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UDC 539.3

USE OF COMPUTER TECHNOLOGIES IN MODERNIZATION OF HEAD COVERS FOR ПЛ 20-В-500 KAPLAN TURBINES

Описані методики, розроблені для дослідження динамічного напружено-деформованого стану кришки гідротурбіни, застосування яких обґрунтовано нормативним документом «Розрахунок залишкового ресурсу елементів проточної части гідротурбін ГЕС та ГАЕС. Методичні вказівки» СОУ-Н МЕВ 40.1 -21677681-51:2011. Вперше в тривимірній постановці враховано вплив приєднаних мас води конструкції із застосуванням математичних моделей, що ґрунтуються на гіперсингулярних рівняннях і поєднанні методів скінченних та граничних елементів. Отримано чисельні результати, що дозволяють оцінити з урахуванням впливу води динамічний напружено-деформований стан литої чавунної кришки гідротурбіни ПЛ 20 В-500, а також розробленої для її заміни конструкції сталевої зварної кришки. Виконано аналіз чисельного дослідження та надано рекомендації для проектування зварної кришки, динамічні характеристики якої дозволяють виключити резонансні явища та забезпечити експлуатаційну надійність.

Introduction

In recent years, the level of requirements for the efficiency and reliability of energy equipment has increased sharply, and a considerable use of energy potential in many countries of the world, including Ukraine, has resulted in the necessity to modernize and replace the HPS turbine equipment which has been in operation for a long time. Evaluation of the efficiency and scope of the reconstruction requires computer technologies using specialized software to study the strength and dynamics of the parts and components of the turbines.

When deciding on the scope of modernization, in particular, one considers the necessity to replace or extend the service life of the head covers of turbines, which are one of its most metal consuming units. In the previous designs of the turbines, the head covers were usually made in the form of cast iron castings, whereas at present, they are welded from carbon steel sheets. It should be emphasized that the elastic properties of gray cast iron used for the castings earlier depend on the amounts of graphite inclusions: a modulus of elasticity can amount to 40 − 75 % of the elastic modulus of steel, about 67 % of Poisson's ratio, and the density of cast iron − to 90 − 95 % of the density of steel. If in the process of modernization of the turbine a decision has been made to replace the head cover, it is of interest to carry out a comparative numerical analysis of the stress-strain state of the head cover used and the head cover being designed.

The main requirements for the design of the head cover are to provide not only strength but also rigidity, as well as vibration reliability since the head cover vibrations in both axial and radial directions must meet the existing standards. A special feature of the problem is the necessity to fit the new cover into the existing flow section.

A regulatory document has been developed to assess the service life of the elements of the flow section, including the head covers for the said turbines [1]. The reliability of the results obtained by the developed procedure is confirmed in [2]. This approach was developed in [3 − 4] to determine the stress-strain state of a constructive and orthotropic body under asymmetric stress, which makes it possible to reduce the computations of the required displacements to the solution of independent problems for each term in the Fourier series expansion. One of the important tasks solved both in forecasting the service life of the head covers, and in the case of replacing cast iron head covers with steel ones, is an accurate determination of their eigenfrequencies, taking into account the effect of a liquid. This paper describes a technique in which, in contrast to [5 − 7], the modes of vibration of the head cover in a liquid are represented as the eigenmode expansion of its vibrations in a vacuum. The developed technique for constructing matrices of the additional masses of the load-bearing structures interacting with a liquid is described in [8 − 9] and is given below. This approach makes it possible to reduce the computations of the required displacements to the solution of independent problems for each term in the Fourier series expansion.

