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Abstract

A method is proposed for solving the problems of optimal design of cyclically symmetric structures under static loading, which has been tested on critical structural elements of hydraulic turbines. One of the basic problems in the design of hydraulic turbines is considered, namely, ensuring their strength and reliability under continuous operation under the influence of a static loading. The problem of optimal design of the initial and modified covers of a rotary-blade hydraulic turbine operating in the normal mode has been solved. A Kaplan turbine cover is a complex spatial structure consisting of thin-walled elements. Therefore, the finite element method is used for the calculation to most fully take into account the design features and the spectrum of external influences acting during operation. As the initial design, covers with an initial and modified hole in the rib were selected. The geometric parameters of the cover are modified to minimize the cover weight. The thicknesses of structural elements are taken as design variables. The minimum and maximum thicknesses, as well as maximum stress intensity values are limited. The objective function is the cover weight. The problem of optimal design is solved with the help of the gradient method using a finite-difference analogue of a gradient of the objective function. The distribution of axial displacements and stress intensity in the original and modified cover design during normal operation was obtained. It was found that the mass of the cover structure was reduced by 30%, and the rolled stock thickness range was downsized by five positions, which is significant in the manufacture of a new design. In this case, the stress values in the optimal structure during the modification of the hole in the ribs did not exceed the admissible values. The proposed approach will subsequently be applied to the analysis of elements of aircraft structures.

Keywords: Optimization - Kaplan turbine - Strain-stress state - Gradient method

Optimal Design of the Cyclically Symmetrical Structure Under Static Load

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Abstract. A method is proposed for solving the problems of optimal design of cyclically symmetric structures under static loading, which has been tested on critical structural elements of hydraulic turbines. One of the basic problems in the design of hydraulic turbines is considered, namely, ensuring their strength and reliability under continuous operation under the influence of a static loading. The problem of optimal design of the initial and modified covers of a rotary-blade hydraulic turbine operating in the normal mode has been solved. A Kaplan turbine cover is a complex spatial structure consisting of thin-walled elements. Therefore, the finite element method is used for the calculation to most fully take into account the design features and the spectrum of external influences acting during operation. As the initial design, covers with an initial and modified hole in the rib were selected. The geometric parameters of the cover are modified to minimize the cover weight. The thicknesses of structural elements are taken as design variables. The minimum and maximum thicknesses, as well as maximum stress intensity values are limited. The objective function is the cover weight. The problem of optimal design is solved with the help of the gradient method using a finite-difference analogue of a gradient of the objective function. The distribution of axial displacements and stress intensity in the original and modified cover design during normal operation was obtained. It was found that the mass of the cover structure was reduced by 30%, and the rolled stock thickness range was downsized by five positions, which is significant in the manufacture of a new design. In this case, the stress values in the optimal structure during the modification of the hole in the ribs did not exceed the admissible values. The proposed approach will subsequently be applied to the analysis of elements of aircraft structures. AQ1

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1 Introduction

When designing hydraulic turbines, one of the main problems is to ensure their strength and reliability during continuous operation under the influence of static and dynamic loadings [1–3].

The specific feature of a hydroelectric power station workflow requires special design solutions that ensure reliable operation of units and structures, one of which is the cover of a hydraulic turbine. It is a large-size welded fixed ring part that limits the flow part from above and serves as a base for accommodating guide apparatus parts.

Despite significant achievements in the study of the strength of cyclically symmetric metal structures, the study of their reliability remains relevant. That is why, design of hydropower turbines for hydroelectric power plants requires methods for determining their strain-stress state, allowing to create design models with sufficient accuracy [4].

Typically, the shape of impellers of a hydraulic turbine is optimized [5–7]. The problems of optimal cover design are less studied [8, 9].

In this work, the strain-stress state of a Kaplan turbine cover is analyzed using advanced effective methods and programs for calculating the strength and characteristics of welded load-bearing structures. The methods and programs are based on the elasticity theory the finite element method and the theory of thin plates and shells [10–13]. The optimization problem is solved using the gradient method [14, 15].

