft and runner. The model has to take into account the method of tightening and the contact interaction between the bolt, shaft, and runner body; Determine the loads acting on the structure in the case of normal operation and emergency shutdown;

3. To determine the stress state that occurs in a bolted joint at different tightening levels, determine the effect of tightening deviation within ± 10% of the nominal value on the stress state of the bolt;

4. To determine the stresses in the bolt during normal operation and load rejection of the turbines, based on the calculations, determine the amplitude and average stresses of the loading cycle;

5. To evaluate the number of cycles before the occurrence of cracks from high-cycle fatigue of the bolted joint;

Determination of Loads Acting on the Bolted Connection. When calculating the stress state of a bolted connection, one of the mandatory steps is to determine the operating loads. In general, to determine the axial force and torque on the shaft of the hydropower unit during start/stop, it is necessary to solve the problem of non-stationary hydromechanics, which is a complex scientific problem and requires considerable effort and time. On the other hand, it is possible to conduct a field experiment to determine the load parameters, but such studies are complex and require significant material costs for equipment. Based on this, it has been used field tests of similar hydropower units of Shardarin HPP (PL 661-VB-500 Turbine Units, Syr Darya River, Kazakhstan, 25 MW (31,5 MW after renovation), head – 15,8 m) and Irkutsk HPP (PL-577-WB-720, Angara River, head – 26 m, power – 107,5 MW), in the process of which many parameters has been measured, including axial force and torque on the turbine shaft. Since the hydropower unit considered in this work is also a rotary-blade type, according to the theory of similarity between the dependences of the axial force and torque on time on the turbine shaft, the relations are:

$$M^{AU}(t) = \frac{H^{AU}}{H^{S}}\left(\frac{D^{AU}}{D^{S}}\right)^{3} M^{S}(t), \tag{1}$$

$$P_{ax}^{AU}(t) = \frac{H^{AU}}{H^{S}}\left(\frac{D^{AU}}{D^{S}}\right)^{2} P_{ax}^{S}(t) \tag{2}$$

where $M^S(t)$ is the torque in the hydropower unit of Shardarin HPP;

$M^{AU}(t)$ is the torque in the hydropower unit of Argentina and Uruguay HPP;

$P_{ax}^{S}(t)$ is the axial force in the hydropower unit of Shardarin or Irkutsk HPP;

$P_{ax}^{AU}(t)$ is the axial force in the hydropower unit of Argentina and Uruguay HPP;

H^S is the pressure at Shardarin or Irkutsk HPP;

H^{AU} is the pressure at Argentina and Uruguay HPP;

D^S is the diameter of the runner at Shardarin or Irkutsk HPP;

D^{AU} is the diameter of the runner at Argentina and Uruguay HPP;

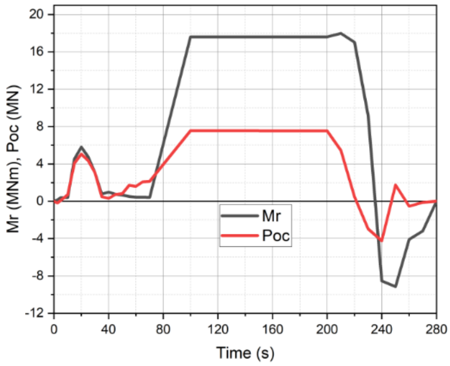


Fig. 2 – Axial force (Poc) and torque (Mr): Load rejection

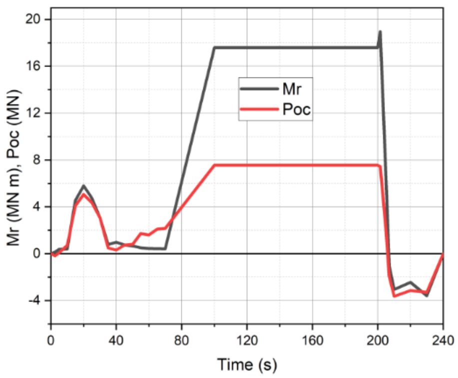


Fig. 3 – Axial force (Poc) and torque (Mr): Normal stop

The results of field measurements have been digitized. In the nominal operation mode, the torque values is considered constant, and the time to enter the stationary mode is 30 seconds. Thus, the digitized data using (1)–(2) have been converted into axial force and torque occurring in the PL40-VB850 hydro turbine. Based on these data, a load cyclogram has been completed. Graphs of the dependences of the axial force and torque on the shaft of the PL40-VB850 hydro turbine for one cycle are shown in Fig. 2 and Fig. 3. The difference in loading between stop at normal operation and load rejection can be seen comparing Fig 2 and Fig 3. As can be seen, the main difference is in higher values of axial force and torque in the reverse direction.