Free hydroelastic vibrations of turbine head covers

We write the system of equations of motion of a deformable construction symbolically as

$$L(\mathbf{U}) + M(\mathbf{U}) = \mathbf{P}\,, \tag{1}$$

where L and M are the operators of elastic and mass forces; **P** is the pressure of the liquid on the structural element in question (blade); $\mathbf{U} = (u_1, u_2, w)$ is a vector-function of displacements. The speed of the oncoming stream is assumed to be zero. The liquid motion is studied in a 3D formulation by the methods in potential theory. It is assumed that the liquid is ideal, free vortices are not formed and they do not descend from the lifting surface. In this case, there exists a velocity potential that satisfies the harmonic equation everywhere outside the plate, and on the face surfaces of the plate $S^{\pm}$ − the no-flow condition. For a potential flow, the perturbed velocity of the liquid is represented as

$$\mathbf{v}(x, y, z, t) = \mathrm{grad}\boldsymbol{\Phi}(x, y, z, t)\,, \tag{2}$$

where $\boldsymbol{\Phi}$ (x, y, z, t) is the potential of velocities induced by small free vibrations of the plate. To determine the pressure of a liquid on wetted surfaces, the Cauchy-Lagrange integral is used. To find the pressure on the deformable surface from the liquid side, it is necessary to determine function $\boldsymbol{\Phi}$ (x, y, z, t) by solving the Laplace equation with the following boundary condition:

$$\left(\mathrm{grad}\boldsymbol{\Phi}\cdot\mathbf{n}\right)\Big|_S = \frac{\partial \overline{w}}{\partial t}\,, \tag{3}$$

Thus, it is required to determine functions **U**, $\boldsymbol{\Phi}$ (x, y, z, t), satisfying the system of differential equations (1) − (2), the no-flow conditions (3), the conditions for fixing and damping the perturbed velocity of the liquid at infinity. The literature has no numerical studies to determine the eigenfrequencies and vibration modes of such structural elements in a liquid. Earlier, to estimate the influence of a liquid on the frequency of eigen-vibrations, the results obtained using the approximate Rayleigh-Lamb approach were used. In this case, shapes of the radial plate were taken as forms of free vibrations for the turbine head cover, while 2D models were used for the turbine blade array.

In this paper, a method for calculating the frequencies and forms of free vibrations of the structures interacting with a liquid is proposed, based on the attraction of the apparatus of singular and hypersingular integral equations.

To solve this problem, we apply the method of given forms [10]. At the first stage, in a 3D formulation, a calculation of the frequencies and modes of vibration of the structure in a vacuum is carried out with the help of the finite element method and its modification for the body of revolution. The obtained modes of free vibrations are chosen as the basic system of functions, by which they are decomposed into a series of the vibration modes of the structure in question in a liquid.

Let us study the case of harmonic vibrations. Then the problem under consideration reduces to

$$L(\mathbf{u}) + M(\mathbf{u}) = \left(0, 0, i\Omega\rho_2\left(\varphi^- - \varphi^+\right)\right), \quad \nabla^2\varphi = 0\,; \quad \left.\frac{\partial\varphi}{\partial\mathbf{n}}\right|_{S^{\pm}} = i\Omega w\,. \tag{4}$$

We represent the function φ (x, y, z) as a potential of a double layer with an unknown density. Then the problem of determining the pressure (4) reduces to solving the integral equation

$$\frac{1}{4\pi}\iint_S \Gamma(\xi)\frac{\partial^2}{\partial\mathbf{n}_x\partial\mathbf{n}_\xi}\left[\frac{1}{|\mathbf{x}-\xi|}\right]dS_\xi = i\Omega w\,. \tag{5}$$

Suppose that the eigenmodes of vibrations in a liquid are representable as

$$w = \sum_{k=1}^{N} c_k w_k\ .$$

Let the functions $\Gamma_k(\xi)$ be solutions of the hypersingular equation (5) with appropriately chosen right-hand sides:

$$w = w_k .$$

To solve the hypersingular equation (5), the discrete singularity method was applied [5, 6]. In this case, the integration domain was divided into a finite number of quadrangular subdomains N_S, in each of which an unknown density was replaced with a constant value. When calculating the finite part according to Adamar for the integrals in (5) across the quadrangle arbitrarily oriented in space, the formula obtained in [5] was used. The elements of the matrix of the additional masses were found by the formula

$$P_{ik} = \iint_S \Gamma_i(x) w_k(x) dS ,$$

where $\Gamma_i(x)$ are the amplitude values of the pressure induced by its eigenform $w_k(x)$. After determining the elements of the matrix of the additional masses, the eigenvalue problem can be solved according to the method developed in [5, 10].

