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ESTIMATING THE INFLUENCE OF THE RIGIDITY OF SUPPORT ASSEMBLIES ON THE RESONANCE PHENOMENA AND THE VIBRATION STATE OF A HYDRAULIC UNIT

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Many factors affect the vibration of the hydraulic unit, including the natural oscillation frequencies of the structural elements, which in some cases may contribute to an increase in vibration because of the occurrence of resonance phenomena.

In this work, the case of design optimization of the support assemblies of the shaft pipeline of the hydrogenerator-motor was considered, installed at HSPP during its reconstruction in order to improve vibration parameters. The effect of increasing the stiffness of the upper crosspiece and reducing the weight of the rotating parts on the value of the first and second critical rotation frequencies was analyzed. Based on the results of computations, operability of the proposed new reinforced structure of the upper crosspiece of the hydrogenerator-motor with increased rigidity was confirmed. Using this design could increase the first critical rotation frequency from 9 Hz (540 rpm) to 17.6 Hz (1056 rpm). This would lead to the avoidance of resonance phenomena caused by the proximity of the first critical rotation frequency of the rotor to the rated rotation frequency (600 rpm).

The computations were performed in a three-dimensional setting in two stages for each of the considered design cases. At the first stage, the supporting elements rigidity of the shaft line of the hydrogenerator-motor were studied by determining the structural element elastic deformations when it was loaded by a transverse force. At the second stage, these determined stiffnesses of the support elements were used as input data for calculating the critical rotation frequencies of the hydraulic unit rotor

Keywords: umbrella-type hydrogenerator, rotor, vibration, stiffness, critical rotation frequency, three-dimensional computation, flexibility of hydrogenerator supports, hydrogenerator crosshead

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1. Introduction

One of the main directions in the development of modern science and technology is the transition to the use of three-dimensional approaches in the design and calculations of structural elements. At the same time, the approaches should take into account the effect of various factors that can affect the strength and other parameters of the structure.

The use of this principle in the design of new electric machines makes it possible to optimize their structure, as well as opens up additional possibilities for increasing reliability and reducing the weight of the equipment.

A similar approach can also be used in the process of restoration and modernization of existing electrical machinery equipment, in particular, in the reconstruction of generator equipment at hydroelectric power stations. In this case, the use of three-dimensional modeling makes it possible to determine the shortcomings of existing structures and evaluate the effectiveness of technical solutions proposed in the reconstruction project.

Currently, a very important issue is the reconstruction of hydro generators installed at hydroelectric power stations. Existing hydrogenerator-motors had problems related to the high level of vibration of support structures over the entire period of operation. In practical operation, one of the most influential factors of the failure of the support systems of the hydraulic unit, which is associated with vibration, are fatigue cracks and the destruction of components.

The vibration state of the hydraulic unit significantly affects the reliability indicators of its operation. High vibrations of individual structural elements can lead to the destruction and failure of both directly vibrating parts and those adjacent to them.

Analysis of the structure of existing hydrogenerator-motors revealed that the possible reason is the insufficient rigidity of supporting elements, which was obviously not taken into account during the development of the basic design. This significantly affected the critical rotation frequencies of the shaft line of the hydraulic unit, which consists of rigidly interconnected rotors of the hydrogenerator-motor and the pump-turbine.

To verify existing hypotheses, appropriate calculations of critical rotation frequencies can be performed with basic support structures, as well as with the crosshead with increased rigidity proposed in the reconstruction project for existing hydraulic units. Current methodological base does not allow for calculation of natural frequencies of hydro generators without simplifying the geometry. Therefore, analysis of the influence of stiffness of supports on the critical rotation frequencies is relevant from the point of view of optimizing the structures of rotating electric machines and increasing their reliability. The practical significance of a refined analysis of the impact of vibrations will allow us to devise principles for constructing a reconstructed structure of supporting elements.

2. Literature review and problem statement

Analysis of the dynamics of mechanisms with rotating parts has come a long way along with the general development of science and technology [1, 2]. In work [1], the main details representing the basic structure of the case and graphically marked types of the main modes are given. Designs of rotating machines are discussed in detail in [2] but these structures correspond to the design trends adopted earlier. Moreover, their feature is the construction of cases with loss mass, which ensures the transition of vibration to higher harmonics but worsens the mass-dimensional indicators. The main drawback of the material is the inconsistency with the modern concept of the greatest power with the least weight. In particular, the main principles of calculating vibrations and strength in a two-dimensional formulation are specified in [3]. Despite the deep analysis reported in [4], the defined boundary and initial conditions cannot be transferred to three-dimensional problems, and the presented accuracy of results gives a rather high error, which creates a risk of the unit operating on the line of critical modes.

In [5], the finite element method (FEM) is considered, which currently makes it possible to obtain the most accurate results of studies on the dynamics of rotating parts, but because of the rather significant size between the components of the elements, when using the same type of mesh, it is often not possible to obtain the required accuracy at the level of 95 % of the fluidity limit. The development of this method was reported in [6] but taking into account the complex geometry and technological stresses, the work cannot be used in its entirety to calculate the dynamics and strength of the hydrogenerator. In general, FEM is used to analyze many technical aspects in various fields of science and technology, in particular, the stress-strain state of structures, tasks of ventilation, heat exchange, etc.

The topic of dynamic analysis using FEM for the rotor system of an aircraft engine is considered in [7]. Paper [8] describes the features of dynamic analysis of turbine generator rotors in a three-dimensional statement. However, the proposed analysis methods cannot be fully adapted for the rotor of the hydrogenerator-engine because of differences in design caused by a large difference in rotation frequency; in particular, a different spatial position of the rotation axis and a different number of supports. This is because of the impossibility of correctly setting the stiffness of the structure and the change of the modulus of elasticity according to the characteristic geometric dimensions of the structure. The development of this method is [9] but one of the unsolved problems is the lack of boundary conditions for taking into account thermal stresses.

Currently, several approaches are used to consider the problems of dynamic state analysis in the field of hydropower generation. The first is consideration from the point of view of vibration measurements and analysis of vibration changes during the operation of the hydraulic unit [10]. The second is consideration from the point of view of the peculiarities of the operation of hydraulic turbines associated with hydrodynamic processes during the flow of water through the working parts of the turbines [11]. The third approach is to consider the effect of rotor eccentricity and other manufacturing inaccuracies of generators on their vibration [12]. Another research area is the consideration of the influence of hydrodynamic processes in sliding bearings on the dynamic behavior of the rotor [13]. Taking into account the above, designing a reconstructed structure of supporting elements, as well as determining their stiffness for the introduction of the latest technologies in hydro-generator construction requires a solution.

At the same time, the question of the influence of stiffness of the supports on critical rotation frequencies is practically not considered in the field of hydropower. First of all, this is due to the fact that for the vast majority of hydro generators, the first critical rotation frequency is outside the zone of possible operating modes, including the operating mode with an advance rotation frequency when the load is suddenly dropped. Thus, in the general case, the values of critical rotation frequencies of the hydrogenerator rotor practically do not affect its vibration state.

