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DEVISING A METHOD FOR CALCULATING THE STRUCTURE OF EFFICIENT COOLING SYSTEMS FOR THRUST BEARINGS AND GUIDE BEARINGS IN HYDROGENERATORS

Oleksii Tretiak
Doctor of Technical Sciences, Associate Professor
Department of Aerohydrodynamics*
Stanislav Kravchenko
Corresponding author
PhD
Department of Aerohydrodynamics*
E-mail: ksatfer@gmail.com
Oleksii Mykhailychenko
PhD Student
Department of Electrical Machines
National Technical University "Kharkiv Polytechnic Institute"
Kyrpychova str., 2, Kharkiv, Ukraine, 61002
Viacheslav Nazarenko
PhD Student**
Serhii Smyk
PhD
Scientific Center Air Force
Ivan Kozhedub Kharkiv National Air Force University
Poltavskyi Shliakh str., 123, Kharkiv, Ukraine, 61123
Oleksandr Vasyliev
PhD Student
Department of Aircraft and Helicopter Design*
Mariia Arefieva
PhD Student
Private Institution of General Secondary Education
"Kharkiv Lyceum "IT STEP SCHOOL Kharkiv"
Maika Yohansena str., 27, Kharkiv, Ukraine, 61024
Iryna Tretiak
PhD Student**
Serhii Serhiienko
PhD Student**
Volodymyr Selevko
PhD, Associate Professor
Department of Postgraduate and Doctoral Studies*
*National Aerospace University "Kharkiv Aviation Institute"
Vadyma Manka str., 17, Kharkiv, Ukraine, 61070
**Limited Liability Company "Kharkov Electric Machine-Building Plant"
Heroiv Kharkova ave., 199, Kharkiv, Ukraine, 61037

The object of this study is the thermal and hydraulic characteristics of the oil coolers in a thrust bearing and guide bearing cooling system of hydrogenerators.

The study addresses the task to improve the efficiency of the cooling system of the supporting elements of hydrogenerators while maintaining mass and dimension indicators. The main problems in this case are the unbalanced flow of oil within the typical cooling system and different stiffness of the thrust bearing segment supports.

The main result is the design of a cooling system structure that ensures effective heat removal from the oil. According to calculations, an oil cooler for a hydrogenerator with a capacity of 56 MW provides removal of 250 kW of losses with a heat transfer margin of 26.2%, for a hydrogenerator with a capacity of 96 MW – 150 kW and 31.8%, respectively.

The results are explained by the use of new antifriction materials for supporting elements (fluoroplastic-4) and a higher strength group material for the thrust bearing disk. The designed structure of the cooling system enables operation with preservation of isotropy of cooling liquids and does not have "dead" zones of these liquids flow.

A feature of the proposed method for calculating oil coolers is the addition to existing criterion equations of an experimental coefficient B, which takes into account the peculiarity of the geometry of the structure in addition to the nature of the oil flow and gas-dynamic parameters. The action of three-dimensional forces, temperatures, and pressures within the base metal and cooling liquids were comprehensively taken into account.

The proposed structures of the thrust bearing and guide bearing, as well as their cooling systems, could be implemented in the design and modernization of medium and high-power hydrogenerators

Keywords: hydrogenerator, single-row thrust bearing, oil cooler, hydraulic calculations, thermal calculations

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1. Introduction

The basic method of modern design of new structures of units and structural elements in mechanical engineering is three-dimensional modeling. In this case, current computing capabilities make it possible to take into account the action of various factors, the joint effect of which may have a significant impact on the design parameters [1–3]. Taking these factors into account when developing new components of hydrogenerators makes it possible to more accurately determine the loads to which these components will be subjected, as well as the technical parameters [4]. This makes it possible to optimize their structure and opens up additional opportunities to ensure the required reliability while reducing the weight and dimensions of the equipment.

A similar approach can also be used in the process of performing work on the restoration and modernization of existing electromechanical equipment, in particular during the reconstruction of generator equipment at hydroelectric power plants. In this case, three-dimensional modeling makes it possible to identify the shortcomings of existing structures and assess the effectiveness of technical solutions proposed in the reconstruction project.

The most loaded units during the operation of hydrogenerators are the thrust bearing and guide bearings. The thrust bearing is designed to absorb the entire load from the mass of the rotor and turbine, which can exceed 400 tons, as well as the hydraulic force from the water reaction. The guide bearings absorb the radial forces acting on the rotor and also ensure the vertical position of the shaft. Uneven distribution of the load on the thrust bearing segments can cause a significant increase in its temperature and vibration of the entire hydraulic unit. The cooling system of the thrust bearing and guide bearings consists of several oil coolers and an oil seal, which prevents the formation of oil mist during the evaporation of oil from the oil bath [5]. It should be taken into account that an increase in the temperature of the cooling oil leads to a change in its properties and contributes to an increase in the vibration of the hydraulic unit during its operation.

Scientific justification of increasing the efficiency and ensuring the necessary reliability of new structures of the above-mentioned units with a refined assessment of heat exchange processes at the stage of their design is an outstanding scientific task that is of great practical value.

Devising a new method for calculating thermal and hydraulic loads with the addition of a larger number of factors affecting the operation of structural components of hydrogenerators could make it possible to determine the optimal design parameters while maintaining their reliability.

Therefore, it is a relevant scientific and practical task to carry out studies on designing improved structures of the thrust bearing and guide bearing, as well as their cooling system, which could simultaneously allow for reliable operation, ease of maintenance, and high maintainability.

2. Literature review and problem statement

Existing structures of hydrogenerators use classical schemes of supporting elements and their cooling systems. For modern high-power hydrogenerators, double-row thrust bearings on hydraulic supports are quite often applied [6]. Although such a design provides advantages in terms of load distribution on segments, its disadvantage is complexity, which is due to a significant number of elements and the presence of a corrugated oil chamber.

The method proposed in work [7] makes it possible to optimize structures by increasing their rigidity. The disadvantage of the work is that the components of forces in general are not considered. The solution to this problem is provided in [8, 9], in which mechanical factors are analyzed in detail. It is also necessary to take into account that thermal components are significant factors when choosing existing deformations. The disadvantage of the structures considered in those works is the presence of detachable connections using one group of pins. This leads to the occurrence of eccentric loads. In accordance with the technology, for ease of assembly, clamps with a relatively long geometry are used, for which there is a possible loss of stability and, as a result, the occurrence of uneven distortions. In the proposed study, this problem is planned to be solved by optimizing thrust bearing through the use of a structure with equally rigid elements.

In works [10, 11] solutions to the problem of optimizing the design of air coolers of hydrogenerators are reported. However, given that the liquid is only a heat carrier and does not participate in the operation of the entire system, the proposed equations are unacceptable for calculating the cooling system of thrust bearings. The reason for this is that the oil, which is the main coolant of the cooling system of the hydrogenerator thrust bearing, is also used to create hydraulic force. This is the so-called oil wedge, which contributes to the suction of oil between the bearing surfaces during the operation of the unit and forms an anti-friction layer). It should also be noted that the rigidity and viscosity of the oil depend on its temperature (the operating range of the oil temperature in the thrust bearings of existing hydrogenerators is approximately 15–65°C). Going beyond the permissible range, as a rule, leads to an increase in the vibration of the hydraulic unit. In the proposed study, this problem is planned to be solved by maintaining the thermal state at the same level. This would eliminate the compressibility factor of coolants.