Investigation of the vibration frequency spectra of the head covers for the ПЛ 20-В-500 turbines, taking into account the influence of a liquid

The design of the head covers for the ПЛ 20-В-500 turbines in service consists of bodies of revolution and a system of multiply-connected meridional plates (Fig. 1).

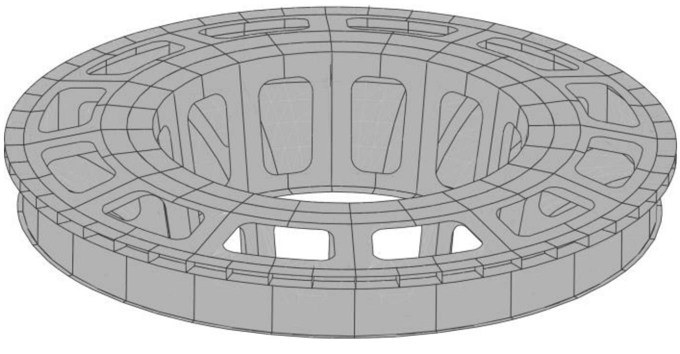

Fig. 1. Head cover of the ПЛ 20-В-500 turbine

For calculations, the mechanical properties of materials were used in accordance with the data given in [11−13].

For grey cast iron Сч20 it was assumed that the modulus of elasticity $E = (0.8 \div 1.2) \times 10^5$ MPa, the Poisson ratio $\nu = 0.21 \div 0.25$, the tensile strength $\sigma_B = 210$ MPa, the material density $\rho = 7{,}100$ kg/m^3.

For steel Ст3 the modulus of elasticity $E = 2.1 \times 10^5$ MPa, the Poisson ratio $\nu = 0.3$, the tensile strength $\sigma_B = 380$ MPa, the material density $\rho = 7{,}800$ kg/m^3 were considered.

The calculation of the eigenfrequencies of the head cover vibrations was carried out for two variants of fixation, imitating, depending on the tightening force of the fasteners, a possible contact of the head cover flange surface with the stator surface: resting along the line of its fastening to the stator with studs ($u_r = 0$, $u_z = 0$, $u_\varphi = 0$) and a rigid fastening of the flange cover to the stator ($u_r = 0$, $u_z = 0$, $u_\varphi = 0$).

The eigenfrequencies and vibration modes were calculated, taking into account the masses of the turbine parts and units placed on the head cover (added masses of the parts): the regulating ring ($G_{reg\,r} = 5{,}365$ kg), half of the stator shackles ($G_{sh} = 366$ kg), half of the pins of the of the stator ($G_{pin} = 96$ kg), the rod ($G_{rod} = 495$ kg), guide bearing ($G_{gd\,brg} = 114{,}690$ N), the turbine shaft ($G_{turb\,sft} = 23{,}220$ kg), the generator rotor ($G_{gen\,rtr} = 137{,}650$ *кг*), the shaft extension ($G_{sft\,ext} = 900$ kg), the exciter rotor ($G_{exc\,rtr} = 5.160$ kg), the thrust block ($G_{thr\,bl} = 9{,}500$ kg), the thrust ($G_{thr} = 2{,}800$ kg), the cowl cone ($G_{cc} = 11{,}469$ kg).

The design diagram of the ПЛ 20 В-500 turbine head cover is shown in Fig. 2. The values of the masses of the parts and units located on the turbine head cover (Fig. 2) are as follows:

$$G_2 = G_{reg\,r} + ½\, G_{sh} + ½\, G_{pin} + G_{rod} = 6{,}322 \text{ kg};$$

$$G_3 = G_{gen\,rtr} + G_{turb\,sft} + G_{sft\,ext} + G_{exc\,rtr} + G_{thr\,bl} + G_{thr} = 247{,}330 \text{ kg};$$

$$G_4 = G_{cc} = 1{,}469 \text{ kg}.$$

The additional masses of the parts G_i ($i = 2, 3, 4$) are uniformly distributed over the annular portions of the head cover as shown in Fig. 2.