The distance of the critical rotation frequency of the rotor from the line of critical modes reduces the direct influence of vibration on the condition of the hydraulic unit. In turn, this makes it possible to increase the reliability of the structure through the exclusion of sign-changing loads caused by multifactorial vibrations.

3. The aim and objectives of the study

The purpose of our study is to assess the influence of stiffness of the support nodes on resonance phenomena and the vibration state of the hydraulic unit and to devise methods for reducing the vibration of hydraulic units, as well as analyze the causes of its occurrence. This will make it possible to compile recommendations for the reconstruction of the hydrogenerator-motor in order to increase its reliability.

To achieve the goal, the following tasks were set:

– to determine the boundary and initial conditions for the calculation in a three-dimensional statement by analyzing the geometry and features of technology for assembling the supporting elements of the structure to determine their stiffness;

– to analyze the critical rotation frequencies of existing hydrogenerator-motor and compare them with the rotation frequencies in the part of substantiating the reliable operation of the hydro unit under normal modes;

– to design a reconstructed structure of supporting elements, to determine their stiffness for the introduction of the latest technologies in the hydro-generator construction in Ukraine;

– to analyze the critical rotation frequencies of the reconstructed hydrogenerator-motor.

4. The study materials and methods

The object of our study is a hydrogenerator-motor with a capacity of 136.7 MVA/123 MVA under generator mode and 150 MVA/135 MVA under engine mode with a rated rotation frequency of 600 rpm.

The main hypothesis of our research assumes that the deformation processes in the existing structures of the hydrogenerator support nodes occur simultaneously. For a structure with several types of materials, additional forces equivalent to prestressing are introduced. The modulus of elasticity of the steel for the rotor rim is reduced by two times (as for the charged part). As a basis, the decision was made to investigate the existing umbrella-type hydraulic unit.

The main assumption in the work is to consider the rim of the rotor in the form of a solid cylinder. To ensure the accuracy of the calculation, the values obtained from the results of vibration tests are used.

As a simplification, it is assumed that the geometry changes relative to the far radial point of the rotor rim and this change is not taken into account on the opposite element located at an angle of 180° from the base.

The hydrogenerator-motor has a so-called suspended design, that is, it has a support of the cradle on a crosshead located above the rotor. Also, in the upper crosshead of the hydrogenerator-motor, an acceleration engine is installed, which is used during the start-up of the hydrogenerator-motor under pumping mode.

The vibration condition of existing hydrogenerator-motors does not meet the requirements of the ISO 7919-5:2005 standard. Based on the results of analyzing the design, the probable reason is insufficient stiffness of the supporting elements, which reduces the critical frequency of rotation of the rotor to a value close to the rated frequency of rotation.

The existing upper crosshead of the bridge type (Fig. 1) is a welded non-separable structure. The crosshead has a central part, which forms an oil bath for the support and the upper guide bearing. Also, the cross has four legs of a double cross-section. FEM was chosen to assess vibrations and self-oscillations. FFEPlus was used as the solution program. The FFEPlus solver employs advanced matrix reordering, making it more efficient for large problems. Given the limited time of the study, the SolidWorks Simulation software package was used as it makes it possible to solve the problems of vibration and stress-strain state research in one software package.

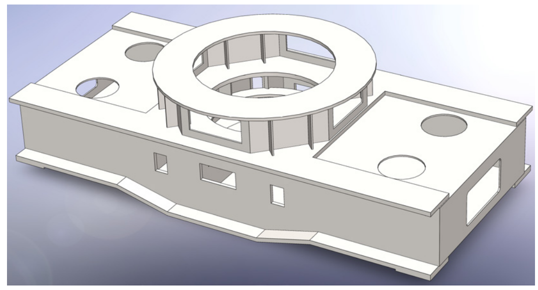

Fig. 1. Actual upper crosshead of the hydrogenerator-motor

Factors affecting the stress-strain state of the upper crosshead are the following loads that it perceives:

a) in the axial direction:

– from the mass of rotating parts of the generator and turbine;

– from the axial hydraulic force of the turbine;

– from the mass of parts of the acceleration engine;

b) in the radial direction:

– from the one-sided magnetic attraction of the rotor of the hydrogenerator-motor;

– from the imbalance of the rotating parts of the hydraulic unit;

c) in the tangential direction:

– from the torque of the acceleration engine;

– from the torque of the turbine.

Taking into account the perceived loads, the crosshead must have sufficient stiffness to minimize flexibility under these loads.

While analyzing the structure of an actual upper crosshead, the following shortcomings were identified, which significantly affect the vibration state of the hydraulic unit:

– the bridge-type crosshead has an imbalance of rigidity in the radial directions;

– in the radial direction, the crosshead is fixed on the stator housing. There are no radial stops of the crosshead legs to the foundation, while the stator body has a relatively low radial stiffness because of the peculiarities of its design, which should compensate for possible radial thermal expansion of the stator core during the operation of the hydrogenerator-motor;

– the legs of the crosshead have low rigidity because of the small thickness of the vertical ribs of 15...20 mm.

The problem of analyzing the stiffness of the support nodes and calculating the critical rotation frequencies of the rotor in the hydrogenerator-motor is solved in a three-dimensional statement.

5. Results of investigating the critical frequencies of rotation of the hydrogenerator-motor

5. 1. Representation of the boundary and initial conditions for the calculation in a three-dimensional statement

The calculation of the critical frequencies of rotation of the shaft of actual hydraulic unit is performed according to the diagram of the location of supports of the shaft conduit of the hydraulic unit, which is shown in Fig. 2, where the locations of the following support nodes are marked:

– the upper generator bearing, located at the place of installation of the upper crosshead;

– the lower generator bearing, located at the place of installation of the lower crosshead;

– turbine bearing.

Table 1 gives main parameters and loads acting on the hydrogenerator-motor.

Table 2 gives the calculated vibrations of the upper crosshead.

Table 3 gives the calculated vibrations of the lower crosshead.

To determine the natural frequencies, a three-dimensional solid model was built (Fig. 3, 4).