Also, to increase the correctness of the axial force and torque, these values on normal operation mode have been scaled to the theoretical values, Mr = 17,6 MN×m, Poc = 7,55 MN.

Stress state, which occurs due to tightening only. To assess tightening variation on the stress state of bolted connections the finite element model has been developed (Fig. 4). This model takes into account contact interaction between shaft flange, bolt, and runner.

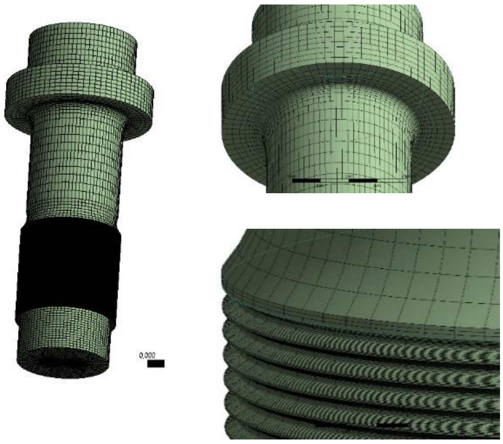


Fig 4 – Finite element mesh

According to the regulations document, the elongation during tightening the bolts is δ = 0.30 mm ± 0.03 mm. Thus, it is important to study how changes in elongation during tension in the bolt. For this purpose, additional calculations have been performed for the two extreme cases $\delta_{-10\%}$ = 0,27 mm and for $\delta_{+10\%}$ = 0,33 mm. The boundary condition and dependence between δ and Δ are shown in Fig. 4. Empirically the dependence of initial tension (Δ) on elongation during tightening (δ) from bolt M140×4 has been estimated as formula (3) and Table 2.

$$\delta = 0{,}617\Delta \qquad (3)$$

Table 2 – Correspondence between initial tension and elongation during tightening

Δ, mm	0,4374	0,4860	0,5346
δ, mm	0,2698	**0,2998**	0,3298

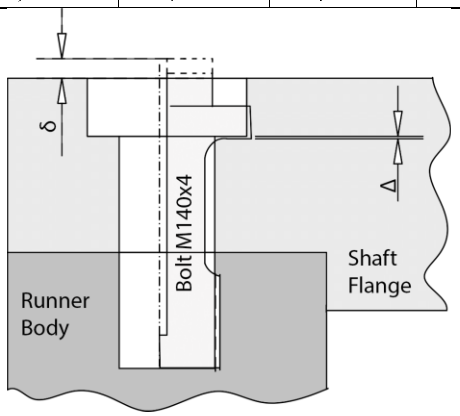


Figure 5 – Boundary conditions for initial tension (Δ) and elongation during tightening (δ)

Table 2 – Maximum stress dependence on tightening variation

Δ, mm	0,4374	0,4860	0,5346
δ, mm	0,2698	0,2998	0,3298
σ_{allmax}, MPa	647	641	681
	+0,93%	0%	+6,24%
σ_{same_node}, MPa	545	641	668
	-14,9%	0%	+7,33%

Calculating Stresses and Strains of the Bolted Connection which occurred during start/stops. *Normal start/stop*. The greatest interest in the distribution of stresses is the time when the highest loads are applied. We can choose such time points as t = 201 s for normal start/stop. Fig. 7 shows the distribution of von Mises equivalent stresses (Fig. 7a) on fillet (Fig. 7b) and threads (Fig. 7c) of the bolts (orientation points 0°, 90°, 180°, 270° are shown in Fig. 6). All stress values during the start/stop cycle are presented in graphs Fig. 8 and Fig 9.