Criterion equations are usually used to solve the heat transfer problem. In [12], the problem of optimizing the internal structure of a semi-annular cooler of a turbine guide bearing at a hydroelectric power plant is considered, and a numerical calculation is also performed to study the heat transfer characteristics under several operating conditions. The results show that the spiral-twisted flat tubular cooler proposed in the work effectively reduces the risk of clogging of the heat exchange tube and significantly increases the heat transfer efficiency of the cooler. The advantage of the study is the use of CFD methods in a three-dimensional statement. The disadvantage is that at the initial stage of design, the use of the proposed algorithm is impractical since the main geometric solutions have not yet been accepted. The problem is planned to be solved by using refined criterion equations with the introduction of a geometry accounting coefficient.

This method was further developed in the calculation of air coolers for hydrogen generators at the Aswan Dam (Egypt) [11]. However, a significant drawback of the development of the method in this direction was the locality of its application due to the peculiarities of the considered design and the difficulty of transferring it to calculations of other types of design designs of coolers. In order to supplement the above, the materials reported in [13] could be used, in which thermal and mechanical factors are studied comprehensively in detail.

In work [14], a method for numerical study of the heat transfer mechanism in the cooling system of a hydrogen generator is presented. This method involves solving three separate problems: ensuring rigidity, maintaining the thermal state within the permissible limits, and creating the necessary drops in hydraulic systems. Its disadvantage is that the problems are solved as unrelated, and the thermal calculation does not take into account the compressibility of the liquid. Also, the change in its characteristics under the influence of pressure is not taken into account. Deformations caused by mechanical forces, which often cause increased vibration during the operation of hydraulic units, are not comparable to thermal ones.

A further solution to the heat transfer problem consists in determining the nature of heat transfer and convective heat exchange of the contacting parts of the thrust bearing of the hydrogenerator in order to take into account all critically important factors in the calculations.

Inclined segment thrust bearings are key components in hydraulic units. The changed thrust bearing angle makes it possible to optimize the load and, as a result, reduce the influence of thermal factors. However, as axial loads increase with increasing hydroturbine power, many thrust bearings are at risk of failure due to friction between the collar and the pads. To repair damaged pads, some installations have to be stopped for a period of several weeks to months, which leads to huge economic losses.

The processes of thrust bearing lubrication are quite complex and include fluid-thermostructural interactions between the pad, collar, oil film, and oil surrounding the pad. In [15], the bearing operation processes are considered, including thermoelastic deformation, transient and dynamic characteristics, features of the operation of a bidirectional thrust bearing, as well as various methods for predicting lubrication efficiency. The paper also gives an overview of some special designs, including material coatings, a support system, and a hydrostatic lift system to improve axial lubrication. However, a significant drawback of the structures considered in [15] is the very high complexity of operation. One of the reasons for this is the presence of areas with "dead" zones (zones that include close to zero coolant flow rates). The presence of these areas in existing structures is due to the lack of deflectors that change the flow vector and optimize the trajectory of the fluid. This drawback is planned to be prevented by using a rigid design of a single-row thrust bearing and a cooling system that excludes these zones.

3. The aim and objectives of the study

The purpose of our research is to devise a new method for calculating the structure of an improved cooling system for the thrust bearing and guide bearings in medium and high-power hydrogenerators during their design and reconstruction.

To achieve the goal, the following tasks were set:

– to design a rigid structure of a single-row thrust bearing and guide bearing, which makes it possible to work with high contact stresses and a reduced level of vibration;

– to design a new cooling system for the thrust bearing and guide bearings, which combines operation under different modes using new anti-friction materials;

– to conduct thermal and hydraulic calculations of the designed structures of oil coolers for hydraulic units of various capacities to confirm their normal operation within the resource requirements;

– to compile recommendations for the maintenance of thrust bearings and guide bearings of hydraulic units to ensure reliable operation.

4. The study materials and methods

The object of our study is the thrust bearing in a hydrogenerator, guide bearings, a cooling system of the thrust bearing and guide bearings, as well as their structural elements.

The principal scientific hypothesis of our work is the principle of the combined action of pressure forces, thermal factors, and the physical influence of different parameters under the conditions of maintaining the equal strength of the structure acting on the common areas of the bearing surfaces. At the same time, it was also necessary to ensure a minimum temperature gradient within one unit. Moreover, not only the geometric different stiffness of the structure was taken into account but also the use of different elastic moduli for materials. The purpose of setting these conditions was to exclude critical tipping moments and ensure structural stability in order to enable reliable operation of the thrust bearing assembly design.

To scientifically substantiate the effectiveness of the proposed cooling system design using new materials in the structure of the thrust bearing and bearings, it is necessary to calculate the heat exchange processes of oil coolers. This will make it possible to compile recommendations for designing hydrogenerators in order to improve their reliability.

The cooling system of the thrust bearing and guide bearings of the vertical type hydrogenerator has a common oil flow circuit.

The disadvantage of a typical structure of the cooling system of thrust bearing and guide bearings is the unbalanced flow of oil within this system. The use of classical designs implies different stiffness of the supports and, as a result, additional deformations along the thrust bearing segment. This leads to the ellipticity of the structure, which results in a significant increase in the vibration of the hydraulic unit and the temperature of the thrust bearing assembly elements.

The construction of a new cooling system for thrust bearing and guide bearings of a reversible type hydrogenerator was carried out using three-dimensional design methods. To determine the heat transfer reserve, thermal and hydraulic calculations of oil coolers for hydraulic units of various capacities (up to 300 MW) were carried out.

Based on existing methods of thermal and hydraulic calculations [16–18], in order to eliminate calculation errors, criterion equations were used. These equations take into account the change in oil flow regimes from laminar to turbulent, as well as the viscosity of the fluid when pressure and temperature change. The main feature of the proposed method was the addition to the method specified in [8] of the influence of the physical properties of the heat carrier on heat transfer.

At the first stage of the study, the task of designing a new structure of the assembly (thrust bearing, guide bearing, cooling system) is solved by initially drawing up a draft design. The second stage is to conduct preliminary analytical calculations in terms of heat and strength. At the third stage, the dimensions are refined, and three-dimensional models of the nodes and elements of the assembly are built, within which the tasks of operation, manufacturability, and maintainability are addressed.

As an area of further research, it is planned to conduct a full-fledged three-dimensional analysis of the assembly, on the basis of which new materials are selected and the uniform strength and rigidity of the structure and the working design of the assembly as a whole are ensured.

5. Results of research on the proposed structures of thrust bearing and guide bearing in a hydrogenerator

5. 1. Proposed structures of the thrust bearing and guide bearing

Proposed structure of the thrust bearing.

The basic technical parameters of the hydrogenerator, which are necessary for calculations reported in our study, are given in Table 1.

Table 1

Basic technical parameters of the hydrogenerator

Parameter ID	Value
Type of hydrogenerator	reverse
Speed, rpm	600
Number of thrust bearing segments, units	12
Internal diameter of the working surface of the segments, m	0.87
External diameter of the working surface of the segments, m	1.33
Radial eccentricity, mm	10
Tangential eccentricity, mm	0

In our study, a new structure of a single-row thrust bearing on a rigid support is proposed.

As a condition for calculating the stress-strain state of the thrust bearing structures, guide bearings, and their cooling system, the uniformity of the action of mechanical and thermal forces exerted by fluids within the system was assumed.

The thrust bearing of the presented design has twelve self-aligning segments, supported through elastic disk supports on the heads of the support bolts. The elastic deformation of the disk supports compensates for the uneven load distribution between the individual segments and the disk runout during rotor rotation (Fig. 1).