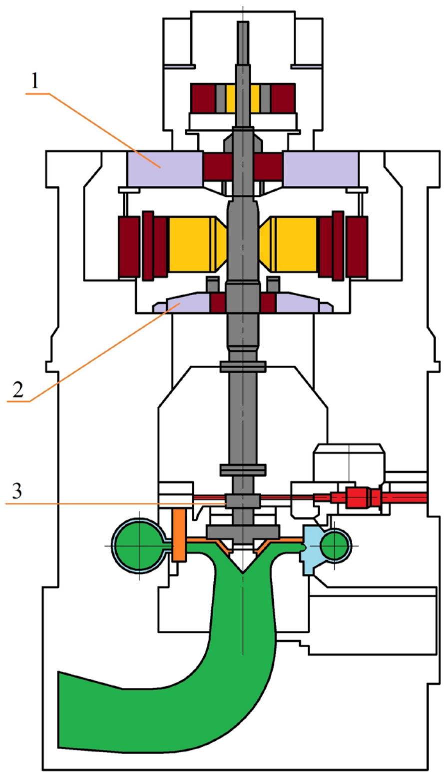

Fig. 2. Location of hydraulic unit shaft support: 1 – upper generator bearing, 2 – lower generator bearing, 3 – turbine bearing

The stiffness of the upper and lower crossheads was determined by applying a unit transverse force to the structure and determining the response of the structure (determination of elastic deformations).

Table 1

Basic parameters and loads acting on the hydrogenerator-motor

Parameter ID	Value
Gross power S, MV·A (generator/engine)	136.7/150
Active power P, MW (generator/engine)	123/135
Rated power factor, a.u.	0.9
Rated voltage U_{rated}, V	13,800
Phase current I, A (generator/engine)	5,719/6,276
Rated speed n, rpm	600
Electric current frequency f, Hz	50
Number of poles 2p	10
Phase connection diagram	star

Table 2

Calculated vibrations of the upper crosshead

Parameter ID	Value before reconstruction	Value after reconstruction
Type of crosshead	Bridge	Radial
The number of paws	4	6
Vibration of the upper crosshead, μm	100–300	<30
Radial supports in the foundation	Not used	Applied

Table 3

Calculated vibrations of the lower crosshead

Type of crosshead	Value before reconstruction	Value after reconstruction
The number of paws	Radial	Radial
Vibration of the upper crosshead, μm	6	6
Radial supports in the foundation	80–200	<30
Type of crosshead	Not used	Applied

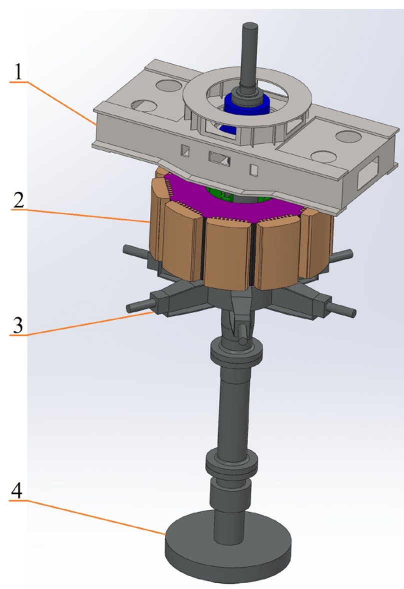

Fig. 3. 3D model of the structure of actual hydrogenerator-motor: 1 – upper crosshead, 2 – rotor, 3 – lower crosshead, 4 – turbine

To ensure the optimal accuracy of the calculation, analysis was performed for the sector of the structure, symmetry conditions were set at the edges. Temperatures obtained in a three-dimensional statement by the CFD method were indicated as thermal boundary conditions.

Calculations in a three-dimensional statement were performed using the SolidWorks Simulation software package.

Fig. 5, 6 show the initial data and the resulting diagram of displacements of an actual upper crosshead.

The calculation is performed in a three-dimensional statement for 1/6 of the crosshead section; the boundary conditions that take into account symmetry are indicated. Fig. 7, 8 show boundary conditions with acting forces and grid parameters under the rated mode and half-pole shorting mode, respectively. In both cases, the grid layout is the same. Grid parameters are given in Table 4.

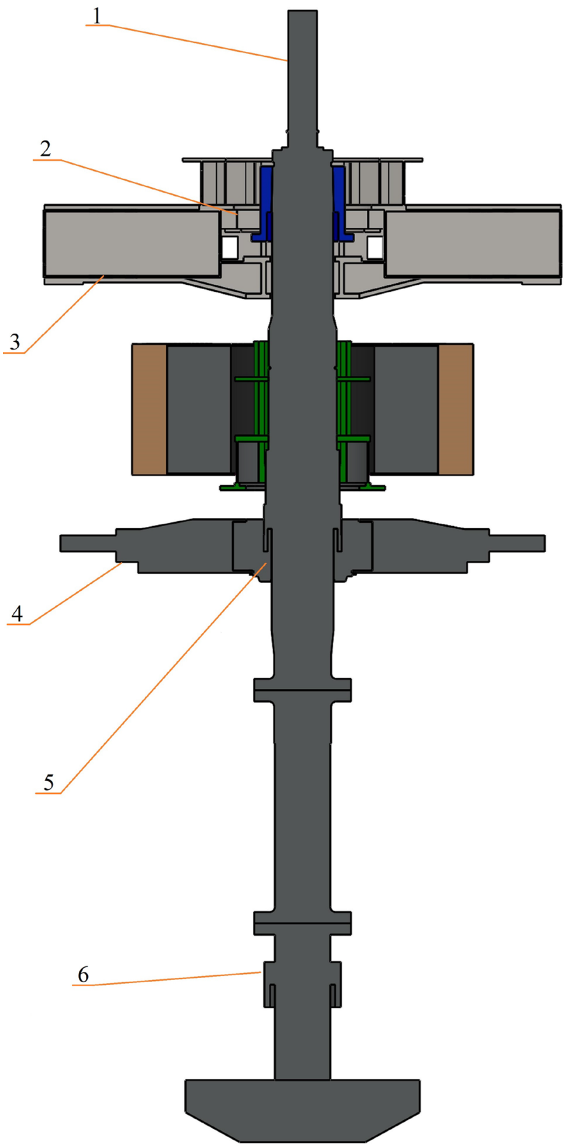

Fig. 4. 3D model of the structure of actual hydrogenerator-motor in cross-section: 1 — acceleration engine, 2 — upper generator bearing, 3 — upper crosshead, 4 — lower crosshead, 5 — lower generator bearing, 6 — turbine bearing

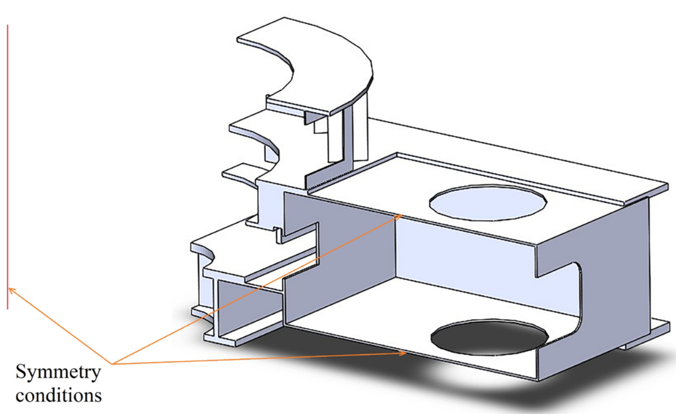

Fig. 5. Three-dimensional model of the sector of an actual upper crosshead with the specified symmetry conditions

A temperature of 20 °C is set as initial conditions for the calculation.