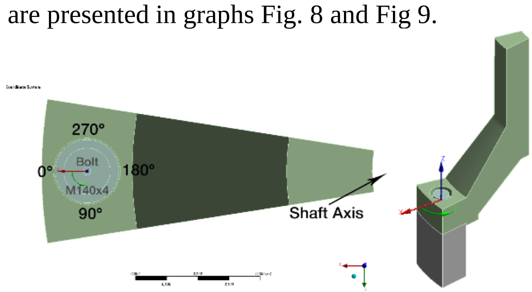
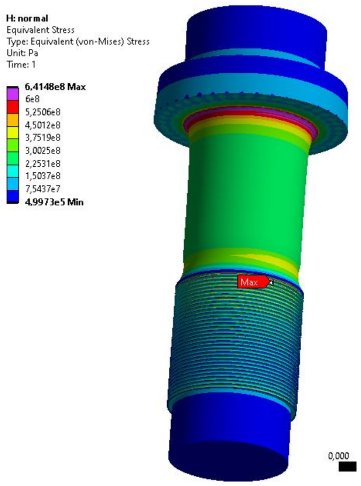
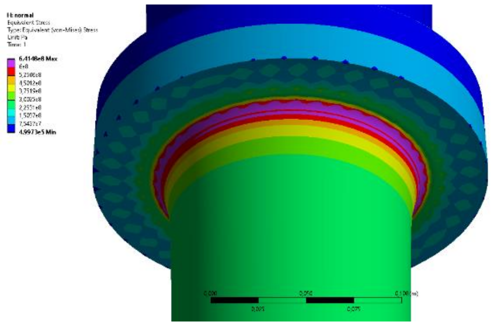


Fig. 6. Sketch of the orientation 0°, 90°, 180°, 270° points

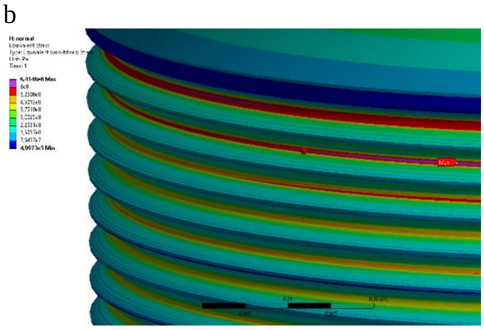
Extreme start/stop. The same calculation is performed for the case of load rejection. In the case of an extreme stop (load rejection) of the HPU, the maximum load coincides with the case of a normal stop, but this mode of stop is characterized by significant loads in the opposite direction. Therefore, the points with the largest negative values of axial force and torque at t = 201 s are selected for visualization of stresses. Stress dependences on time are shown in Fig. 11 and Fid 12.



a



b



c

Fig. 7 – von Mises Stress distribution (Pa) in bolt: a – general view; b – on bolt fillet; c – on the thread