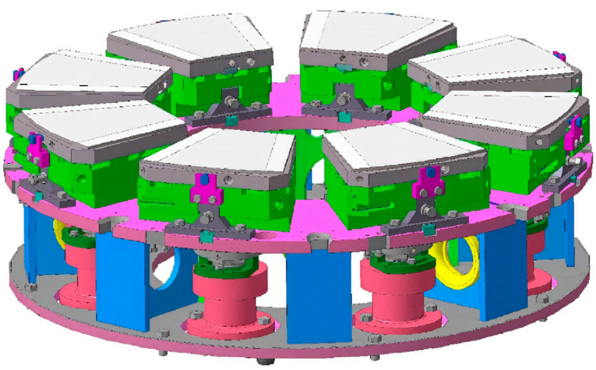

Fig. 1. Three-dimensional model of a single-row thrust bearing on a rigid support

The three-dimensional model of the new thrust bearing disk is shown in Fig. 2.

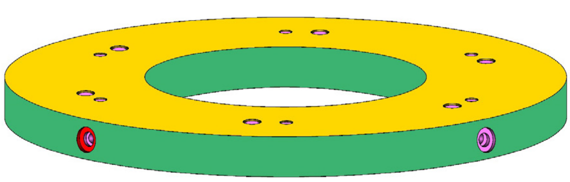

Fig. 2. Three-dimensional model of the new thrust bearing disk

The disk has an increased mirror surface area, which ensures optimal load distribution from the weight of the rotor and turbine and leads to a decrease in the temperature of the functional elements of the thrust bearing elements. The disk is insulated to prevent the flow of bearing currents.

The disk material is forging with a hardness of HB 143-179 (strength group KP 490 with a yield strength of 490 MPa). The mirror surface of the disk has a roughness of Ra 0.32 μm. The use of forgings with a yield strength of 245 MPa in existing structures of thrust bearing disks [19] led to the emergence of surface defects, which was associated with a low yield strength and low hardness of the material. The transition to a higher strength group made it possible to completely eliminate the growth of contact defects due to a higher fatigue limit.

Fig. 3 shows three-dimensional models of the thrust bearing assembly parts.

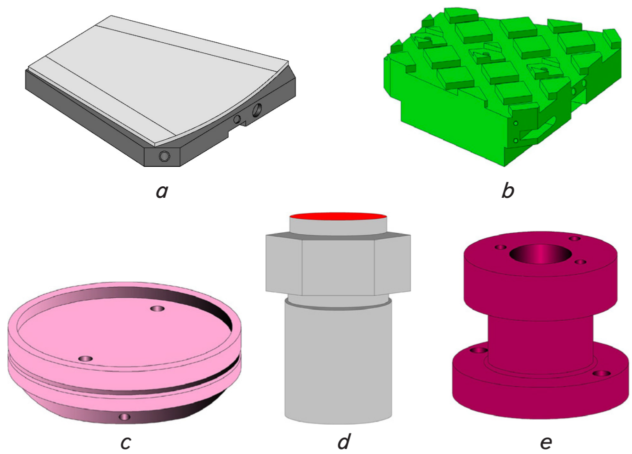

Fig. 3. Three-dimensional models of the thrust bearing assembly parts: *a* — thrust bearing segment; *b* — trapezoid; *c* — elastic disk support; *d* — support bolt; *e* — bushing

The friction surface of the segments is covered with an elastic antifriction layer, the basis of which is fluoroplastic-4 with a thickness of 2.6 ÷ 2.9 mm (Fig. 3, *a*). The thrust bearing segments have western bevels of a certain configuration, which create a virtual tangential eccentricity and ensure optimal supply of cooled oil to the gap between the disk and segments. The segments have holes for installing temperature sensors.

To ensure the necessary rigidity, the segment rests on a massive trapezoid, which has channels for the circulation of cooled oil (Fig. 3, *b*). The arrangement of the channels enables normal circulation of oil when the rotor rotates in two directions. The trapezoid rests on an elastic plate-shaped support, which perceives vertical vibration, and also compensates for uneven load distribution between the segments (Fig. 3, *c*).

The elastic plate-shaped support is made of forging with a yield strength of $Rs \geq 785$ MPa and a hardness of HB 293-331.

In the lower part of the elastic plate-shaped support there is a flat surface, which rests on the support bolt (Fig. 3, *d*). The support surface of the bolt has the shape of a sphere. The hardness of the support surface of the bolt is greater than the hardness of the elastic plate-shaped support to ensure the absence of crushing of the support surfaces in the contact spot. The bolt has a fine thread to ensure accuracy of adjustment. The support bolt is installed in a sleeve that is attached to the thrust bearing housing (Fig. 3, *e*).

The thrust bearing operates on self-lubricating. When the disk rotates, oil is sucked between the disk and segments and forms an oil film that separates the friction surfaces. The movement of the segment on the support is limited by radial and tangential stops.

The oil is cooled by U-shaped coolers built into the oil bath of the thrust bearing. The layout of the oil bath enables such oil circulation that all hot oil passes through the coolers.

The basic geometric dimensions of thrust bearing and its operational characteristics, which determine its performance, are given in Table 2.

Table 2

Basic geometric dimensions of the thrust bearing and operational characteristics

Parameter ID	Value			
Hydrogenerator operating mode	Turbine	Pumping	Synchronous compensator	Transitional
Maximum load on the thrust bearing, t	130	140	180	220
Minimum oil film thickness, mm	0.0756	0.0716	0.0595	0.0517
Maximum segment temperature, °C	56	57	62	67
Specific pressure on the segment at nominal speed, kg/cm^2	26 (2.6)	28 (2.8)	36 (3.6)	44 (4.4)
Total friction losses, kW	174	177	185	193
Operating time of the thrust bearing without cooling water at nominal speed, min	8.6	8.5	8.2	8.0

The basic design parameters of the thrust bearing ensure the operability of the hydraulic unit under all operating modes.

The thrust bearing is designed to operate with TP-30 or REMITZ TU 46 LOTOS ISO 6743-5:2006 oil, which is widely and successfully used in modern thrust bearings and bearings in hydrogenerators and turbines. The basic technical parameters of the oils used in cooling systems are given in Table 3.

Table 3

Basic technical parameters of oils used in cooling systems

Parameter ID	Value	
	TP-30	REMITZ TU 46 LOTOS
Kinematic viscosity at 40°C, mm^2/s	41.1–50.6	45
Viscosity index, not less than	95	96
Pour point, °C	–10	–15
Flash point, °C	190	235

The proposed structure of the thrust bearing in a reversible hydraulic unit has the following advantages over the typical one:

– rejection of the use of a high-pressure oil supply system (hydraulic lift) since the antifriction fluoroplastic coating has a low coefficient of friction and high anti-seize properties. This makes it possible to extend service life due to reduced segment aging even after a long generator shutdown;

– elimination of thermal deformation ("para-saltiness") of the segment by making it two-layer. A relatively thin segment is located on a massive base that has channels for the circulation of cooled oil in the thrust bearing bath. This reduces the temperature gradient along the height of the base and segment, and therefore the "para-saltiness" of the segment;

– increase in the maximum permissible operating temperature and average specific load of segments with a fluoroplastic coating during operation compared to a babbitt coating;

– mechanical braking of the rotor at lower speeds, which significantly facilitates the operation of the hydraulic unit brake system;

– improved mixing of hot oil at the segment outlet with cooled oil in the thrust bearing bath, due to sufficient distance between segments. This reduces the influence of hot oil draining from a segment on the oil temperature at the inlet of the next segment.

Proposed design of a guide bearing.