The aim of the paper is optimal design of the initial and modified Kaplan turbine cover configurations. The objective function is the cover weight. The optimization parameters are the thicknesses of the structural elements.

2 Model of a Kaplan Turbine Cover

A finite-element cover model of a Kaplan turbine under static axisymmetric load is offered. The cover is a spatial cyclically symmetric structure consisting of thin-wall shells of revolution joined by n ribs. The ribs are meridional plates of complex configuration. Thus, the cover consists of sectors, on whose boundaries the conditions of cyclic symmetry are satisfied. The development of a model with such structures begins with developing a sector model.

When constructing a sector model, the key points in the plane of the rib, along which the lines are drawn, are first defined, and then a rib model is created. To obtain the shell parts of the structure and a complete sector model, the lines of intersection of ribs and shell surfaces are rotated clockwise and counterclockwise through an angle of $360/(2n)$, where n is the number of sectors.

Since the cover is a spatial structure consisting of thin-wall elements, for which the ratio of the thickness of the structural elements and the characteristic size does not exceed 1/10, the theory of thin plates and shells is used. The system of governing equations is.

$$[K]\{u\} = \{F\},$$

where $[K]$ is stiffness matrix; $\{u\}$ is vector of nodal displacements; $\{F\}$ is vector of forces determining the influence of external loads.

To solve the problem, a triangular elastic shell finite element with three nodes is used. An element in each node has six degrees of freedom, namely displacements in the direction of the coordinate X, Y and Z axes and rotations about them.

The model is divided into finite elements, after which the conditions of cyclic symmetry, as well as the conditions of structure fixing and loading are introduced at the boundaries with neighboring sectors.

Figure 1 shows the scheme of the cover. To place the mechanisms and reduce the cover weight, round holes are provided in the ribs. Curved holes are created in the annular plates in the form of a blade profile, which are designed for dismantling and repairing individual blades without completely disassembling the guide apparatus. The cover has such overall dimensions: diameter 3.44 m; height, 1.05 m.

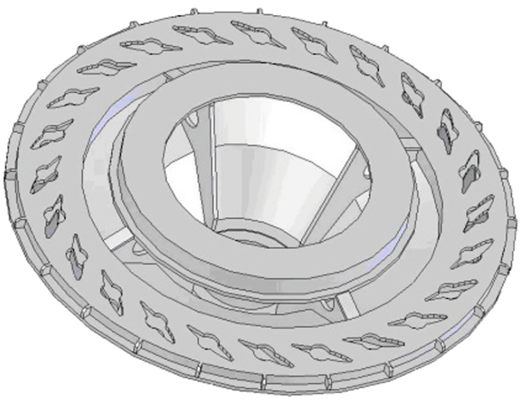

Fig. 1. Kaplan turbine cover.

The cover is made of sheet steel St20 or its analog ASTM A516 Gr.60. The mechanical properties of the material are as follows: $E = 2.1 \cdot 10^5$ MPa is Young's module; $\nu = 0.3$ is Poisson's ratio; $\rho = 7850$ kg/m^3 is material density; $\sigma_\tau = 215$ MPa is yield strength; $\sigma_s = 430$ MPa is ultimate strength; $[\sigma] = 0.5 \cdot \sigma_\tau = 107.5$ MPa is admissible stress.

The design scheme is adopted as a cover sector with a solution angle of 90° and symmetry conditions at the boundaries (see Fig. 2).

The conditions are introduced for fixing the cover on the supporting surface of the flange connecting it to the stator ring, which is considered absolutely rigid, along the circumference on which the studs of the flange connection are located.

Figure 3 shows the scheme of cover loading and fastening. The weight of the generator and impeller is taken into account in the form of equivalent pressure $P = 2.45 \cdot 10^5$ N applied to the surface of the upper ring (see Fig. 3).