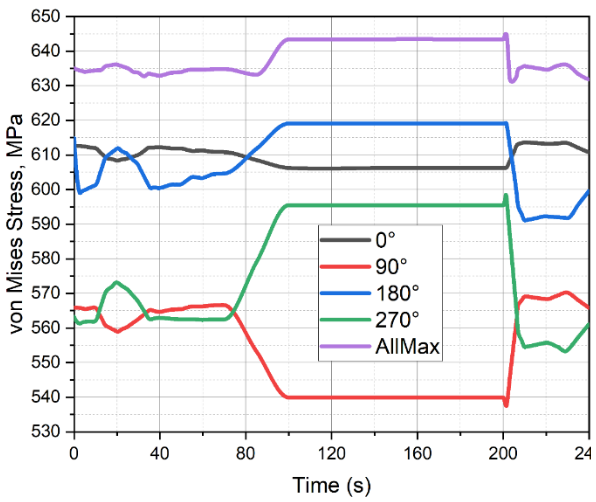


Fig. 8 – von Mises stress on the fillet during normal start/stop

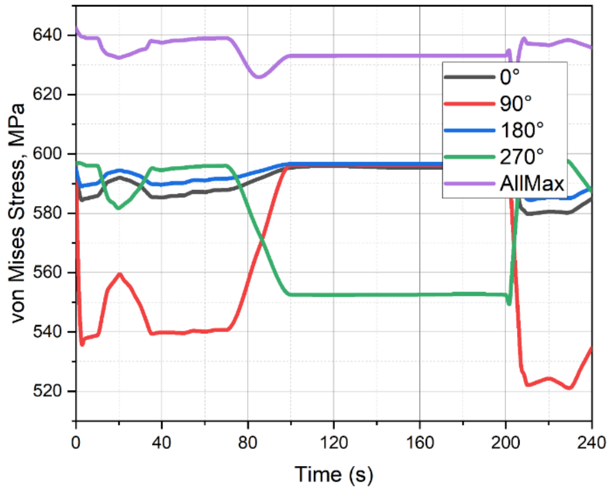


Fig. 9 – von Mises stress on the thread during normal start/stop

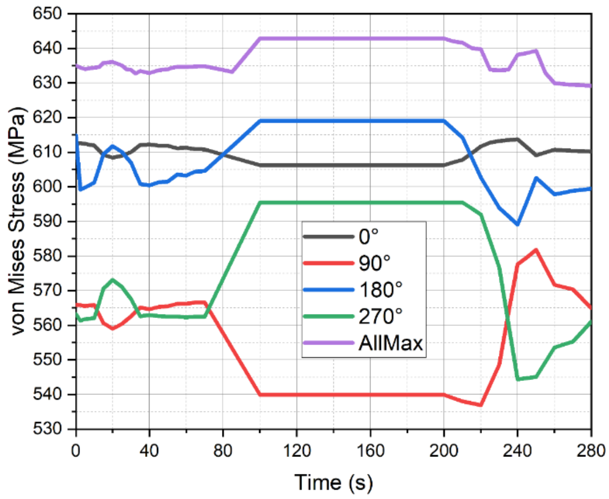


Fig. 10 – von Mises stress on the fillet during extreme start/stop

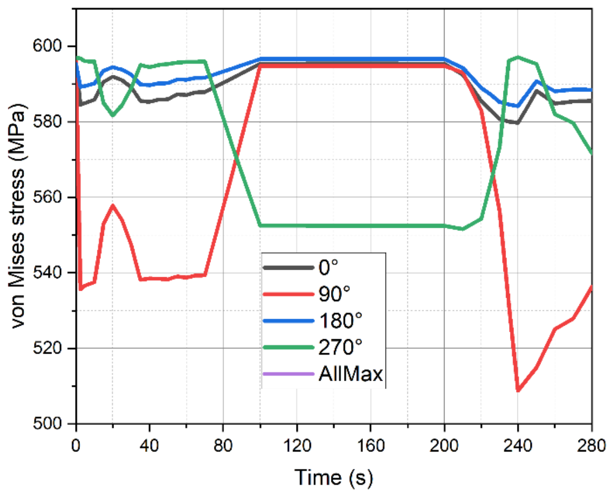


Fig. 11 – von Mises stress on the thread during extreme start/stop

Life-time assessment. To estimate the number of cycles before the occurrence of a fatigue crack, it is necessary to introduce the fatigue curve equation. The constructed diagram and the equation of the fatigue curve are defined for the case of symmetrical loading (R = –1). Similarly, the equations of the fatigue curve can be constructed for asymmetric load schemes

$$\sigma_a^m \cdot N = \sigma_R^m \cdot N_0 \tag{4}$$

where m, N_0 – material parameters;
N – number of cycles;
σ_a – stress amplitude;
σ_a – stress endurance limit;

The dependence of the endurance limit on average stresses can be expressed (approximated) by a phenomenological equation. The most common are the Goodman and Gerber equations.

Gerber parabola (more accurately describes the behavior of plastic materials):

$$\sigma_R = \sigma_{-1} \cdot \left[1 - \left(\frac{\sigma_m}{\sigma_u}\right)^2\right] \tag{5}$$

The results of research to determine the limits of endurance under different asymmetric load modes are presented as a limit diagram of stresses and stress amplitudes of the cycle. The cycle limit amplitude diagram characterizes the relationship between the values of the limit amplitudes and the values of the average cycle stresses for a given durability. Any point on the curve corresponds to a certain coefficient of asymmetry of the cycle.