In our study, a new structure of a guide bearing is proposed.

The design of a guide bearing must have the ability to accept longitudinal loads. The structural model of the upper and lower guide bearings with twelve segments that are self-aligning and cover the bearing sleeve is shown in Fig. 4.

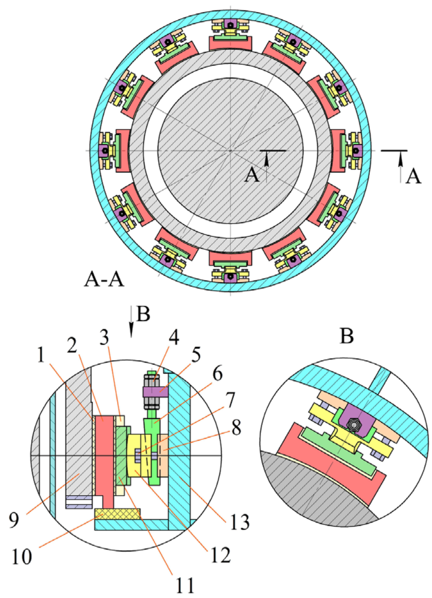

Fig. 4. Guide bearing: 1 — fluoroplastic layer of the bearing segment; 2 — bearing segment; 3 — insulating plate; 4 — nut; 5 — wedge holder; 6 — adjusting wedge; 7 — bolt; 8 — external support; 9 — bearing sleeve; 10 — insulating support shelf; 11 — rusk; 12 — support plate; 13 — support ring of the cross (bearing housing)

Each segment 2 rests on the spherical surface of support plate 12. Support plate 12 rests on the cross body through adjusting wedge 6. This design provides the possibility of convenient adjustment of the gap by moving the adjusting wedge.

Rusk 11 is isolated from segment 2 by special insulating plates 3 that protect the surface of the segments from bearing currents. Segment 2 rests on insulating support shelf 10, which is attached to the bearing housing. The shelf is also insulated from bearing currents.

Increased rigidity of the bearings is ensured by direct support of the segments on the cross body without the use of removable housings.

The segment guide bearing operates on self-lubrication. In this case, each segment is immersed in oil by 1/2–1/3 of its height. Due to the pressure developed in the oil wedge between the segment and the bushing, the oil spreads over the entire surface of the segment and provides lubrication of the bearing friction surface. For forced circulation of the oil in the bearing bath, radial holes are made in the bushing, which work as centrifugal pumps.

The working surface of the guide bearing segments is a composite material consisting of two layers: antifriction and support. The antifriction layer 2.6 ÷ 2.9 mm thick is made of elastic material – fluoroplastic-4. The support elastic layer is made with a height of 6 ÷ 7 mm.

The fluoroplastic coating has the following advantages:

– high performance under all operating modes (including start-ups) due to the elastic vibration-damping coating with high anti-burr properties and a low friction coefficient. Segments with a fluoroplastic coating can operate for a long time with insufficient lubrication of the friction surfaces (violation of liquid friction);

– no need to lift the rotor on the brakes after a long downtime;

– the possibility of braking the rotor at lower speeds, which greatly facilitates the operation of the brake system, or stopping the unit without braking (on free run);

– if one segment is damaged, the rest remains intact;

– simplification of operating conditions. Elimination of grinding and scraping of the segment surface during installation and repair of the thrust bearing;

– the ability to rotate the rotor of the unit without lubricating the working surfaces, i.e., without opening the thrust bearing oil bath and the rejection of forced lubrication of the segment friction surface during starts and stops;

– the ability to operate the thrust bearing with a higher oil temperature, which makes it possible to maintain the operational reliability of the thrust bearing for a long time with a reduced number of oil coolers;

– ensuring the isolation of the thrust bearing disk from the segments.

The basic parameters of the fluoroplastic coating (elastic metal-plastic layer) are given in Table 4.

The basic geometric dimensions of guide bearings and their operational characteristics are given in Table 5.

Based on the values of the working specific load (Table 4) and the maximum thickness of the oil film (Table 5), taking into account the total friction losses, the following conclusion can be drawn. For a maximum temperature of the guide bearing segment of 51°C, the oil cooler has a fairly high efficiency. A small difference between the maximum temperatures of the upper and lower guide bearing segments of 3°C eliminates the dependence of change in oil parameters on its actual temperature during operation of the hydraulic unit.

Table 4

Basic parameters of fluoroplastic coating

Parameter ID	Value
Permissible working specific load, MPa	6.5
Maximum specific load, MPa	12.0
Permissible operating temperature of the antifriction layer, °C	90
Thermal stability of the antifriction layer, °C	260
Maximum sliding speed, m/s	80
Friction coefficient without oil in a pair with steel	0.05 ÷ 0.08
Shaft misalignment compensation, μm	40 ÷ 60
Average wear of the friction surface at 100,000 hours of operation, mm	0.3
Estimated service life of the elastic layer before the onset of fatigue manifestations, years	20

Table 5

Basic geometric dimensions of guide bearings and performance characteristics

Parameter name	Value	
Bearing type	top	bottom
Bearing sleeve bore diameter, m	1.1	0.96
Number of segments, pcs.	12	
Radial gap, mm	0.2	
Segment height (width), m	0.15	
Segment length, m	0.2	
Minimum oil film thickness, mm	0.0451	0.0345
Maximum segment temperature, °C	49	51
Total friction losses, kW	95	67

5. 2. Cooling systems for thrust bearing and guide bearings

Oil cooling is carried out by U-shaped coolers built into the oil baths of the thrust bearing and lower guide bearing. The layout of the oil bath ensures such oil circulation that all the hot oil passes through the coolers (Fig. 5).

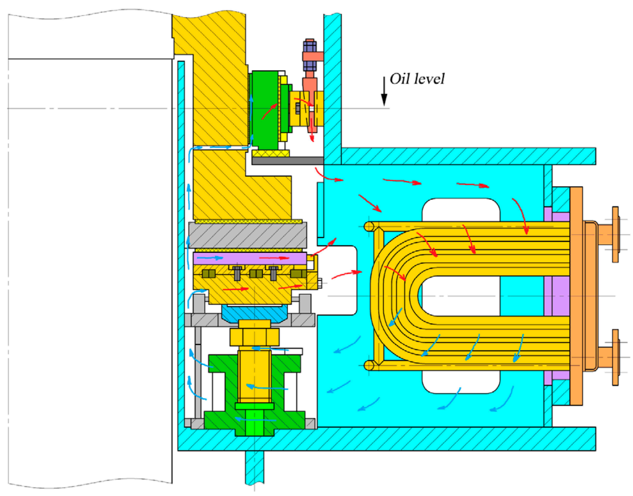

Fig. 5. Oil circulation diagram

Modernization of the cooling system for thrust bearing and bearings does not require changing the pressure and flow of cooling water, which are currently available on existing hydrogenerators.

6 oil coolers are used to cool the thrust bearing and the upper guide bearing of the generator. 3 coolers are used to cool the lower guide bearing of the generator. The water coolers are of the cassette type (Fig. 6). The tubes of the oil coolers are made of cupronickel (CuNi), which has high thermal conductivity and corrosion resistance. The tube sheets and pipelines for connecting the coolers are made of stainless steel.

To prevent oil vapors from escaping through the seals of the oil pans of the thrust bearing and bearings, they are drained into the drainage through a special pipeline. The oil exits through these pipes in the form of a small amount of vapors or individual condensed drops. An additional obstacle to the escape of oil vapors from the oil pans is created by supplying air to the seals of the oil pans through special pipes from the high-pressure zone of the generator. The seals located in the low-pressure zone of the generator are connected to the atmospheric pressure zone (Fig. 7).

