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DEVELOPMENT OF HORIZONTAL BULB HYDROTURBINES FOR HIGH HEADS WITH A WIDE RANGE OF RELIABLE OPERATION MODES

The issues and directions for improving the energy-cavitation and operational performance of hydro turbine equipment of hydroelectric power plants are considered. The paper analyzes in detail the directions for improving the main indicators characterizing the energy and operational advantages of horizontal hydro turbines. Straight-axis Kaplan hydraulic turbines with a horizontal axis of rotation of the hydraulic unit have incomparable advantages compared to hydraulic turbines with water supply using a spiral case, in terms of higher throughput and a wider range of operation. The practice of hydraulic turbine construction has determined the range of heads for which different types of hydraulic turbines are used. The use of horizontal direct-flow hydraulic turbines for heads of more than 40 meters encounters a number of problems of a hydrodynamic nature, strength, and reliable operation. The paper analyzes the advantages of direct-flow bulb hydro turbines and the possibility of using them for high heads. New design solutions are considered, for which Ukrainian patents have been obtained, allowing the use of horizontal bulb hydraulic units for higher heads (up to 300 meters) and at the same time obtaining a wider operating area not only in terms of flow rates, but also in terms of heads. The use of twin bulb hydraulic units will significantly expand the operational ranges of highly efficient and reliable operation of horizontal bulb hydraulic turbines at flow rates (power) that allow them to successfully operate at variable peak loads of daily regulation. Based on the analysis of the working process of various horizontal and diagonal turbines, the analysis of their universal characteristics, scientifically based proposals was developed for the nomenclature of twin bulb hydraulic units. The design of a horizontal hydraulic turbine using inlet nozzle channels is presented. The use of nozzle diaphragms as elements that create the angular momentum necessary for optimal operation of the hydraulic turbine makes it possible to use bulb direct-flow hydro units for high heads (80–100 meters).
Keywords: bulb hydro turbine, runner, wicket gate, nozzle diaphragm, cate-blade relationship, efficiency.

Є. С. КРУПА

РОЗРОБКА ГОРИЗОНТАЛЬНИХ КАПСУЛЬНИХ ГІДРОТУРБІН НА ВИСОКІ НАПОРИ З ШИРОКИМ ДІАПАЗОНОМ НАДІЙНИХ РЕЖИМІВ ЕКСПЛУАТАЦІЇ

Розглянуті питання та напрями підвищення енергокавітаційних та експлуатаційних показників гідротурбінного обладнання гідроелектростанцій. Докладно проаналізовано напрями удосконалення основних показників, що характеризують енергетичні та експлуатаційні переваги лопатевих гідротурбін горизонтального типу. Прямовісні поворотно-лопатеві гідротурбіни з горизонтальним розташуванням осі обертання гідроагрегату мають незрівнянні переваги в порівнянні з гідротурбінами з підведенням води за допомогою спіральної камери, в частині більш високої пропускної здатності і ширшого діапазону експлуатації. Практика гідротурбобудування визначила діапазон напорів, на які застосовуються різні типи гідротурбін. Застосування горизонтальних прямоточних гідротурбін на напори понад 40 метрів зустрічає ряд проблем гідродинамічного характеру, міцності, а також проблем надійної експлуатації. Проаналізовано переваги прямоточних капсульних гідротурбін та можливість їх застосування на високі напори. Розглянуто нові конструктивні рішення, на які отримано патенти України, що дозволяють застосовувати горизонтальні капсульні гідроагрегати на вищі напори (до 300 метрів) і при цьому отримати ширшу зону експлуатації не лише за витратами, а й за напорами. Використання здвоєних прямоточних гідроагрегатів дозволить істотно розширити експлуатаційні діапазони високоефективної та надійної роботи горизонтальних капсульних гідротурбін на витратах (потужностях), які дозволяють їм успішно функціонувати на змінних, пікових навантаженнях добового регулювання. На основі аналізу робочого процесу різних горизонтальних та діагональних турбін, аналізу їх універсальних характеристик було розроблено науково-обґрунтовані пропозиції щодо номенклатури здвоєних прямоточних гідроагрегатів. Представлено конструкцію горизонтальної гідротурбіни з використанням підвідних соплових каналів. Застосування соплових апаратів як елементів, що створюють необхідний для оптимальної роботи гідротурбіни момент кількості руху, дозволяє використовувати капсульні прямоточні агрегати на високі напори (80–100 метрів).
Ключові слова: капсульна гідротурбіна, робоче колесо, направляючий апарат, сопловий апарат, комбінаторна залежність, коефіцієнт корисної дії.