The influence of mass forces is taken into account by adjusting the density of the head cover sections along the boundary of their application [14]. The material densities for the primary discretization zones of the head cover 1, 2, 3, 4 and the body of revolution -1, -2, -3, -4 (Fig. 2) are given in Table 1.

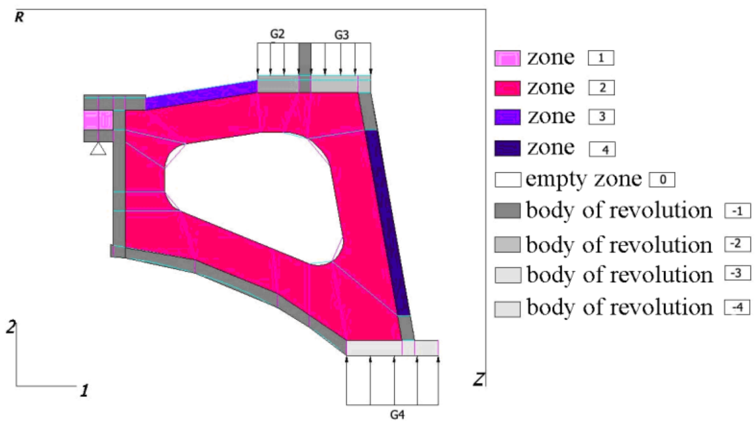

Fig. 2. Peculiarities of the design scheme and the loading of the ПЛ 20 В-500 turbine head cover

The eigen and forced vibrations of both the head cover in service and the new steel cover made of materials with different elastic characteristics, namely of steel Ст3 ($E = 2.1\times10^5$ MPa, $\nu = 0.3$), cast iron '1' ($E = 0.8\times10^5$ MPa, $\nu = 0.3$), cast iron '2' ($E = 1.2\times10^5$ MPa, ν=0.21), were calculated both in a vacuum and in liquid.

The influence of the additional liquid masses on the eigenfrequencies of the head cover was investigated both in a vacuum and liquid, with the head cover resting on the stator along the line of the head cover fastening. The results of the calculations are given in Tables 2 − 4.

Table 1. Densities of materials of primary discretization zones of head cover

Zone sign	G_i, kg	R_2, m	R_1, m	F_i, m^2	h_i, m	$\rho_i = \dfrac{G_i}{F_i \cdot h_i} + \rho_{-1}$
						ρ_i, kg/m^3
1, 2, 3, 4	–	–	–	–	–	steel 7,800 / cast iron 7,100
-1	–	–	–	–	–	steel 7,800 / cast iron 7,100
-2	G_2=6,322	1.925	1.760	1.91017	7.0	5,513 / 5,438
-3	G_3=24,7330	1.710	1.470	2.39766	7.0	14,814.9 / 14,807.4
-4	G_4=11,469	1.568	1.200	3.20010	6.0	6,753 / 6,683

Table 2. Eigen vibration frequencies of steel cover (Ст3), taking into account additional masses of parts, resting

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	29.341	94.941	179.621
	liquid	29.102	94.925	173.978
1	vacuum	23.169	49.471	101.569
	liquid	23.068	49.405	101.545
2	vacuum	23.142	68.512	134.764
	liquid	23.032	68.420	133.440

Table 3. Eigen vibration frequencies of cast iron cover (cast iron '1'), taking into account additional masses of parts, resting

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	15.328	50.373	128.097
	liquid	15.186	50.365	114.879
1	vacuum	12.539	25.419	53.460
	liquid	12.477	25.385	53.454
2	vacuum	12.687	36.695	69.340
	liquid	12.620	36.651	69.198

Table 4. Eigen vibration frequencies of cast iron cover (cast iron '2'), taking into account additional masses of parts, resting

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	18.623	62.306	157.287
	liquid	18.449	62.296	140.539
1	vacuum	15.419	30.862	66.202
	liquid	15.342	30.823	66.193
2	vacuum	15.606	44.853	85.848
	liquid	15.523	44.803	85.726

The influence of the additional masses of the parts on the eigenfrequencies of the head covers was investigated both in a vacuum and in liquid. The results of calculating the eigenfrequencies of the vibrations of the cast iron head covers (cast iron '2') without taking into account the additional masses of the parts, with the head cover resting on the stator, are given in Table 5.