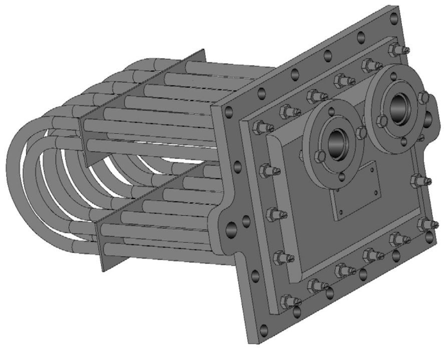

Fig. 6. Three-dimensional model of oil cooler

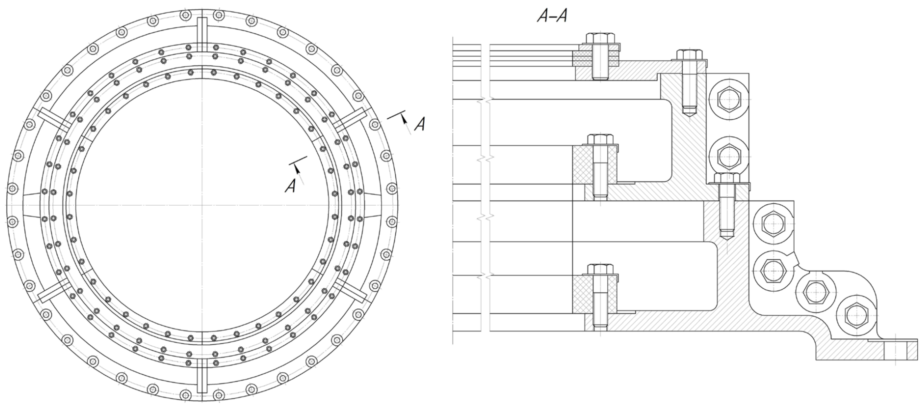

Fig. 7. Oil seal

To extract oil vapors, a system is provided, consisting of an exhaust centrifugal fan, an oil vapor separator, and a fan control cabinet. The fan is turned on simultaneously with the start of the hydrogenerator-engine and is turned off with a delay of 5–10 minutes after the hydraulic unit stops.

A drainage pipeline is provided for draining oil from the lower seals of the upper and lower crosspieces. Oil is pumped out of the lower seal of the upper crosspiece automatically by means of a pump into a container for "dirty" oil. Oil is drained from the lower seal of the lower crosspiece by gravity into the oil drainpipe from the oil bath of the lower guide bearing.

5. 3. Thermal and hydraulic calculations of the designed structures of oil coolers for hydraulic units of different capacities

In order to determine the thermal and hydraulic characteristics of the oil cooler of the developed geometry, thermal and hydraulic calculations of oil coolers for hydraulic units of different capacities were carried out. Calculation of oil coolers of thrust bearings for reversible type hydrogenerators with a capacity of 56 MW and 96 MW.

The initial data for the calculation of oil coolers of thrust bearings of reversible type hydrogenerators with a capacity of 56 MW and 96 MW are given in Table 6.

The sequence of performing the thermal calculation: the oil cooling surface of one section of the oil cooler F_M, m^2, is determined from the formula

$$F_M = \pi d_{out} l_a n_t. \tag{1}$$

The cross-sectional area of the pipes for one water pass f_W, m^2

$$f_W = \frac{n_t}{q_W} \cdot \frac{\pi \cdot d_{in}^2}{4}. \tag{2}$$

Water overheating in the parallel branch Δt_W, °C

$$\Delta t_W = \frac{N}{C_{m_w} \rho_W Q_W}. \tag{3}$$

Average water temperature taking into account overheating t_{Wav}, °C

$$t_{Wav} = t_{W1} + \frac{\Delta t_W}{2}. \tag{4}$$

The water velocity in the tubes ω_W, m/s, is determined from the formula

$$\omega_W = \frac{Q_W}{f_W n_W}. \tag{5}$$

The Reynolds criterion for water Re_W is determined from the following formula

$$\mathrm{Re}_W = \frac{\omega_W \cdot d_{in}^2}{\nu_W}. \tag{6}$$

The Reynolds criterion for oil Re_a is determined from the following formula

$$\mathrm{Re}_a = \frac{\omega_a \cdot d_{out}^2}{\nu_a}, \tag{7}$$

where $\omega_a = 1.0$ m/s – oil velocity at the location of the cooler tubes.

Table 6

Initial data for calculating oil coolers for hydrogenerator thrust bearings

Parameter ID	Value	
Active power of the hydrogenerator P, MW	56	96
Discharged losses N, kW	250	150
Outer diameter of the tube d_{out}, m	0.019	0.019
Inner diameter of the tube d_{in}, m	0.017	0.017
Tube material	Cupronickel	Cupronickel
Number of sections n_s, pcs.	12	12
Number of parallel branches n_w, pcs.	6	2
Number of consecutive branches n_{w1}, pcs.	2	6
Number of water passes in the section q_w, pcs.	4	3
Number of oil passes q_a, pcs.	1	1
Number of strings of tubes along oil pass n_a, pcs.	34	32
Number of tubes in a row m_a, pcs.	4	3
Number of tubes in a section n_t, pcs.	136	96
Cold oil temperature t_a, °C	45	40
Cold water temperature t_{w1}, °C	32	32
Water heat capacity C_{mw}, kJ/(kg°C)	4.174	4.174
Water density ρ_w, kg/m^3	995	995
Average tube length along oil pass l_a, m	0.611	0.856
Water consumption Q_w, m^3/g	80	60

Table 7

Physical parameters of water at average temperature

Parameter ID	Value	
Active power of the hydrogenerator *P*, MW	56	96
Density ρ_{w1}, kg/m^3	994	994
Thermal conductivity coefficient λ_w, kW/(m·°C)	$0.624 \cdot 10^{-3}$	$0.618 \cdot 10^{-3}$
Kinematic density ν_w, m^2/s	$0.729 \cdot 10^{-6}$	$0.729 \cdot 10^{-6}$
Prandtl number Pr_w	4.85	4.85

Table 8

Physical parameters of oil at a given temperature

Parameter ID	Value	
Active power of the hydrogenerator *P*, MW	56	96
Density ρ_a, kg/m^3	892	895
Oil heat capacity C_{ma}, kJ/(kg·°C);	1.907	1.907
Thermal conductivity coefficient λ_a, kW/(m·°C)	$0.127 \cdot 10^{-3}$	$0.127 \cdot 10^{-3}$
Kinematic density ν_a, m^2/s	$0.41 \cdot 10^{-4}$	$0.41 \cdot 10^{-4}$
Prandtl number Pr_a	531	531

The heat transfer coefficient from the oil to the tube wall α_a, kW/(m^2·°C), is determined from the following formula

$$\alpha_a = B\,\mathrm{Re}_a^{0.5}\,\mathrm{Pr}_a^{0.333}\,\frac{\lambda_a}{d_{out}}, \tag{8}$$

where $B = 0.543$ is the generalized experimental coefficient.

The heat transfer coefficient from the tube wall to the water α_W, kW/(m^2·°C), is determined from the following formula

$$\alpha_W = 0.023\,\mathrm{Re}_W^{0.8}\,\mathrm{Pr}_W^{0.4}\,\frac{\lambda_W}{d_{in}}. \tag{9}$$

Heat transfer coefficient from oil to water *K*, kW/(m^2·°C)

$$K = \frac{1}{\dfrac{1}{\alpha_a} + \dfrac{d_{out}}{2\lambda}\ln\dfrac{d_{out}}{d_{in}} + \dfrac{d_{out}}{d_{in}\alpha_W}}, \tag{10}$$

where $\lambda = 0.0372$ kW/(m·°C) is the thermal conductivity coefficient of cupric chloride.