Introduction. An analysis of the prospects for the development of the electric power industry for the coming decades shows that the increase in energy capacities will mainly be associated with the commissioning of new nuclear power units, and in the long term, thermonuclear ones are also possible. This is due to the acute shortage and high cost of organic fuel [1–6].

Due to the increase in energy consumption by urban transport and facilities not associated with a round-the-clock technological process, the share of peak capacities of daily regulation is increasing. Therefore, with the prospect of developing nuclear energy, many countries are planning a significant increase in hydropower capacity through the construction of new hydroelectric power plants (HPPs) and especially pumped storage plants.

When developing new, more advanced hydro-turbine equipment that is competitive on the world market or when modernizing equipment in operation at HPPs and PSPPs, naturally, in addition to the tasks of increasing the reliability and durability of operation, reducing the cost of building HPPs and manufacturing equipment, tasks are set aimed at developing new technologies for converting the energy of river flows into electrical energy, reducing energy losses at thermal and nuclear power units in the process of generating electricity by improving their operation schedule, closer to working at base loads, while covering peak loads of daily regulation is transferred to more mobile hydraulic units for regulation [1–6].

When developing hydro turbine equipment, along with the requirements for improving operational reliability, the following requirements are also put forward [7–8]:

a) increasing the optimal efficiency and expanding the zone of optimal operation;

b) increasing the average operating efficiency of the hydro turbine;

c) expanding the zone of reliable operation of a hydro turbine in terms of heads and flow rates with high energy-cavitation performance and with an acceptable low level of non-stationary phenomena in the flow, during operation in modes other than optimal;

d) speeding up the process of energy conversion without reducing the energy-cavitation performance of the hydro turbine, by developing more high-speed hydro turbines, which will certainly lead to a reduction in the cost of building HPP and manufacturing equipment, or, when modernizing HPP, will increase the capacity of hydroelectric units and electricity generation by increasing the average operating efficiency and increase the throughflow capacity of the flow path.

Formulation of the problem. The practice of hydraulic turbine construction has determined the range of pressures for which various types of hydraulic turbines are used [8].

Straight-axis adjustable-blade turbines with a horizontal axis of rotation of the hydraulic unit are operated at heads up to 30–40 m.

They have incomparable advantages compared to hydro turbines, in which the water supply is carried out using a spiral case in terms of higher flow rates and wider range of operating power.

The paper considers new design solutions that allow the use of straight-axis hydraulic units for higher heads up to 300 m and at the same time obtain a wider operating area not only in terms of flow rates, but also in terms of heads.

At HPPs, hydro turbines, passing in parallel water flows, carry out energy conversion without interconnection of the working processes occurring in each flow path of a single hydro turbine [4].

There are examples of hydraulic units with a sequential arrangement of flow paths. In this case, the water flow leaving the first hydraulic unit enters directly into the inlet parts of the second hydraulic unit. This arrangement was dictated by the desire to use two faster hydraulic units for a total higher head, compared to that for which each of them was designed.

However, this solution has not found wide application, because the disadvantages of such a solution prevailed in comparison with the advantages obtained.

The paper considers the issue of combining two hydraulic units in series into one dual with a total workflow, in which the cate-blade relationship, which performs the interdependent rotation of the wicket gate vanes (one or two) and the blades of the runners, ensures the optimal distribution of actuated heads in various blade systems and mutual coordination the moments of momentum at the inlet and outlet of the blade systems, necessary to ensure minimal hydraulic losses in the entire flow path.