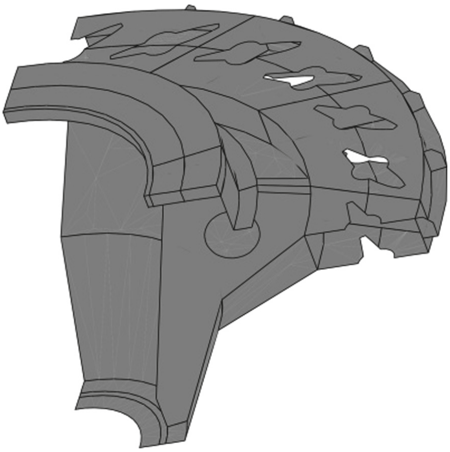

Fig. 2. Cover sector.

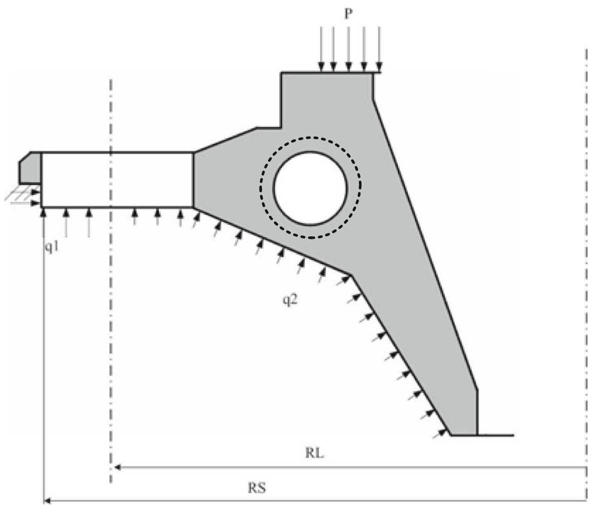

Fig. 3. Scheme of cover loading and fastening.

In the normal mode, the hydraulic pressure $q2 = 0.0965$ MPa is applied to the bottom. During an emergency shutdown of the turbine unit, the pressure in the supply pipe from the radius of the circle *RL*, on which the guide apparatus vanes are located to the circle radius on which the studs of the flange connection *RS* are located, rises sharply from $q2$ to $q1 = 0.1254$ MPa. Therefore, numerical results are presented precisely for this case.

In Fig. 3, the dotted line shows the contour of a modified hole in the cover rib to place equipment and reduce the weight. The radius of this hole is increased 1.5 times relative to the original one. Figure 4 shows the finite element model of the cover sector.

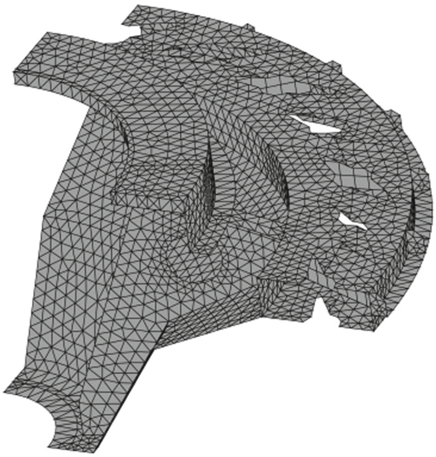

Fig. 4. Finite element model of the cover sector.

3 Stress Analysis of the Cover

First, we obtain the distribution of axial displacements (Fig. 5) and stress intensity (Fig. 6) in the cover in the normal mode.

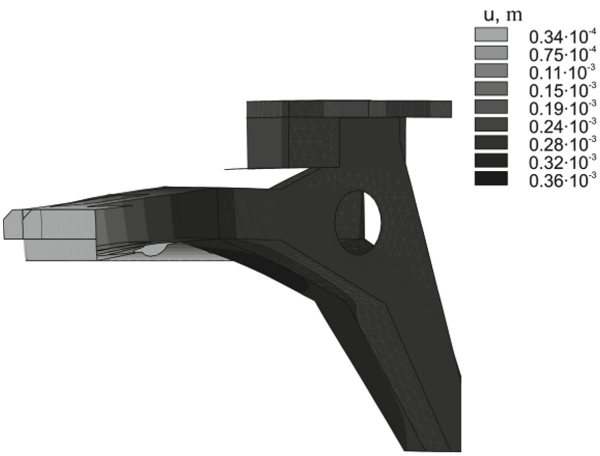

Fig. 5. Distribution of axial displacements in the cover sector.