Table 5. Eigen vibration frequencies of cast iron cover (cast iron '2'), without taking into account additional masses of parts, resting

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	108.324	287.873	351.912
	liquid	80.388	225.678	317.934
1	vacuum	78.852	176.833	250.943
	liquid	68.334	166.779	245.280
2	vacuum	71.312	173.965	272.786
	liquid	63.232	168.137	266.285

The results of calculating the eigenfrequencies of the vibrations of the head covers, taking into account the additional masses of the parts, with the head cover fastened rigidly along the flange cover to the stator, are given in Tables 6 − 8.

Table 6. Eigen frequencies of vibrations of steel head cover (Cm3), taking into account additional masses of parts, rigid fastening

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	31.303	103.008	181.841
	liquid	31.093	102.977	175.425
1	vacuum	26.435	51.617	108.212
	liquid	26.314	51.567	108.189
2	vacuum	24.997	76.546	135.258
	liquid	24.875	76.455	133.862

Table 7. Eigen frequencies of the vibrations of the cast iron head cover (cast iron '1'), taking into account the additional masses of the parts, rigid fastening

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	16.445	54.215	129.208
	liquid	16.301	54.211	114.652
1	vacuum	14.297	26.401	56.607
	liquid	14.222	26.376	56.604
2	vacuum	13.606	40.672	69.801
	liquid	13.532	40.634	69.619

Table 8. Eigen frequencies of vibrations of cast iron head cover (cast iron '2'), taking into account additional masses of parts, rigid fastening

Harmonic number, KF	Medium	Vibration frequency, Hz		
		1	2	3
0	vacuum	20.016	66.994	158.763
	liquid	19.839	66.989	140.239
1	vacuum	17.578	32.031	70.076
	liquid	17.485	32.003	70.072
2	vacuum	16.732	49.613	86.521
	liquid	16.641	49.572	86.323

Analysis of the results of calculating the forced vibrations of the head covers for the ПЛ 20 В-500 turbines

The forced vibrations of the construction under harmonic loading in time are described by the equation [10].

$$\mathbf{K}u - \omega^2 \mathbf{M}u = Q\,, \tag{6}$$

where **K**, **M** − the stiffness matrix and the mass matrix of the structure, respectively; ω − frequency of vibrations; u, Q − time-varying displacement t vectors and external node load t vectors, respectively.

When solving the dynamics problem (6) by the finite element method, the method of direct integration and the displacement eigenfunction expansion method are usually applied [4, 5].

When using the direct integration method, we build the mass matrix $M_к$ and the rigidity matrix $K_к$ of the construction for any k^{th} harmonic of the expansion with respect to the vector of the amplitude values displacements u_{ik}, applying the developed finite element approach [4].

The dynamic stress-strain state of the head covers of the existing and the new design, made of materials with different elastic characteristics, was investigated under the action of hydrodynamic loads on the head cover at the maximum values of liquid pressure H_{max} = 21 m and power N_{max} = 24.5 MW.

In addition to the mass forces G_i (i = 2, 3, 4) and the hydrodynamic liquid pressure q_1, acting on the liquid contacting surface of the head cover, the latter receives the hydrodynamic axial thrust Q_3, acting on the impeller through the thrust block, and the hydrodynamic force Q_4 from the flow-washed cowl cone. The law of change in hydrodynamic pressure was accepted in the form of $q = q_i \cos(\omega t)$, where t is the time and ω is the loading frequency. The scheme of application of the acting dynamic loads is shown in Fig. 3.

The values of the dynamic loads q_i (i = 1, 3, 4) accepted during the calculation are given in table 9.