Development of a series of horizontal capsule hydraulic units for heads up to 250 m. Twin bulb hydraulic unit. The use of direct-flow hydraulic units for heads over 40 m encounters a number of problems of a hydrodynamic, strength nature, as well as problems of reliable operation. However, these difficulties and problems can be solved, and the advantages obtained from the use of direct-flow units are undeniable [4, 8].

As is known, up to 75 % of the costs in the construction of hydroelectric power stations are purely construction (concrete) costs. The cost of hydro turbine and generator equipment rarely reaches 50 % of the total cost of all work.

Bulb-type direct-flow hydroelectric units or with a generator located in a pier, due to the absence of a spiral supply and a curved draft tube, provide great savings in reinforced concrete construction work, because they have a minimum plan width and height of the hydraulic unit block. The advantages of the direct-flow part from a hydraulic point of view are also undeniable. Let's consider them in more detail.

Direct-flow (bulb) hydroelectric units, such as those at Kyiv HPP, Kaniv HPP, etc., despite the complexity of the design and high metal consumption, have significant advantages compared to hydro turbines with a spiral water supply [8, 9]:

1. 30–40 % higher throughput, which allows, with the same runner diameter, to increase power by 30–40 % or reduce the runner diameter at the same power and obtain significant savings in the size of the hydroelectric power station and the metal consumption of reinforcement embedded in concrete.

2. Higher efficiency at optimum and significantly higher average operating efficiency (by 5–10 %).

3. Wider range of operation in terms of heads and flows, wider maneuverability when starting and stopping the hydraulic unit, which is important when working at peak loads.

4. More quiet operation of the hydraulic unit in modes other than optimal in a wide range of changes in flow rate (power) and head. Lower level of flow unsteadiness.

However, direct-flow hydraulic units were not used for heads over 30–40 m, due to the difficulty of creating the necessary moment of momentum of the flow supplied to the runner for optimal operation of the hydraulic turbine without a spiral case.

The invention [10] completely solves this problem and allows the use of direct-flow hydraulic units for heads from 30 to 200 m or more with a capsule layout of the hydraulic unit or in concrete piers (columns). In this case, along with the above, the following advantages, which are characteristic only for this constructive solution, will take place:

1. The range of reliable operation in terms of pressure from H_{max} up to $(0,25–0,3)H_{max}$ with high efficiency, while maintaining a wide range of operation in terms of the flow rate.

2. The advancement of direct-flow hydraulic units to heads over 100 m allow extending the advantages of these hydraulic units to the range of heads where Francis hydraulic turbines are used, which allows:

- with the same diameter of the runner, more than double the power of the hydraulic unit (i.e., reduce the number of hydro units at HPPs, reduce the metal consumption of hydro turbine equipment and especially the metal consumption of reinforcement embedded in concrete, etc.);

- for HPPs, where there is a need for the draft of the reservoir up to $(25–30\ \%)H_{max}$ (for example, the Central Asian republics with irrigation agricultural technologies) allows in the hot summer months with a small inflow of water into the HPP reservoir to generate electricity with a sufficiently high efficiency and a low level of unsteady flow in the hydro turbine tract (low level of pressure pulsation and vibration of the hydroelectric unit) and at extremely low heads;

- increase the average operating efficiency by 5–15 %, i.e. increase electricity generation by the same 5–15 %.

This problem is solved by installing the twin bulb hydraulic unit in the water conduit, with runners of a hydraulic turbine or a pump-turbine of a diagonal or axial type [8–10].

It consists of two turbine-generators located in the same water conduit one after the other, united by a single control system with a triple cate-blade relationship between the opening of the guide vane, common to two turbine-generators, and the turning angles of the blades of the runners.

Fig. 1 shows the twin bulb hydraulic unit. It consists of a water conduit 1, stay vanes 2, a common wicket gate 3, two turbine runners 4 and 5, two hydro generators 6 and 7, and two shafts 8 and 9. Each runner is of axial or diagonal type. Fig. 2 shows a three-dimensional model of the twin bulb hydraulic unit.

Fig. 1. Twin bulb hydraulic unit

Fig. 2. 3D model of the twin bulb hydraulic unit

Twin bulb hydraulic unit works as follows [8–10].