The mean logarithmic heat pressure between oil and water Δt, °C, is determined from the following formula

$$\Delta t = \frac{\varphi\left(\Delta t_l - \Delta t_S\right)}{\ln\dfrac{\Delta t_l}{\Delta t_S}}, \tag{11}$$

where $\varphi = 0.9$ is the coefficient that takes into account the nature of the heat carrier movement; $\Delta t_l = t_a - t_{w1}$ – larger temperature difference for counterflow; $\Delta t_s = t_a - t_{w2}$ – smaller temperature difference for counterflow; $t_{w2} = t_{w1} + \Delta t_w$ – water temperature at the outlet of the parallel branch of the oil cooler.

The apparent cooling surface F_p, m^2, is determined from the following formula

$$F_p = F_M n_S. \tag{12}$$

Required cooling surface, taking into account 20% reserve, F_{req}, m^2

$$F_{req} = \frac{N}{k_3 \Delta t K}. \tag{13}$$

Heat load of cooler K_1, kW/(m^2·°C)

$$K_1 = \frac{N}{F_p \Delta t}. \tag{14}$$

Heat transfer reserve *M*, %

$$M = \frac{K - K_1}{K}100. \tag{15}$$

Sequence of hydraulic calculation.

The coefficient of hydraulic friction for a cupronickel tube with a diameter of 17 mm at equivalent roughness ($k_E = 0.1$ mm) λ is determined from the following formula

$$\lambda = 0.11\left(1.46\frac{k_E}{d_{in}} + \frac{100}{\mathrm{Re}_W}\right)^{0.25}. \tag{16}$$

Water pressure drop in one section of the cooler ΔH_{W1}, Pa

$$\Delta H_{W1} = \left(\lambda\frac{l_a}{d_{in}} + \sum\xi\right)\frac{\rho_W \omega_W^2}{2} q_W, \tag{17}$$

where $\sum\xi$ is the sum of the coefficients of local resistances in the oil cooler.

The water pressure drop in the parallel branch when connecting the coolers in series ΔH_W, Pa, is determined from the following formula

$$\Delta H_W = n_{W1} \cdot \Delta H_{W1}. \tag{18}$$

The results of thermal and hydraulic calculations of oil coolers of thrust bearings in reversible hydrogenerators with a capacity of 56 MW and 96 MW are given in Table 9.

Table 9

Results of thermal and hydraulic calculations of thrust bearing oil coolers

Parameter ID	Value	
Active power of the hydrogenerator P, MW	56	96
Thermal calculation		
Oil cooling surface of one section of oil cooler F_M, m^2	4.96	4.91
The area of the passage section of the tubes along one water pass f_W, m^2	0.00772	0.00726
Overheating of water in a parallel branch Δt_w, °C	2.7	2.2
Average water temperature, taking into account overheating t_{Wav}, °C	33.4	33.1
Water velocity in tubes ω_w, m/s	0.48	1.15
Reynolds criterion for water Re_w	11191	26755
Reynolds criterion for oil Re_a	463	463
Heat transfer coefficient from oil to tube wall α_a, kW/(m$^2\cdot$°C)	0.631	0.631
Heat transfer coefficient from tube wall to water α_w, kW/(m$^2\cdot$°C)	2.75	5.48
Oil-to-water heat transfer coefficient K, kW/(m$^2\cdot$°C)	0.496	0.551
Greater temperature difference for countercurrent Δt_l, °C	13	8
Smaller temperature difference for countercurrent Δt_s, °C	10.29	5.8
Water temperature at the outlet of the parallel branch of the oil cooler t_{w2}, °C	34.7	34.2
Mean logarithmic thermal pressure between oil and water Δt, °C	11.48	6.79
Apparent cooling surface F_p, m^2	59.5	58.9
Required cooling surface taking into account 20% margin F_{req}, m^2	54.95	50.15
Heat load of the cooler K_1, kW/(m$^2\cdot$°C)	0.366	0.375
Heat transfer reserve M, %:	26.2	31.8
Hydraulic calculation		
Coefficient of hydraulic friction for cupronickel tube at equivalent roughness (k_E = 0.1 mm) λ	0.04	0.037
Water pressure drop in one section of the cooler ΔH_{w1}, Pa	5237	18341
Water pressure drop in a parallel branch ΔH_w, Pa	10475	110050

Our calculations of the oil cooler of the thrust bearing of a reversible type hydrogenerator with a capacity of 56 MW showed that when creating a pressure of more than 10 kPa for the oil-water cooling medium, the designed cooling system provides a normal thermal mode of operation of the thrust bearing. The nature of the heat exchange in this case will be characterized by supercritical Reynolds numbers, which is characteristic only for hydrogenerators and is not used anywhere else in mechanical engineering.

The calculations showed that the proposed criterion equations could be extended to hydrogenerators not only of medium but also of high power, which are characterized by a stator diameter of more than 8 m and a stator height of more than 2 m.

5. 4. Recommendations for adjusting and maintaining the thrust bearing and guide bearings

Recommendations for adjusting the thrust bearing.

The purpose of adjusting the thrust bearing is to evenly distribute the load between the segments, taking into account the stiffness of the elastic plate-shaped supports. To ensure the technological cycle and compliance with the parameters of displacements and deformations according to analytical calculations, it is necessary to perform operations in the following sequence:

a) installation of the thrust bearing – an assembly procedure performed on the basis of technological maps and accompanied by the design department to take into account structural innovations;

b) installation of a lever-indicator device on all twelve elastic plate-shaped supports (Fig. 8);

c) distribution of the load between the segments.

The implementation of points b) and c) is carried out simultaneously to further ensure the uniformity of vertical and horizontal loads.

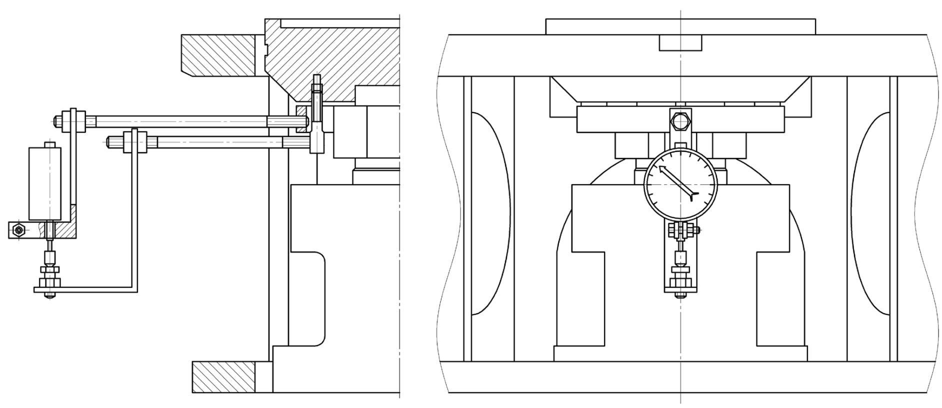

Fig. 8. Installing indicators for adjusting the thrust bearing

To reduce the action of eccentric forces, it is necessary to ensure uniform distribution of loads between the thrust bearing segments. To achieve this, it is proposed to perform technological operations in the following sequence:

1) with the rotor raised (on brake jacks), set the indicator scales (all twelve) to "zero";

2) lower the rotor onto the segments, record the indicator readings, and determine the average deviation;

3) compare the readings of each indicator with the average value of the readings recorded under the previous item. The maximum deviation should not exceed 3%;

4) raise the rotor on the brake jacks and redistribute the loads to the segments by adjusting the height position of the support bolts (to reduce the load, the bolt must be screwed in, and to increase it, it must be unscrewed from the support);

5) after adjusting the height position of the thrust bearing segments according to item 4, it is necessary to lower the rotor and perform repeated measurement according to items 2 and 3.