Table 9. Dynamic loads

Load variant	Total hydrodynamic load		q_i, MPa
	Q_i, N	Action area, m^2	
1	–	–	0.2100
3	$Q_{3.}$ =3,500,000	2.397664	1.4598
4	Q_4 =1,536,942	3.200102	0.4803

Depending on the frequency of loading, dynamic displacements as well as dynamic stresses are defined both as when the cover is supported along the circumference formed by the studs (option 1 − loading frequency ω_1 = 2.08 Hz, option 2 − loading frequency ω_2 = 8.33 Hz), and when the head cover is fastened rigidly onto the flange surface (option 3 − loading frequency ω_1 = 2.08 Hz, option 4 − loading frequency ω_2 = 8.33 Hz).

The discretization of the meridional section of the design model of the steel head cover on the finite elements for the investigation of the dynamic stress-strain state is shown in Fig. 4, which shows the nodes necessary for an analysis of dynamic displacements.

The values of the dynamic displacements of the steel cover, in the fixed mesh nodes of the finite elements, are shown in Table 10.

The values of the dynamic displacements of the cast iron head cover (cast iron '1'), in the fixed mesh nodes of the finite elements, are shown in Table 11.

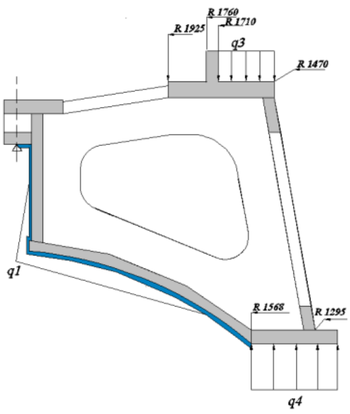

Fig. 3. Dynamic loading on head cover

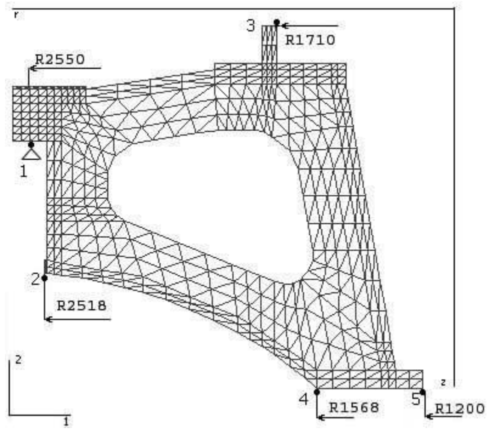

Fig. 4. Fixed mesh nodes of finite elements of design model of head cover

Table 10. Dynamic movements of steel cover depending on type of loading

No. of variant of loading	Displacements	Displacements in nodes ($u_r \times 10^3$, $u_z \times 10^3$), m				
		No. of nodal point				
		1	2	3	4	5
1	u_r	0.000	-0.0042	-0.0052	0.0031	0.00315
	u_z	0.000	0.00098	0.0172,	0.0113	0.0201
2	u_r	0.000	-0.0043	-0.0056	0.0035	0.00358
	u_z	0.000	0.00110	0.0185,	0.0125	0.0218
3	u_r	0.000	-0.00424	-0.00478	0.00291	0.00297
	u_z	0.000	0.00044	0.0162	0.0104	0.0189
4	u_r	0.000	-0.00430	-0.00516	0.00327	0.00335
	u_z	0.000	0.00046	0.0173	0.0115	0.0204

Table 11. Dynamic movements of the cast iron head cover (cast iron '1'), depending on the type of loading

No. of variant of loading	Displacements	Displacements in nodes ($u_r \times 10^3$, $u_z \times 10^3$), m				
		No. of nodal point				
		1	2	3	4	5
1	u_r	0.000	-0.0134	-0.0014	-0.00309	0.00358
	u_z	0.000	-0.01028	0.0208	-0.00117	0.0145
2	u_r	0.000	-0.014	-0.00284	-0.00149	0.00191
	u_z	0.000	-0.00041	0.0271	0.00463	0.0224
3	u_r	0.000	-0.0134	-0.00237	-0.00269	0.00316
	u_z	0.000	0.00041	0.0232	-0.00085	0.0173
4	u_r	0.000	-0.014	-0.00135	-0.00112	0.00151
	u_z	0.000	0.000534	0.0291	0.00631	0.0247

The values of the dynamic displacements of the cast iron head cover (cast iron '1'), in the fixed meshed nodes of the finite elements, are shown in table 12.