A non-circulating water flow (with a zero moment of momentum relative to the axis of the units) through the water conduit 1, the stay vanes 2 flows onto the guide vanes 3 (set at a certain opening).

Passing the guide vane 3, the flow enters the runner 4 with a certain circulation and causes it to rotate. The rotation of the runner 4 is transmitted to the rotor shaft (not indicated in the drawing) of the hydro generator 6. At the same time, the guide vane 3 creates a part of the moment of the momentum generated by the runner 4, and the runner 5, rotating in the opposite direction, provides a non-circulating flow at the outlet in the optimal mode. Therefore, passing through the runner 4, the flow acquires a negative swirl (circulation). From the runner 4, the flow with a negative swirl enters the runner 5, where this circulation of the flow is depleted, and the flow exits without circulation. The rotation of the runner 5 is transmitted to the hydro generator 7.

The backwater created by the runner 5 improves the energy cavitation characteristics of the runner 4, which makes it possible to operate the runner 4 at heads significantly higher than the limiting heads for existing direct-flow hydraulic units. During operation, the guide vane 3 can change the opening angle (depending on the mode of operation). With a change in the opening of the guide vane 3 through a cate-blade relationship, the angles of rotation of the blades of the runners 4 and 5 (triple regulation) are changed to ensure maximum efficiency in a wide range of regulation [11].

Thus, the use of the twin bulb hydraulic unit can significantly increase the average operating performance, expand the operating area in terms of head and flow rate, increase efficiency, and makes it possible to use a direct-flow scheme for higher pressures [8–10].

As in any hydro turbine, the working process is carried out by creating a torque on the runner of the hydro turbine (which is determined by the total moment of hydrodynamic pressure forces on the blades) equal to [4, 9]

$$M_{tor} = \rho Q\left[\left(\overline{rv_u}\right)_1 - \left(\overline{rv_u}\right)_2\right] - \Delta M_{los},$$

where $\rho Q\left(\overline{rv_u}\right)_1 = \oint\limits_{s_1}\rho\left(rv_u\right)v_n ds$ – the total moment of momentum of the liquid in the cross section S_1 at the inlet of the runner; $\rho Q\left(\overline{rv_u}\right)_2 = \oint\limits_{s_2}\rho\left(rv_u\right)v_n ds$ – the total moment of momentum of the fluid in the cross section S_2 at the outlet of the runner; ΔM_{los} – loss of momentum of the fluid due to friction on stationary surfaces and leakage in the seals bypassing the flow path.

In accordance with the Euler equation for a hydraulic turbine:

$$\frac{\eta_h gH}{\omega} = \left(\overline{rv_u}\right)_1 - \left(\overline{rv_u}\right)_2,$$

where η_h – hydraulic efficiency of the hydraulic turbine; H is the head of the hydraulic turbine; g = 9,81 м/с²; ω – frequency of rotation of the rotor of the hydraulic unit.

During operation of a hydraulic turbine, the hydraulic efficiency depends on the level of hydraulic losses. In the balance of losses, especially in off-design (far from optimal) modes, a significant part of the head

loss is circulating losses: $H_2 = \dfrac{\omega}{\eta g}\left(\overline{rv_u}\right)_2$.

In the optimal operating mode, these losses are close to zero, since $\left(\overline{rv_u}\right)_2 \approx 0$.

The working process of the twin bulb hydraulic unit with a combinatorial system for turning the guide vanes, the runner blades of the first hydraulic unit and the runner blades of the second hydraulic unit from the conditions of a minimum total head loss (maximum efficiency in the currently operating mode) is more complex.

On the basis of the analysis of the working process of various axial and diagonal turbines, the analysis of their universal characteristics, scientifically based proposals on the nomenclature of twin direct-flow hydraulic units were developed [7–10].