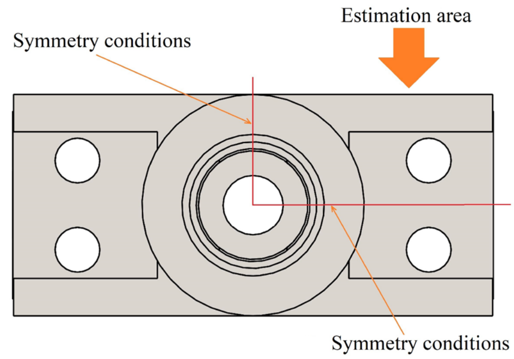

Fig. 6. The estimation area of an actual upper crosshead

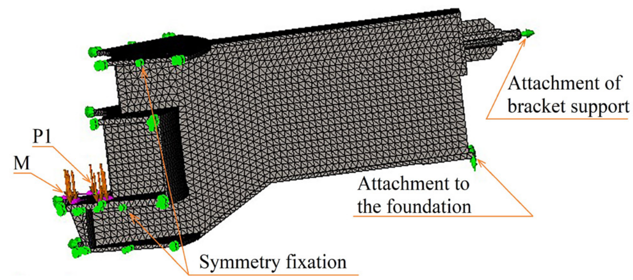

Fig. 7. Boundary conditions, applied loads, grid under the rated mode: $P1$=366.7 kN — 1/6 of the load transmitted by the weight of the rotor and turbine impeller; M=111870 N·m is the moment transmitted by the force of friction in the bracket

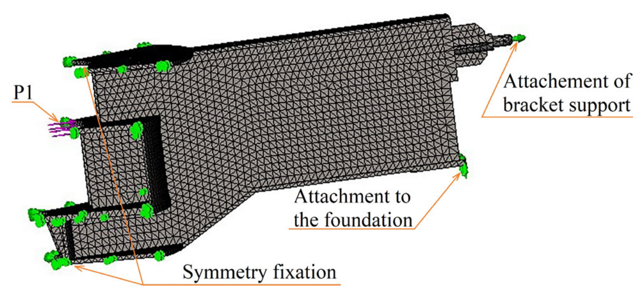

Fig. 8. Boundary conditions, applied loads, grid under the mode of shorting half of the poles: $P1$=520 kN — force from the rotor acting on the paw when shorting half of the poles

Table 4

Grid parameters

Parameter ID	Value
Grid type	Grid on a solid body
The partition used	Regular grid
The maximum size of the element, mm	50
The minimum size of the element, mm	2.5
Total nodes, units	76,028
Total elements, units	38,576
Percentage of elements with an aspect ratio <3 %	78.3
Percentage of elements with an aspect ratio >10 %	0.106

Fig. 9 shows the resulting diagram of displacements of an actual upper crosshead.

Fig. 10–12 show the initial data and the resulting diagram of displacements of the lower crosshead.

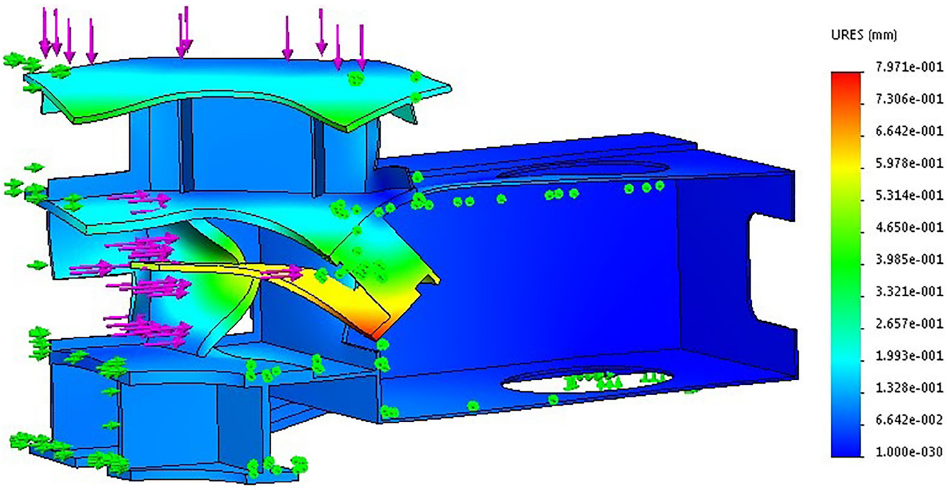

Fig. 9. Diagram of displacements of an actual upper crosshead

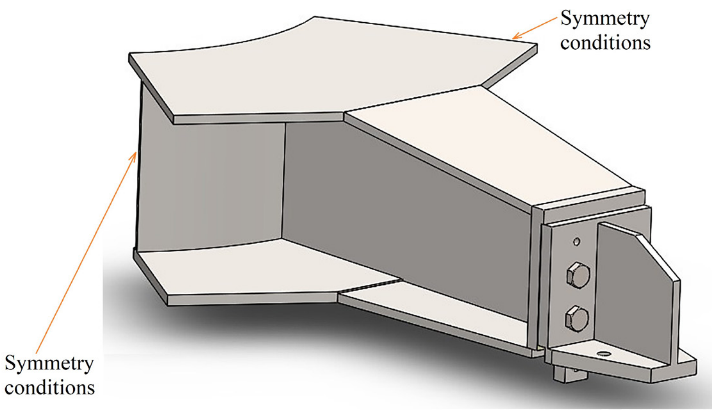

Fig. 10. Three-dimensional model of the lower crosshead with the specified symmetry conditions

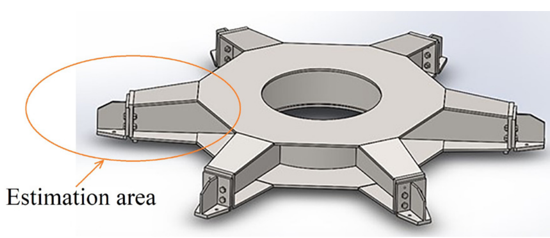

Fig. 11. Estimation area of the lower crosshead

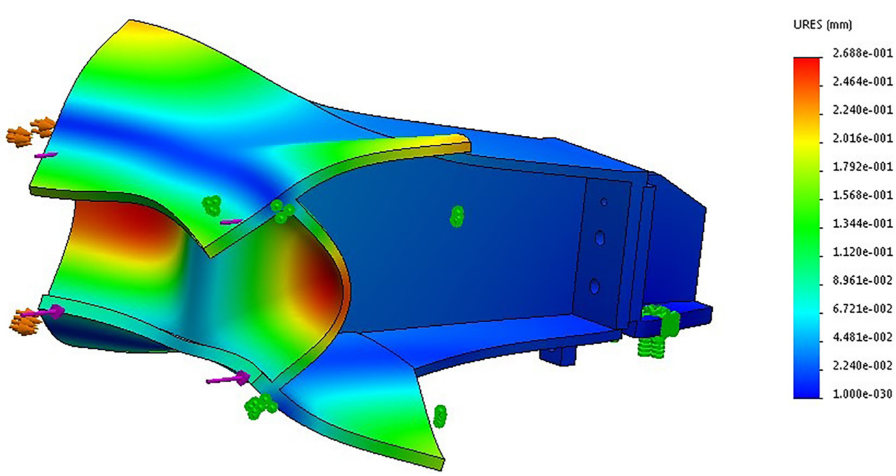

Fig. 12. Diagram of displacements of the lower crosshead

The stiffness obtained from the calculation results for an actual upper crosshead was $2 \cdot 10^8$ N/m, and for an actual lower crosshead – $2 \cdot 10^9$ N/m.