Maximum stresses occur in the ribs, which are located in the duct where the vanes of the guide apparatus pass. In Fig. 6, the arrow (→) shows the zone of highest stress concentration. Calculation yields zones where maximum displacements occur. They are located in the duct on the guide apparatus side. The maximum displacement and

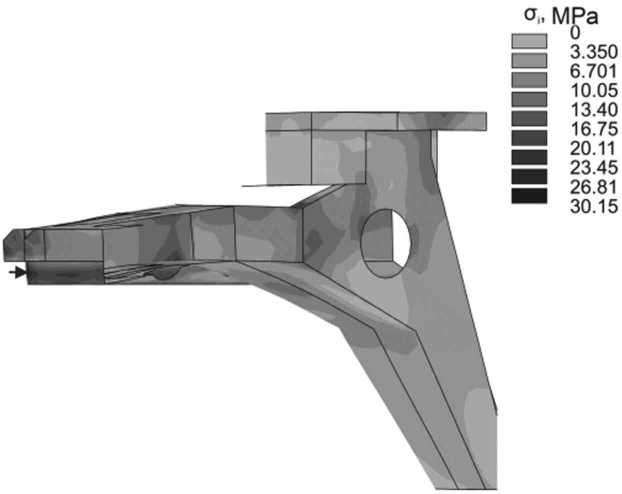

Fig. 6. Distribution of stress intensity in the cover sector.

stresses have the following values: $u_{\max} = 3.6 \cdot 10^{-4}$ m and $\sigma_{\max} = 30.15$ MPa, respectively.

Similar distributions of axial displacements and stress intensities in the cover in the normal mode were obtained when the hole was modified (see Fig. 3). The calculation results are shown in Fig. 7 and Fig. 8.

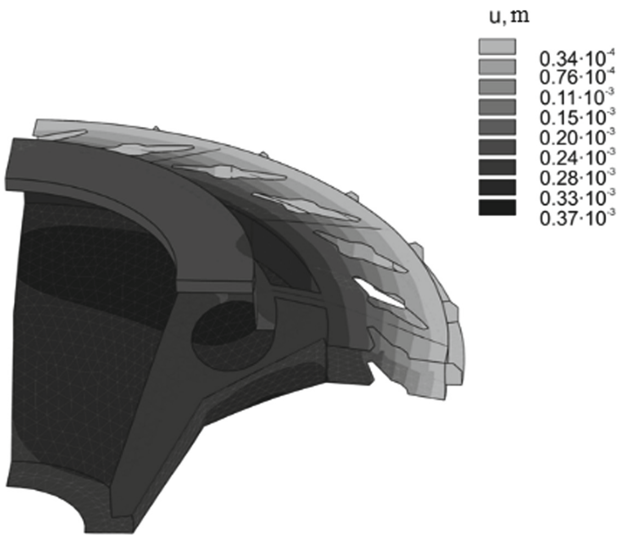

Fig. 7. Distribution of axial displacements in the cover sector with the modified hole.

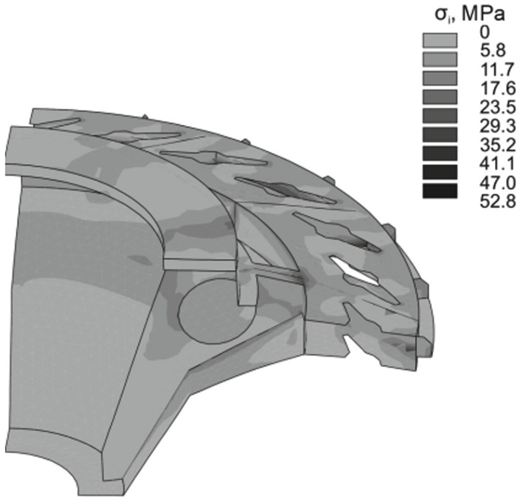

Fig. 8. Distribution of stress intensity in the cover sector with the modified hole.