The endurance limit (σ_{-1}) of the analyzed bolted connection during operation is influenced by the following factors: scale factor (ε = 0.55), surface condition (β = 0.8), corrosive environment, freshwater (γ = 0.46), fretting corrosion in the thread (α = 0.35). Values of the average stresses of the cycle (σ_m) and the amplitude of the stresses of the cycle (σ_a) have been shown in Table 3. The result of the evaluation is shown in Table 4.

$$\sigma_{-1\mathrm{p}} = \alpha\beta\gamma\varepsilon\sigma_{-1} \tag{6}$$

$$N_f = \begin{cases} N_0 \left(\dfrac{\sigma_{-1\mathrm{p}}}{\sigma_{a\mathrm{eqv}}}\right)^m, & \sigma_{a\mathrm{eqv}} \geq \sigma_{-1\mathrm{p}} \\ \infty, & \sigma_{a\mathrm{eqv}} < \sigma_{-1\mathrm{p}} \end{cases} \tag{7}$$

Table 3 – Material properties for life-time assessment

σ_y, MPa	σ_u, MPa	σ_{-1}, MPa	σ_{-1p}, MPa	N_0	m
600	750	380	17,9	10^7	4

Table 4 – Life-time assessment

σ_a, MPa	σ_m, MPa	σ_{a_eqv}, MPa	$N_f \times 10^4$, cycles
38,1	559,3	85,83	1,91

Conclusions. The following conclusions can be drawn as a result of the study on the theoretical assessment of the deformed state of the bolt joints of the runner and the shaft hydro-turbine:

1. A review of the literature, drawings, material properties, and acting loads of a bolted connection has been carried out. The loads acting on the bolted connection for Argentina and Uruguay turbines during normal operation and load rejection are estimated.

2. A detailed nonlinear finite element model of a bolted joint has been built. It takes into account a 3D bolt model and fragments of the runner. The finite element model consists of 501k nodes and 443k volumetric 8-node hexahedral elements with 3 dof-s per node. The model takes into account the nonlinear effects: contact interaction in the joint and plasticity of the material rising at deformation.

3. A methodology of the modeling of bolted joint tightening technology is developed and realized for the estimation of the bolt pre-operation stress and deformed states.

4. An error estimation analysis has been carried out. A comparative analysis of the models with different meshes (basic and densed) has been done for model and results validation. The improved (densed) FE model has 770k nodes and 686k elements (double mesh refinement). According to the obtained comparative results, the difference in stress estimation does not exceed 0.8%, on strain estimation 5.2% (a significant difference in errors is explained by plasticity). Thus, the maximum numerical error does not exceed 5.2%.

4. The stresses arising at different tightening levels δ = 0.30 mm ± 10% have been determined. The tightening value has a non-linear effect on the stresses in the bolt. 10% decrease in tightening from nominal value (δ = 0.28 mm) leads to a stresses reduction by 14.9%. 10% increase in tightening value from the nominal caused the increase of the stress values by 7.33%. The behavior is explained by nonlinearities of contact interaction and plastic deformation of the material. However, it should be underlined that plastic deformations have only a single occurrence and they are formed as a result of tightening technology. Thus, plasticity has no cyclic behavior and there is no cause for low-cycle fatigue in the bolted joint.

5. Estimated stresses in the main operating loading cycle have been carried out. The cycle forms from the normal operation condition and load rejection that occurs several times per day. It has been found that during normal operation the amplitude of the cycle on the fillet is σ_a = 22,2 MPa, and the average stress of the cycle is σ_m = 575,8 MPa. In the thread σ_a = 38,1 MPa, σ_m = 559,3 MPa. When the load rejection occurs the stress on the fillets σ_a = 26,3 MPa, σ_m = 570,7 MPa, in the thread σ_a = 44,3 MPa, σ_m = 553,1 MPa.

7. The number of cycles before the occurrence of a fatigue crack has been found. According to the obtained estimation, the number of cycles before the fatigue crack initiation is $1{,}91 \times 10^4$, that approximately equivalent to 142 years of operation.

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