5. 2. Analyzing the critical rotation frequencies of the hydrogenerator-motor and their comparison with the rotation frequencies in terms of ensuring reliable operation under normal modes

Fig. 13 shows the calculation model of the shaft line of the hydrogenerator-motor, on which rigid bearing assemblies and additional mass from the rotor of the acceleration engine are set as boundary conditions.

Based on the results of our calculation, the values of the first critical frequency of 9 Hz (corresponding to the rotor rotation frequency of 540 rpm) and the second critical frequency of 18.2 Hz (corresponding to the rotor rotation frequency of 1092 rpm) were obtained. Fig. 14, 15 show deformation diagrams for the first and second critical shaft rotation frequency.

In Fig. 13–15, the symbol (▲) indicates the location of bearings (fixing the structure).

Fig. 16 shows the amplitude-frequency characteristic of the vibration of the upper crosshead measured during vibration tests of the hydrogenerator-motor, on which the value of the first critical frequency obtained by calculation is indicated by a red vertical line.

The plot (Fig. 16) was built on the basis of the standard recording of the Fourier series through the sum of sin$x$ and cos$x$ (Microsoft Excel software was used):

$$f(x) = a_0 + a_1 \cos x + a_2 \cos 2x + \\ + a_3 \cos 3x + ... + b_1 \sin x + b_2 \sin 2x + b_3 \sin 3x + ..., \quad (1)$$

where $a_0, a_1, a_2, ..., b_1, b_2, ...$ – valid constants, that is:

$$f(x) = a_0 + \sum_{n=1}^{\infty} (a_n \cos nx + b_n \sin nx). \quad (2)$$

In this range from $-\pi$ to π, the coefficients of the Fourier series are calculated from the following formulas:

$$a_0 = \frac{1}{2\pi} \int_{-\pi}^{\pi} f(x) \mathrm{d}x, \quad (3)$$

$$a_n = \frac{1}{\pi} \times \\ \times \int_{-\pi}^{\pi} f(x) \cos nx \mathrm{d}x \, (n = 1,2,3), \quad (4)$$

$$b_n = \frac{1}{\pi} \times \\ \times \int_{-\pi}^{\pi} f(x) \sin nx \mathrm{d}x \, (n = 1,2,3). \quad (5)$$

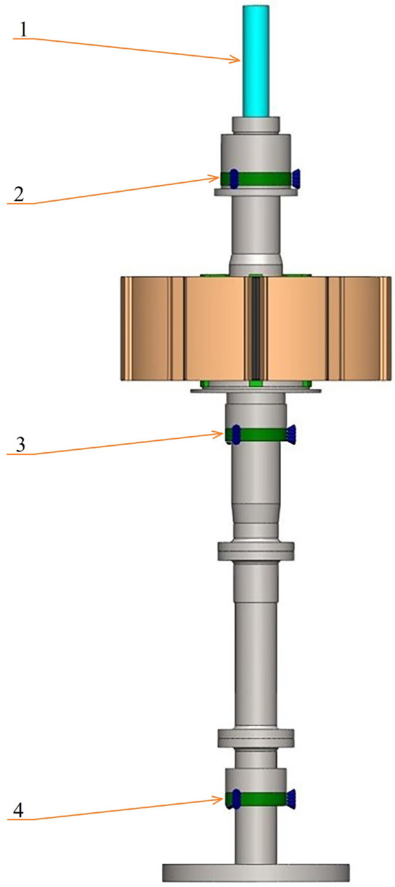

Fig. 13. Calculation model of the shaft line: 1 – additional mass of the rotor of the acceleration motor, 2 – stiffness of the support $C \approx 2 \cdot 10^8$ N/m, 3 – stiffness of the support $C \approx 2 \cdot 10^9$ N/m, 4 – absolutely rigid support

Another way to write the Fourier series is related to the use of the relationship $a\cos x + b\sin x = c\sin(x+\alpha)$:

$$f(x) = a_0 + c_1 \sin(x + a_1) + \\ + c_2 \sin(2x + a_2) + \ldots + c_n \sin(nx + a_n). \quad (6)$$

In this case, a_o is a constant, $c_n = \left(a_n^2 + b_n^2\right)^{1/2}$ is the amplitude of different harmonics, $a_n = \text{arctg}(a_n/b_n)$ is the phase angle of different harmonics.

In the case of numerical expansion of the shape of the rotor or stator in the Fourier series, the integral is replaced by the sum from 0 to 2π (this is equivalent to the sum from −π to π) with a step of $\Delta x = 2\pi/2p$ (where 2p is the total number of generator poles). At the same time, the value of the measured interval opposite the corresponding pole is substituted as $f(x)$ for each of the intervals, and the angle x for each of the intervals is determined from the following formula:

$$x = \frac{2\pi(m-1)}{2p}, \quad (7)$$

where m is the number of the corresponding pole.

In order to eliminate the causes of the above-mentioned vibrations in the reconstruction project, it was proposed to increase the rigidity of crossheads, with the mandatory inclusion of spacer jacks in the structure. It is also proposed to consider the possibility of eliminating the additional weight on the rotor caused by the rotating parts of the acceleration engine.