4 Statement of the Optimal Design Problem

In a broad sense, the general problem of nonlinear programming consists in finding the extreme point.

$$\mathbf{C} = \mathbf{C}^*, \ \mathbf{C} \in E_m,$$

where E_m is the space of design variables at which the objective function reaches a minimum value.

$$F^* = F(\mathbf{C}^*) = \min F(\mathbf{C}),$$

and constraints are met.

$$G_j(\mathbf{C}^*) \geq 0, \ j = \overline{1, J}.$$

Here, **C** is the vector of the space of design variables. The objective function is the cover weight. The design variables were the thicknesses of the cover structural elements, namely shells, plates and ribs. Constraints are imposed on the minimum and maximum values of thicknesses. This is most often the case because of manufacturing and operational requirements. The minimum possible thickness for all elements is 0.016 m. The maximum thicknesses are the initial values of thicknesses of the not modified cover. The maximum stress intensity values are limited by the admissible value $[\sigma] = 107$ MPa.

The problem of optimal design is solved with the gradient method using a finite-difference analogue of a gradient for the objective function described previously [14].

5 Optimization Results

The numerical solution of the optimization problem yields the optimal thicknesses of shells, plates and ribs of the cover. As a start design, the covers with initial and modified holes were selected in turn.

Figure 9 and Fig. 10 show the distributions of axial displacements and stress intensity in the optimal cover, respectively. Maximum stresses arise in the area of ribs as indicated by the arrow (→).

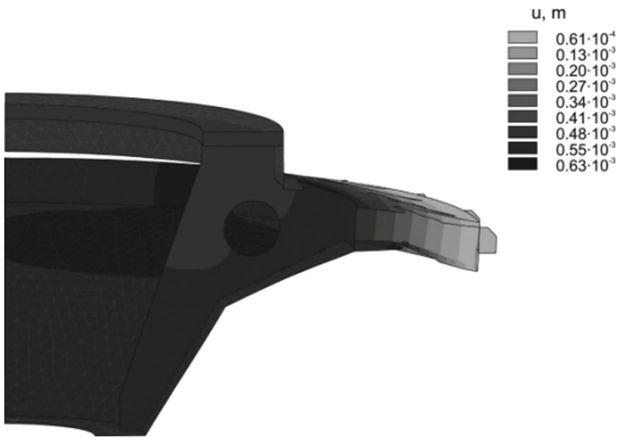

Fig. 9. Distribution of axial displacements in the optimal cover.

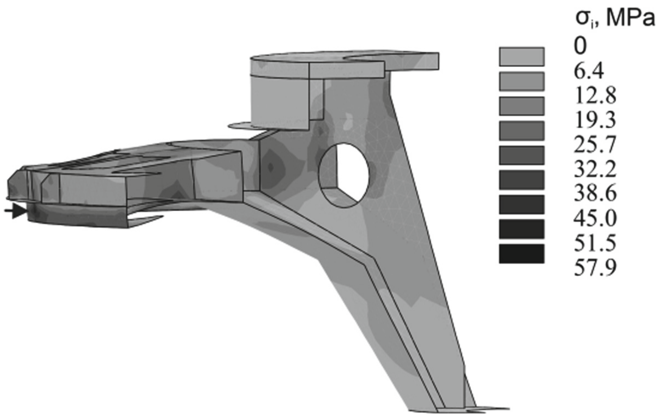

Fig. 10. Distribution of stress intensity in the optimal cover.

Next, the cover with the modified hole was optimized. The distribution of axial displacements and stress intensity in the optimal cover in the normal mode with the modified hole is shown in Fig. 11 and Fig. 12, respectively.