Table 12. Dynamic movements of the cast iron head cover (cast iron '2'), depending on the type of loading

No. of variant of loading	Displacements	Displacements in nodes ($u_r \times 10^3$, $u_z \times 10^3$), m				
		No. of nodal point				
		1	2	3	4	5
1	u_r	0.000	-0.009460	-0.000979	-0.002140	0.00241
	u_z	0.000	-0.000743	0.013600	-0.001200	0.00925
2	u_r	0.000	-0.009670	-0.001540	-0.001500	0.00176
	u_z	0.000	-0.000504	0.016000	0.001060	0.01230
3	u_r	0.000	-0.009410	-0.001600	-0.001860	0.00213
	u_z	0.000	0.000210	0.015200	0.000133	0.01110
4	u_r	0.000	-0.009640	-0.002050	-0.001230	0.00148
	u_z	0.000	0.000260	0.017500	0.002300	0.01400

The signs of displacement correspond to the direction of the *R*, *Z* axes (Fig. 4). For illustration, the level of dynamic stresses and the nature of their distribution along the meridional section of the head cover, depending on the characteristics of the material, fastening conditions and the loading frequency, is shown in Figs. 5 – 8.

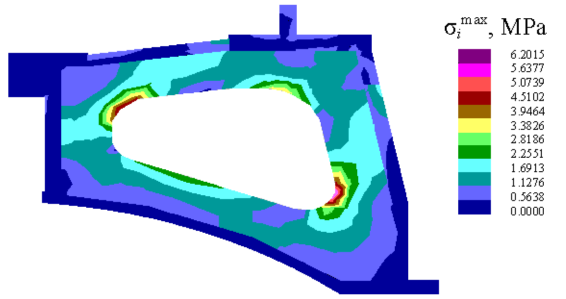

Fig.5. Lines of stress intensity levels in steel head cover, variant 1

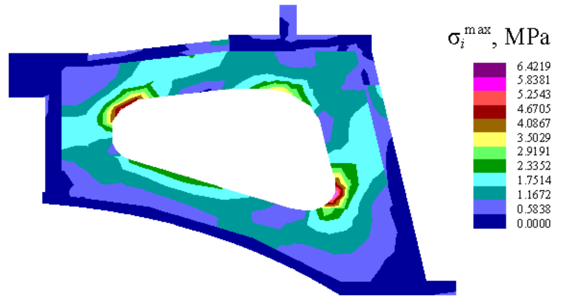

Fig.6. Lines of stress intensity levels in steel head cover, variant 2

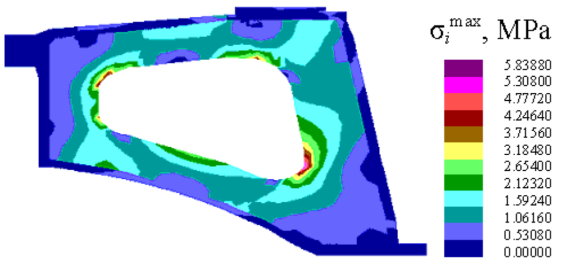

Fig.7. Lines of stress intensity levels in cast iron head cover (cast iron '1'), variant 3

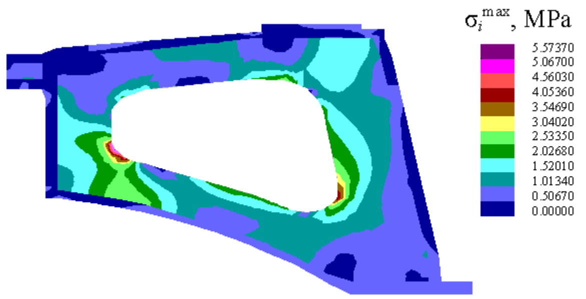

Fig. 8. Lines of stress intensity levels in cast iron head cover (cast iron '2'), variant 4.