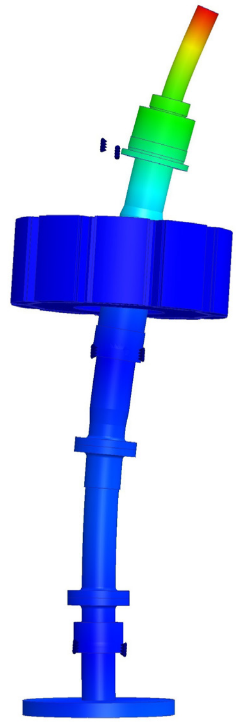

Fig. 14. Deformation diagram for the first critical frequency (9 Hz, scale factor of deformation 220)

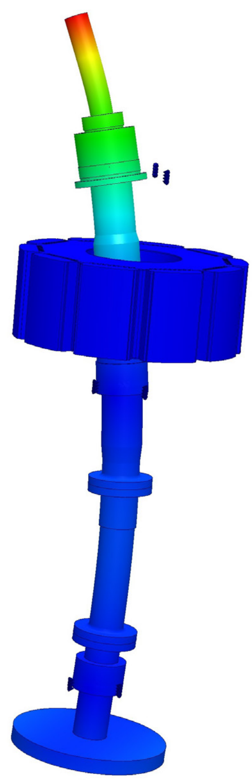

Fig. 15. Deformation diagram for the second critical frequency (18.2 Hz, scale factor of deformation 150)

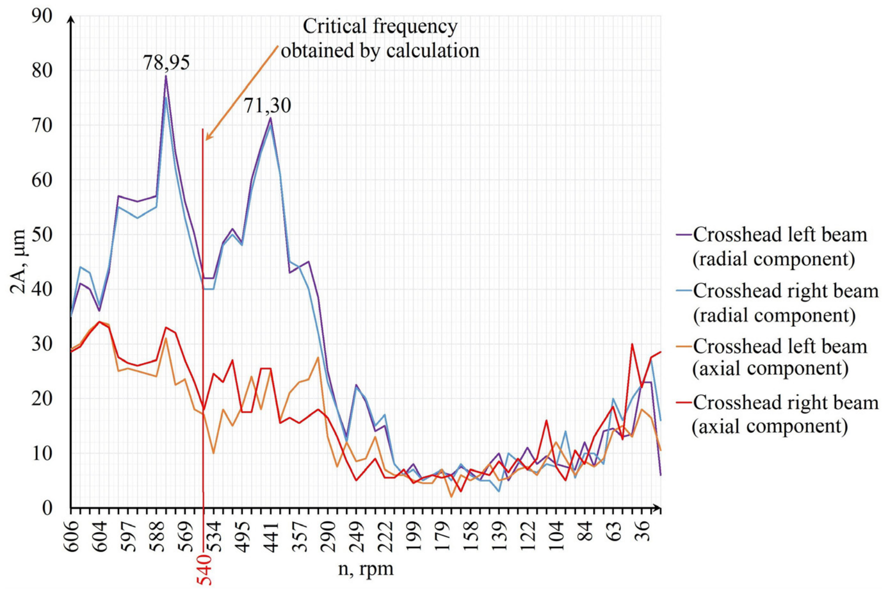

Fig. 16. Amplitude-frequency characteristic of vibration of the upper crosshead (double amplitude)

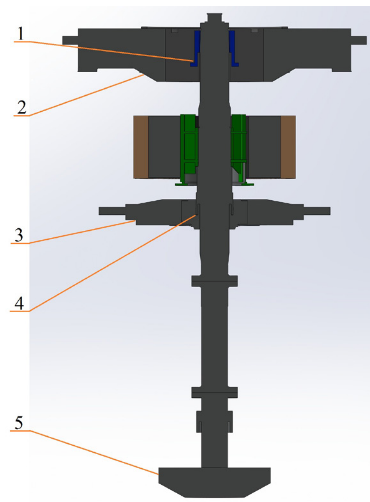

Fig. 18. Three-dimensional model of the structure of the reconstructed hydrogenerator-motor in a cross-section: 1 – upper bearing; 2 – upper crosshead; 3 – lower crosshead; 4 – lower bearing; 5 – turbine bearing

5. 3. Results of investigating the stiffness of the supporting elements of the reconstructed hydrogenerator-motor

The calculation of the critical frequencies of rotation of the shaft of the reconstructed hydraulic unit is carried out according to the diagram of arrangement of the supports of the shaft conduit of the hydraulic unit, similar to the one shown in Fig. 2.

To determine the natural frequencies, a three-dimensional solid-state model with a new crosshead, which has radial legs of increased rigidity, was built (Fig. 17, 18). It is also provided for the installation of radial stops between each of the legs of the crosshead and the foundation.

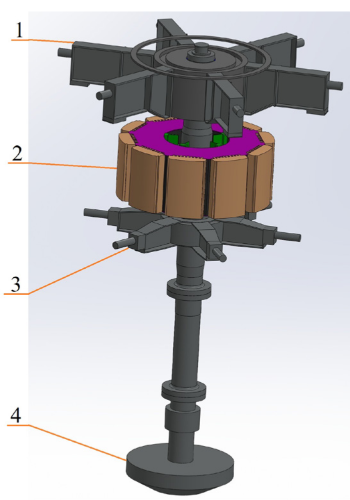

Fig. 17. Three-dimensional model of the structure of the reconstructed hydrogenerator-motor: 1 – upper crosshead, 2 – rotor, 3 – lower crosshead, 4 – turbine

We determined the rigidity of the reconstructed upper beam-type crosshead similarly to determining the stiffness of an actual upper crosshead.

The stiffness obtained from our calculation results for the reconstructed upper crosshead was $2 \cdot 10^9$ N/m, and for the lower crosshead – $2 \cdot 10^9$ N/m.

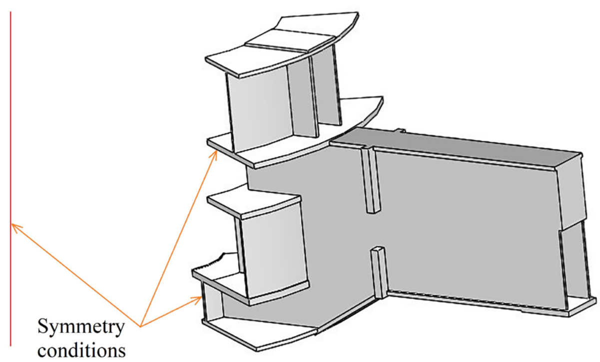

Fig. 19. Three-dimensional model of the reconstructed upper crosshead

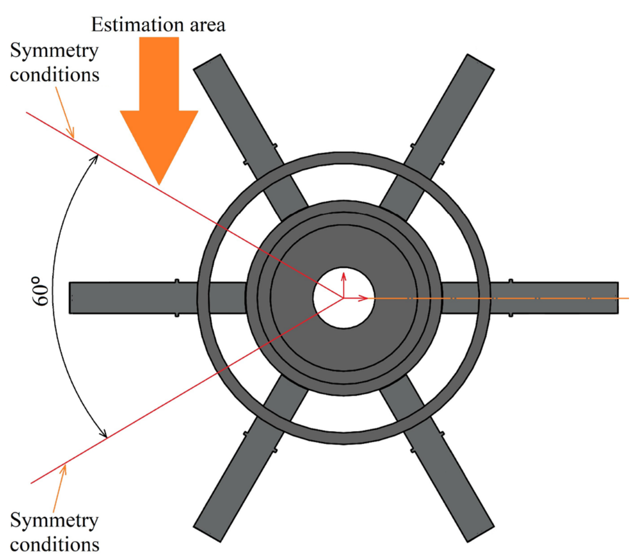

Fig. 20. Estimation area of the reconstructed upper crosshead

A special feature of the reconstructed structure of the upper crosshead is the design of a beam-type crosshead with special stiffness, which fully compensates for the effect of longitudinal loads under normal operation and critical modes. And the proposed jacks make it possible to reduce vibration through the counterforce created in the vertical plane, which fully compensates for the force of shorting half of the poles.