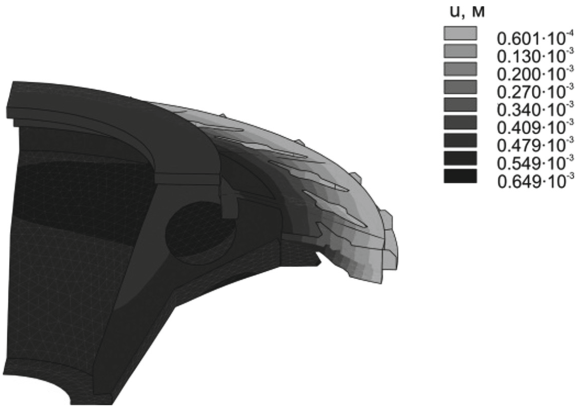

Fig. 11. Distribution of axial displacements in the optimal cover.

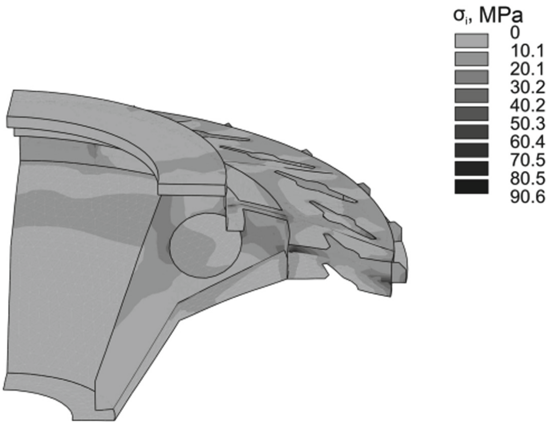

Fig. 12. Distribution of stress intensity in the optimal cover with the modified hole.

Table 1 presents the maximum values of stress intensity $\sigma_{\max}$, axial displacements $u_{\max}$ and the cover weight obtained by design of original and optimal structures. The optimal design is characterized by the following range of structural thicknesses (shells, plates and ribs) 0.018 m, 0.02 m, 0.022 m, 0.03 m, 0.032 m, and 0.04 m.

Table 1. Parameters of optimal design of the cover.

Cover structure	Stress intensity $\sigma_{\max}$ [MPa]	Displacement $u_{\max} \cdot 10^{-3}$ [m]	Weight [kg]
Original cover	32.2	0.371	4080.7
Optimal cover with the hole	57.9	0.630	2578.5
Optimal cover with the modified hole	90.6	0.649	2569.2

During optimization, only constraints on the minimum thicknesses were applied. Hence, the thicknesses of all elements of the optimal cover turned out to be the same and equal to 0.16 m. The thickness of the flange of the cover attached to the stator remained equal to 0.03 m.

Note that the optimal cover weight is 1510 kg less than that of the initial cover. In addition, the nomenclature of optimal design elements was reduced by five positions, which is a clear advantage.

The stresses in the optimal cover for all structures do not exceed admissible values.

6 Conclusions

The paper suggests an approach to minimum weight design of Kaplan covers subject to geometrical and strength constraints. Optimal design parameters are the structural thicknesses of covers. Constraints are imposed on the minimum and maximum values of thicknesses of shells, plates and ribs in the cover. Constraints are also imposed on maximum stress intensity values.

A special technique based on the finite element method was developed to analyze the strain-stress state of covers under a static axisymmetric load. The optimization problem is formulated in terms of nonlinear programming, and then solved using the gradient method.

Several numerical examples are given that allow following the variation of the optimal design depending on the type of covers. Covers with holes and without holes in ribs were considered.

In all the cases, the weight of the optimal cover is one third less than that of the original structures, and the stresses in the optimal covers do not exceed the admissible values. In addition, the rolled stock thickness range is downsized by five positions. That is crucial for manufacturing turbines.

In practice, the execution of an optimal design is limited by the thicknesses of structural elements. However, optimal design makes it possible for a designer to see how close it fits an optimal one. Therefore, optimal designs similar to those given here may be useful when designing real turbines and aircraft structures.

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