The minimum σ_i^{min} and maximum σ_i^{max} values of the intensity of dynamic stresses in the cast iron head cover with different ways of fastening it, possible values of the elastic characteristics and excitation frequencies are given in Table. 13.

The minimum σ_i^{min} and maximum σ_i^{max} values of the intensity of dynamic stresses in the cast iron head cover with different ways of fastening it, possible values of the elastic characteristics and excitation frequencies are given in Table. 14.

Table 13. The intensity of dynamic stresses in the steel cover

Type of fastening	Frequency ω_i, Hz	Minimum stresses σ_i^{min}, MPa	Maximum stresses σ_i^{max}, MPa
		mode 1	mode 1
rigid – along flange line	2.08	0.02240	6.205
	8.33	0.02520	6.413
in points of resting along circumference formed by studs	2.08	0.00849	6.210
	8.33	0.01110	6.433

Table 14. Dynamic stress intensity in cast iron head cover

Type of fastening	Frequency ω_i, Hz	Minimum stresses σ_i^{min}, MPa	Maximum stresses σ_i^{max}, MPa
		E=0.8·10^5 MPa, ν=0.3	E=1.2·10^5 MPa, ν=0.21
rigid – along flange line	2.08	0.00170	0.00170
	8.33	0.00377	0.00200
in points of resting along circumference formed by studs	2.08	0.01410	0.01763
	8.33	0.01610	0.01305

Conclusions

1. The purpose of the investigation was to solve the problem of the possibility of replacing the cast iron cover of the ПЛ 20-В-500 turbine with the one of Ст3 welded carbon steel sheet.

2. In a 3D formulation, the influence of the additional liquid masses of the structure on the eigenfrequencies is taken into account, using mathematical models based on hypersingular equations and a combination of finite and boundary element methods.

The investigation of design models of the head covers for the ПЛ 20-В-500 turbines, whose design features are determined by the composition, type, and size of the turbine, showed that the effect of a liquid on the eigenfrequencies is insignificant (see Tables 2 – 4 and 5 – 8). As the frequency number increases, the effect of a liquid decreases. At the same time, the eigenfrequency of the covers is significantly affected by the value of the additional masses of the parts and units placed on them.

The spectra of the eigenfrequencies of the cast iron and steel covers of the ПЛ 20-В-500 turbines are shifted both relative to each other and the fixed revo-vane frequency ω_2=8,33 Hz during full-scale tests. The detuning of the eigenfrequencies from the excitation frequencies of the steel head cover is higher than that of the cast iron one, which, considering the damping properties of the cast iron, is an important factor.

3. The conducted numerical investigation of the influence of both the material of the head covers and the conditions of fastening on their dynamic stress-strain state revealed that the level of dynamic displacements and stresses is insignificant and depends both on the fastening conditions and the loading frequency. The maximum values of dynamic stresses and displacements occur when the design model of the head cover is fixed along the circumference formed by the studs and the loading frequency ω_2 = 8.33 Hz is fixed during full-scale tests. The maximum level of dynamic displacements in the steel head cover u_z = 0.0218 mm, in the cast iron one u_z = 0.0224 mm. The maximum dynamic stresses in the steel head cover σ_i^{max} = 6.433 MPa, while in the cast iron head cover σ_i^{max} = 6.209 MPa. When the turbine is operating, the dynamic deformations of both steel and cast iron head covers do not disrupt the operation of the shaft seal since the structural radial clearance between the head cover and the shaft seal housing Δ_r is 1.5 to 2.04 mm.

4. The conducted numerical investigations have confirmed the possibility of replacing the cast-iron head cover with the one welded from Ст3 carbon steel sheets, as well as the necessity of tightening the flange connection of the head cover to the stator, which is one of the effective ways of increasing rigidity.

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