The cycle "lowering-calculation of deviations-raising-adjustment" is repeated until the deviation of the indicators when loading the segments is less than 3% of the average deviation value.

Usually, 4...5 cycles are required when adjusting the height position of the thrust bearing to achieve the required deviation value.

Recommendations for maintenance of the thrust bearing and guide bearings.

It should be emphasized that the thrust bearings are resource units, the technical condition of which directly affects the vibration and stationary thermal stress state of the entire hydraulic unit structure. The thrust bearing and guide bearings require systematic monitoring during operation. During operation of the hydraulic unit, the following parameters are subject to control:

– oil level in the thrust bearing and guide bearing baths;
– temperature of bearing segments, thrust bearing, and oil;
– cooling water flow rate;
– condition of thrust bearing and bearing insulation;
– clearance in guide bearings.

The normal oil level in the thrust bearing and bearing baths is specified in the technical documentation. However, it should be borne in mind that this level must be specified during the adjustment process. During operation of the unit, it is necessary to ensure that there is no decrease in the level due to oil leakage or increase due to water from coolers and pipelines entering the oil. An excess of the oil level can occur due to water leakage from coolers into the baths. If the water leakage occurs at the point where the tube is attached to the tube sheet, then flaring should be used. If the leakage comes from a damaged tube, it should be soldered or plugged.

The oil poured into the thrust bearing and bearing baths of the hydraulic unit must be tested at the following times:

– control – once a week, and in the first month of operation – twice a week;
– reduced analysis – once a year if the oil is completely transparent; if the oil becomes cloudy, an extraordinary analysis is performed.

Oil control consists of checking it for water, sludge, and mechanical impurities to determine the need for cleaning.

The scope of the abbreviated analysis includes determination of acid number, viscosity, presence of mechanical impurities and water, determination of the flash point of the oil.

Since the basic component of heat transfer and transmission of hydraulic loads from the weight of the rotor and turbine is oil, its quality directly affects the temperature and force gradients, as well as the vibration state of the hydraulic unit.

Operating oil must comply with the following standards:

– oil must not contain sludge, water, and mechanical impurities, must be transparent;
– the reaction of the water extract must be neutral;
– viscosity must be from 41.4–50.6 mm^2/s (for TP-30 oil);
– the drop in flash point must not exceed 10% of the initial one.

If one of the oil indicators deviates from the above standards and it is impossible to restore its quality without stopping the hydraulic unit, the oil must be replaced. The admissibility of mixing oil when pouring more than 5% must be confirmed by laboratory analysis for sedimentation and stability of oil properties after mixing.

The thrust bearing makes it possible to start the hydraulic unit at a cooling oil temperature of not less than 10°C. The maximum permissible temperature of hot oil in the thrust bearing and guide bearing bath is 65°C. If the oil temperature is exceeded, it is necessary to increase the water flow through the oil coolers to achieve the operating value. The setpoint for "shutdown" by the temperature of hot oil should not be more than plus 65°C, "warning" – no more than 60°C.

The maximum permissible temperature of the thrust bearing and guide bearing segments is 80°C. During the initial period of operation of the hydrogenerator-engine, it is necessary to determine the maximum level of the established temperature, which fluctuates depending on the operating mode of the unit, the temperature of the cooling water, and its flow rate. It is recommended to take the settings of secondary devices that control the temperatures of the thrust bearing and bearing segments and oil in the baths above the actually established temperatures by 5...8°C. The specified "warning" setpoint should not be higher than 75°C. The "shutdown" setpoint should not be higher than plus 80°C. In this case, the settings must be changed taking into account the fluctuations in the temperature that actually occurs at different times of the year.

Any sharp increase in temperature above the set point (approximately 3...5°C or more than 1 min) is an emergency sign and the hydraulic unit must be stopped immediately. Prolonged rotation of the unit at low speed (below 40% of the nominal speed) is a harsh mode for the thrust bearing segments, so it should be avoided.

Periodically, but at least once a month, it is necessary to monitor the insulation resistance of the guide bearing segments and the thrust bearing disk. Insulation failure can lead to damage to the friction surfaces in the thrust bearing and guide bearings by bearing currents. The insulation resistance must be at least 1 MΩ. The insulation resistance is measured using a megohmmeter at a voltage of 1000 V. The insulation resistance of the thrust bearing disk is measured using a special device with the rotor raised on the brake jacks and the disk not touching the thrust bearing segments.

A thrust bearing that has worked for less than 1000 hours after commissioning or repair allows the hydraulic unit to be started without prior lifting on the brakes after a stop lasting no more than 24 hours. A thrust bearing that has worked for more than 1000 hours after commissioning allows the hydraulic unit to be started without prior lifting on the brakes after a stop lasting no more than 120 hours.

6. Discussion of results based on studying the proposed structures of the thrust bearing, guide bearing, as well as their cooling system

Our paper reports a new concept for designing support and thrust elements of the hydrogenerator. The proposed structure of the thrust bearing has higher moments of mechanical resistance (Fig. 1). The working cross-sections were geometrically increased, and new materials with high yield strengths and high compliance characteristics were used (Fig. 2, 3). Thus, the basic causes of vibration were physically eliminated, and the load capacity of the structure was also improved.

A feature of the proposed structure of the thrust bearing and guide bearing assemblies is the consideration of mechanical factors according to the typical ones described in works [7, 8], which allowed us to choose the optimal stiffness of the antifriction coating material (fluoroplastic-4). The structures of the thrust bearing, guide bearing, and their cooling system proposed in this study, in contrast to those considered in [6], have a simplified design. This resolves the issue of complexity of their operation.

The basic geometric dimensions and operational characteristics of the thrust bearing and guide bearing are given in Tables 2, 5, respectively.

The proposed cooling system for the thrust bearing and guide bearings differs from existing ones in the equilibrium heat transfer coefficients along the entire path of the oil and water flow. Physically, these coefficients are provided by high Reynolds numbers. At the same time, the flow rates along all channels are the same, which ensures the absence of thermal distortions. Physically, this value is characterized by equal deformations over large lengths, which eliminates additional eccentric loads.

The solutions to the problem of optimizing the design of the cooling system reported in [10, 11] were supplemented by taking into account additional factors caused by the presence of oil as a cooling fluid. In the designed cooling system for the thrust bearing and guide bearings, the oil circulation is circular, while maintaining the speed of its movement along the cooling circuit. This, in turn, allowed us to preserve the isotropy of the liquid in the cooling system. A significant increase in the rigidity of the thrust bearing design was ensured by the use of self-aligning wedges, which resulted in a significant reduction in deformations in the assembly.

The analysis reported in [7] allowed us to correlate the vector of force action (mechanical and electromagnetic) with the vector of thermal stresses. The proposed structure of the designed assembly (thrust bearing – guide bearings – cooling system) makes it possible to level the action of the above-mentioned forces by geometric subtraction of these factors.

The proposed scheme of the cooling system with equilibrium heat transfer coefficients along the entire path of oil and water flow is shown in Fig. 5.