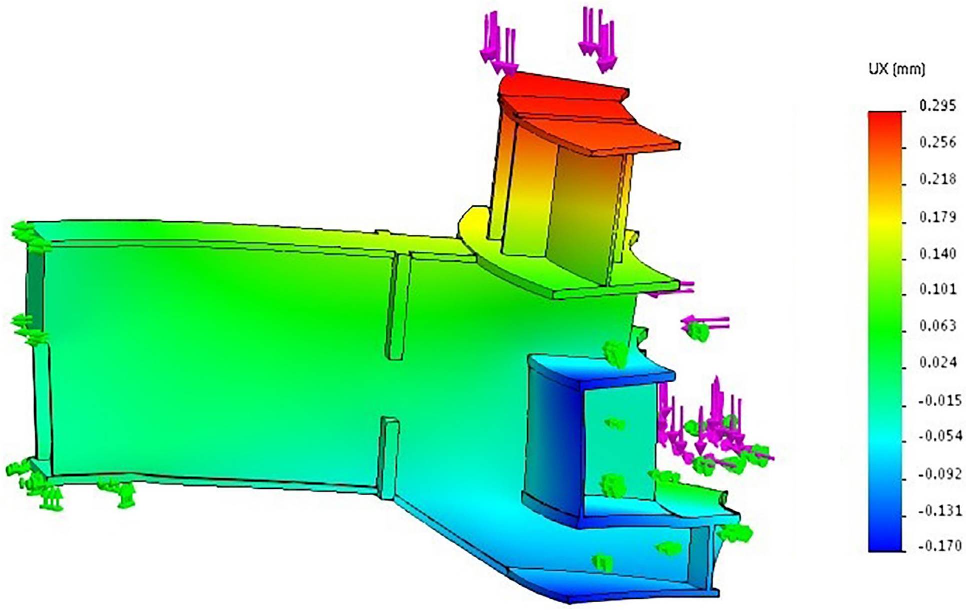

Fig. 21. Diagram of displacements of the reconstructed upper crosshead

5. 4. Investigating the reconstructed structure in terms of critical frequencies of rotation of the hydrogenerator-motor

Fig. 22 shows the calculation model of the shaft line.

For the model (shown in Fig. 22) of the reconstructed hydrogenerator-motor, we set as the boundary conditions the rigidity of the bearing assemblies and the additional mass from the acceleration engine. In this calculation model, the additional weight on the rotor caused by the rotating parts of the acceleration engine is also excluded.

Based on the results of our calculation, the values of the first critical frequency of 17.6 Hz (corresponding to the rotor rotation frequency of 1056 rpm) and the second critical frequency of 23 Hz (corresponding to the rotor rotation frequency of 1380 rpm) were obtained. Fig. 23, 24 show the deformation diagrams for the first and second critical shaft rotation frequency after reconstruction.

In Fig. 22–24, the symbol (▲) indicates the location of bearings (fixing the structure).

Our calculations showed that changing the design of the upper crosshead, eliminating the upper acceleration motor, and installing spacer jacks would make it possible to change the critical frequency of the rotor from 9 Hz (540 rpm) to 17.6 Hz (1056 rpm).

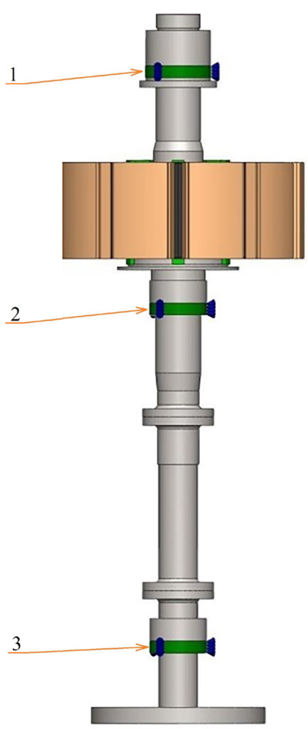

Fig. 22. Calculation model of the shaft line: 1 – stiffness of the support $C \approx 2 \cdot 10^9$ N/m, 2 – stiffness of the support $C \approx 2 \cdot 10^9$ N/m, 3 – absolutely rigid support

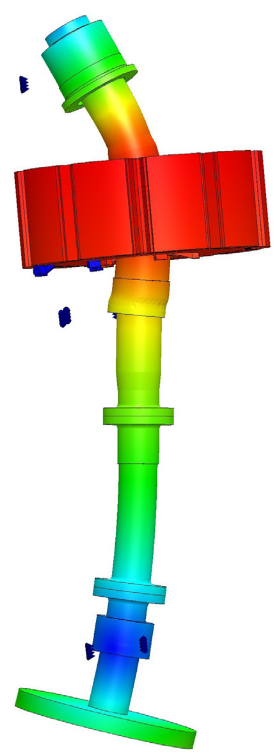

Fig. 23. Deformation diagram for the first critical frequency (17.6 Hz, scale coefficient of deformation 511)

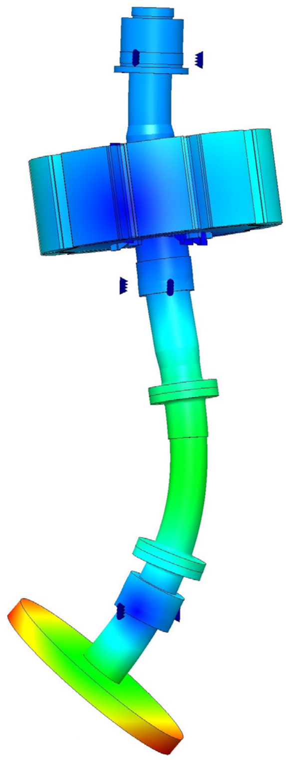

Fig. 24. Deformation diagram for the second critical frequency (23 Hz, scale coefficient of deformation 266)

6. Discussion of results of investigating the critical rotation frequencies

This study has made it possible to determine the causes of vibration based on the experimental research that involved correction by numerical methods. The specified trend lines (Fig. 16) made it possible to conduct a harmonic analysis and theoretically justify the insufficient stiffness of the upper crosshead structure. On the basis of the devised method, the first critical frequency of 9 Hz was obtained for an actual structure, which corresponds to the rotation frequency of the rotor in the hydrogenerator-motor of 540 rpm (Fig. 14). At the same time, it should be noted that the rated rotation frequency of the rotor of this hydrogenerator-motor is 600 rpm, that is, the forced frequency (rotation frequency) and the natural vibration frequency of the shaft line (calculated first critical frequency) are extremely close to each other.

In addition, the critical frequency of rotation is slightly lower than the rated frequency of rotation, which leads to the fact that when starting and stopping, the rotor of the hydraulic unit passes through its first critical frequency. Such a situation is typical for turbogenerators, in which, because of the high rotation frequency and design features caused by this factor, the rotor passes through one critical rotation frequency during startup (and in some cases even two critical frequencies). However, for hydraulic units, such a situation is quite rare and undesirable from the point of view of ensuring their reliable operation.

Thus, owing to close frequency values, resonance is observed, which leads to high vibrations during the operation of the hydraulic unit. Therefore, actual hydraulic units over the entire period of their operation worked under regimes close to critical ones, which is unacceptable.

The second critical frequency of 18.2 Hz (Fig. 15) is at a great distance from the rated one, so its influence on the structure is not considered.

Taking into account actual structures and the possibility of calculating the spatial vectors of forces, the upper crosshead was calculated and reconstructed by the presented method, taking into account the tightening moment of the spacer jacks (pre-tensioning of the crosshead). After conducting an in-depth analysis, it was possible to propose new structural solutions (Fig. 20, 21), which allow us to exclude long-term operation under the resonant mode. Designed on the basis of results obtained by the devised method of three-dimensional modeling, the new structure of the support elements makes it possible to improve the vibration state of the hydraulic unit in accordance with the requirements of the ISO 7919-5:2005 standard (Fig. 22).

The quality of our results is explained by taking into account the geometry of the structure, technological features of assembly (technological stresses), the actual modulus of elasticity of materials, the construction of a complex grid that takes into account the curvature of the components, as well as taking into account thermal boundary conditions. A feature of the devised method for three-dimensional modeling of a hydrogenerator-motor was the introduction of several differences from existing methods. The first is the construction of geometry, which includes even small structural elements, unlike the models reported in [5, 6]. The second is the use of the modulus of elasticity, which was calculated during laboratory studies for the steel actually used in this design, in contrast to the calculation in [3]. The third was mathematically calculated at the pre-design stage of the stiffness of the supports, which depend on the geometry, type of selected oil and temperature factors, in contrast to the calculation algorithms reported in [7, 8]. Because of the fact that the length of the rotor exceeds five meters, the conditions of the first kind were taken into account in the devised method, in contrast to [4].

The use of a reconstructed design of the crosshead for the operating hydraulic unit made it possible to reduce vibrations by removing the critical frequencies beyond the line of operating modes.

The results of our study could be used to improve the vibration state and ensure reliable operation during the reconstruction of hydro-generators and hydrogenerator-motors at the following power plants:

– Dniester HPP (SVO2-1255/255-40 with a capacity of 324 MW under generator mode and 416 MW under engine mode);

– Kyiv HPP (SVO 733/130-36M with a capacity of 33.4 MW under generator mode and 40 MW under engine mode);

– Middle Dnieper HPP (HSV 1230/140-48 with a capacity of 117 MW);

– Dnipro HPP-2 (SV1 1230/140-56M with a capacity of 119 MW);

– Kanivska HPP (SGK3 538/160-70 with a capacity of 22 MW).

The method devised, as well as the structures proposed, cannot be used to calculate the natural frequencies of turbogenerators and asynchronous motors with a rotation frequency of 3000 rpm. This is related to a different type of bearings, the absence of a rocker arm, and the multi-stage structure of the turbine. The design of the rotors is also significantly different: unlike clear-pole hydrogen generators, turbogenerators are implicit-pole machines with another type of mass redistribution.

The main drawback of the study is that existing computer equipment cannot take into account the contact problem for the rotor rim in terms of determining the modulus of elasticity.

Further studies may focus on taking into account the pressing of the FEM rotor rim in terms of taking into account the moment of tightening the tie pins in a three-dimensional statement.

7. Conclusions

1. Our study has made it possible to determine the boundary and initial conditions for the calculation in a three-dimensional statement by analyzing the geometry and features of the technology of assembling the supporting elements of the structure to determine their stiffness. The values of the critical rotation frequencies of the hydraulic unit were obtained, calculated in a three-dimensional statement based on the previously determined stiffness of the supporting elements. In order to eliminate the causes of high-frequency vibrations, it was proposed to increase the rigidity of crossheads, with the mandatory inclusion of spacer jacks in the design. It was also proposed to consider the possibility of eliminating the additional weight on the rotor caused by the rotating parts of the acceleration engine.

2. An actual structure of the supporting elements in a hydraulic unit was considered. The determined value of the first critical rotation frequency was 9 Hz (540 rpm). According to the research results, it was established that actual design of the support nodes has a number of significant shortcomings that negatively affect the vibration state of the hydraulic unit because of resonance phenomena due to the approach of the first critical rotation frequency to the rated frequency of 10 Hz (600 rpm). To ensure a significant increase in stiffness, a new structure of the upper crosshead in a hydrogenerator-motor has been proposed. The double amplitude of vibration of the upper crosshead before reconstruction was 100–300 μm, and after reconstruction it was <30 μm. The double amplitude of vibration of the lower crosshead before reconstruction was 80–200 μm, and after reconstruction – <30 μm.

3. Reconstructed structures of the upper and lower crosshead with increased rigidity were constructed. The stiffness obtained from the calculation results for the reconstructed upper crosshead was $2 \cdot 10^9$ N/m, and for the lower crosshead – $2 \cdot 10^9$ N/m. The reconstructed structures were designed by using the devised method, which takes into account the geometry of the structure, the features of fastening the components and mechanical loads in a three-dimensional statement.

4. The critical rotation frequencies of the reconstructed hydrogenerator-motor were analyzed. The use of three-dimensional modeling has made it possible, based on the obtained stiffness of the reconstructed crosshead, to calculate the critical rotation frequencies of the rotor in a hydraulic unit and to theoretically substantiate the performance of the hydrogenerator-motor with a new crosshead. The derived value of the first critical rotation frequency was 17.6 Hz (1056 rpm). The implemented design solutions allowed us to avoid resonance phenomena due to the distance of the first critical frequency from the rated rotation frequency of 10 Hz (600 rpm). Also, these solutions make it possible to significantly improve the vibration state of the hydraulic unit with implemented reconstructed structures by increasing the rigidity of supports several times. The convergence between the theoretical results of our calculation and measurement of vibration using standard station equipment is at the level of measurement error.

Conflicts of interest

The authors declare that they have no conflicts of interest in relation to the current study, including financial, personal, authorship, or any other, that could affect the study, as well as the results reported in this paper.

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Data availability

All data are available, either in numerical or graphical form, in the main text of the manuscript.

Use of artificial intelligence

The authors confirm that they did not use artificial intelligence technologies when creating the current work.

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