In order to achieve the required accuracy in determining the characteristics, a new method for calculating the design of the cooling system of the thrust bearing and guide bearings of hydrogenerators has been proposed. The principal feature of the proposed method is the introduction of a generalized experimental coefficient B into the existing criterion equations, which takes into account the peculiarity of the geometry of the design of oil coolers in addition to the nature of the oil flow and gas-dynamic parameters. The justification for using the criterion equation was introduced both from the water side and from the oil side. Also, such parameters as the action of three-dimensional forces, temperatures, and pressures within the base metal of the structures and coolants were comprehensively taken into account. The thermal calculation was performed taking into account the specificity of heat exchange processes in the oil bath of the thrust bearing. Our thermal and hydraulic calculations have made it possible to determine the maximum temperatures and pressures of oil and water at which the operation of the cooling system of the thrust bearing and guide bearings is ensured without the occurrence of cavitation.

The quality of our results is explained by taking into account the geometry of the structure, technological features of assembly, actual elastic moduli of materials, and taking into account thermal boundary conditions.

The use of criterion equations for thermal calculation that correlated with the results of thermal calculation allowed us to obtain a bath with minimal mass-dimensional characteristics provided that it had sufficient strength.

The shortcomings of [11, 12] regarding the study of heat transfer characteristics under several operating conditions were eliminated by conducting thermal and hydraulic calculations of cooling systems for medium and high-power hydrogenerators. Solving this problem allowed us to optimize hydraulic pressures.

The method proposed in the study resolves the issue of insufficient accuracy in determining the thermal and hydraulic state of the cooling system by taking into account the comprehensive accounting of the factors indicated in the shortcomings of work [14] for the related task. This is reflected in formulas (11), (17), (29), and (35), which characterize the physical state of liquids in the cooling system.

Implementation of our recommendations for maintenance of thrust bearings and guide bearings of hydraulic units makes it possible to ensure their reliable operation while reducing the time spent on their maintenance. This is achieved due to the simplified design of a single-row rigid thrust bearing and its cooling system compared to the structure of a thrust bearing on a hydraulic support [15].

Our study has made it possible to scientifically substantiate the efficiency and feasibility of using the designed new structures of the thrust bearing, guide bearings, as well as their cooling system. Modern design trends in mechanical engineering were taken into account, and the use of new materials was proposed. Existing methods of thermal and hydraulic calculations did not take into account the multi-component nature of the structure of hydrogenerator units.

The high adaptability of the method devised for calculating structures of the thrust bearing and guide bearing cooling systems allowed us to implement it in the workflow at the "Institute of Power Engineering – Research Institute" (Poland, Warsaw).

The results of our study could be used in the design of new hydrogenerators and hydrogenerator-engines of medium and high power, as well as in the reconstruction of existing ones in terms of replacement and modernization of supporting structures. For example, for hydrogenerators at the following power plants in Ukraine: Kyiv PSP (SVO 733/130-36M with a capacity of 33.4 MW under a generator mode and 40 MW under an engine mode); Dniprovska HPP-2 (SV1 1230/140-56M with a capacity of 119 MW); Dnistrovska PSP (SVO2-1255/255-40 with a capacity of 324 MW under a generator mode and 416 MW under an engine mode; Sredniodniprovska HPP (SV 1230/140-48 with a capacity of 117 MW).

Our work has limitations regarding the application of the method for electric machines with a rotation frequency above 1500 rpm. Also, the ratio of the length of the hydrogenerator shaft to its diameter should not exceed 3 to 1. This is due to the high peripheral speeds at which the shaft transitions from rigid to flexible (the thrust bearing does not provide for operation in transient states close to its own critical frequencies).

Further development of this method involves the use of various criterion equations, which will allow the construction of structures for supporting elements in hydrogenerators that will be able to maintain the rigid state of the shaft at high rotation frequencies.

7. Conclusions

1. As a result of our research, a rigid structure of a single-row thrust bearing and a guide bearing was designed. The thrust bearing and bearings of the new design are capable of operating with high contact stresses and a reduced level of vibration. By minimizing the number of parts, it was possible to increase the manufacturability of operations during assembly and disassembly of the structure. The presented structures of the thrust bearing and bearing make it possible to maintain equal strength during temperature changes, which is ensured by the use of a highly efficient cooling system. This is due to the fact that the temperature increase with a multiple increase in load is not proportional, and therefore the elastic modulus of the base material is preserved.

2. The new cooling system of the thrust bearing and guide bearings allows for effective heat removal from the oil. At the same time, the proposed structure prevents foaming of liquids, which guarantees the preservation of their isotropic nature. A feature of the design is such a system for installing oil coolers, which makes it possible to exclude an increase in the heat gradient along the axis of rotation of the generator. As a result, this ensures the absence of vibrations caused by various temperature deformations. The use of new anti-friction materials in the structures of the thrust bearing and guide bearing (fluoroplastic-4) made it possible to enable uniform radial and tangential flow of oil along the channels of the segments. This was achieved by optimizing the geometry of the design, which became possible due to the greater rigidity of the new materials used as the basic ones.

3. In order to confirm the normal operation of oil coolers of the new geometry within the resource requirements, thermal and hydraulic calculations were carried out for hydraulic units of different capacities. This allowed us to take into account the change in flow regimes from laminar to turbulent, the viscosity of the liquid when changing pressure and temperature, as well as the influence of the physical properties of the coolant on heat transfer. According to the results of thermal and hydraulic calculations of the oil cooler of the new design for a hydrogenerator with a capacity of 56 MW, it was established that this oil cooler, with a cooling water flow rate of 80 m^3/h, provides the removal of 250 kW of losses. The pressure drop in the parallel branch is $0.1 \cdot 10^5$ Pa, and the heat transfer margin is 26.2%.

According to the results of thermal and hydraulic calculation of the oil cooler of a new design for a hydrogenerator with a capacity of 96 MW, it was established that this oil cooler, with a cooling water flow rate of 60 m^3/h, provides removal of 150 kW of losses. The pressure drop in the parallel branch is $1.1 \cdot 10^5$ Pa, and the heat transfer reserve is 31.8%.

The results of our calculations indicate that the basic design parameters of the thrust bearing and guide bearings are within acceptable limits, which will ensure the operability of the hydraulic unit under all operating modes.

The criterion equations used in the calculations confirm the possibility of oil coolers operating at supercritical Reynolds numbers: for water Re = 11191, for oil Re = 463 (hydrogenerator capacity 56 MW); for water Re = 26755, for oil Re = 463 (hydrogen generator power 96 MW).

It should be noted that for each type of hydrogen generator a different type of cooler and, accordingly, its own criterion equations are used. This approach previously led to an error in calculations, which on average was approximately 50%.

By taking into account geometric factors in the criterion equations, it was possible to increase the accuracy of calculations to an error of no more than 10%. Subsequently, this allowed us to justify the reduction in temperatures that do not cause expansion within the framework of the calculated tension (according to the 14^{th} quality).

This solution can be extended to hydrogenerators with a capacity of over 100 MW, provided that the design type is maintained (preservation of equivalent pressures and operating temperatures).

4. Recommendations have been compiled for the adjustment and maintenance of thrust bearings and guide bearings of hydraulic units to ensure their reliable operation. The recommendations include a list and sequence of technological operations to achieve uniform load distribution between the thrust bearing segments. Considerable attention was paid to the quality control of the cooling oil. The temperature parameters of the supporting elements of hydraulic units are given, compliance with which ensures the effective use of the resource of these elements.

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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