Two turbines of diagonal or axial type, the shafts of which are placed horizontally, are installed in the flow path of the twin bulb hydraulic unit (Fig. 3). The choice of layout is determined by the head of the designed hydraulic unit (Table 1).

a   *b*   *c*
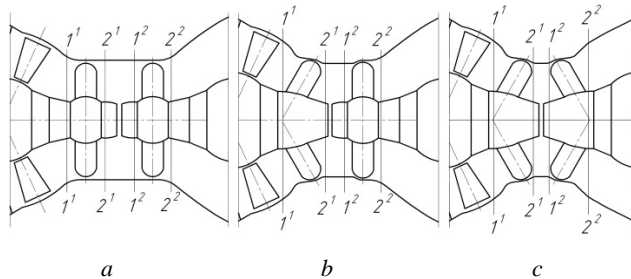
Fig. 3. Schemes of the runner layouts and cross-sections of the twin bulb hydraulic unit:
a – two axial runners; *b* – diagonal and axial runners; *c* – two diagonal runners

Table 1 – Head ranges for the different runner layouts

Scheme	*a*	*b*	*c*
Head range, m	40–75	75–180	180–320

If we designate:

$$H_1^{(1)} = \left[\frac{\omega_1}{\eta_1 g}\left(\overline{rV_u}\right)_1\right]^{(1)} \text{ and } H_2^{(1)} = \left[\frac{\omega_2}{\eta_2 g}\left(\overline{rV_u}\right)_2\right]^{(1)}$$

for the first hydraulic unit and

$$H_1^{(2)} = \left[\frac{\omega_1}{\eta_1 g}\left(\overline{rV_u}\right)_1\right]^{(2)} \text{ and } H_2^{(2)} = \left[\frac{\omega_2}{\eta_2 g}\left(\overline{rV_u}\right)_2\right]^{(2)}$$

for the second hydraulic unit, then the head generated by the first hydraulic unit will be

$$H_{1-2}^{(1)} = H_1^{(1)} - H_2^{(2)},$$

and by the second hydraulic unit

$$H_{1-2}^{(2)} = H_1^{(2)} - H_2^{(2)}.$$

It should be noted that the rotor of the second hydraulic unit rotates in the opposite direction with respect to the direction of rotation of the rotor of the first hydraulic unit and the negative moment of momentum determined by the head $H_2^{(1)}$ is positive for the second hydraulic unit: $\left|-H_2^{(1)}\right| = H_1^{(2)}$.

With increasing the head, the moment of momentum of water in front of the runner $(V_u r)_1$ increases and for its formation in the inlet elements of hydraulic turbines, spiral cases and an increased number of stay vanes are used (at heads of 400 m and more). The difficulty of creating the moment of momentum necessary for the optimal operation of the hydraulic turbine is one of the main reasons that impede the advancement of direct-flow hydraulic units to higher heads. Therefore, in world practice, direct-flow hydraulic units up to 40 m of head are used, in which the angular momentum of the flow leading to the runner (with a non-spiral supply) is created in the stay vane channels and guide vanes.

When operating the twin hydraulic unit in modes other than the optimal drawdown of heads $H_{1-2}^{(1)}$ and $H_{1-2}^{(2)}$ in the first and second hydraulic units is redistributed (according to a cate-blade relationship) in order to minimize energy losses in each operating mode. This also leads to a redistribution of the moments of momentum of the fluid in front of and behind the first runner and in front of and behind the second runner. The cate-blade relationship of the turn of the blades of the first and second runners as a function of the opening of the guide vanes (i.e., changing the water flow or power of the twin hydraulic unit), along with a decrease in all types of hydraulic losses, minimizes the circulation losses of the flow leaving the second unit, and, consequently, minimizes friction losses when flowing around the second capsule.

The fact that for two capsules we have one inlet and one outlet flow organ will certainly lead to an increase in efficiency and in the optimal mode of operation compared to Francis hydro turbines.

Bulb hydraulic unit using nozzle diaphragms. The main disadvantages of the twin bulb hydraulic unit [10] are the long length of the hydraulic unit along the axis of rotation of the hydro turbine-generator rotors, the complexity of the design, which leads to the complexity of the maintenance of the hydraulic unit, the complexity of the work process control system.

The useful model [12] is based on the task of increasing the average operating and energy-cavitation indicators of direct-flow horizontal bulb hydraulic units, expanding the operating area in terms of heads up to 80–100 m.

The task is achieved by the fact that in the turbine in front of the guide vane a nozzle inlet diaphragm of the hydraulic turbine is installed, which is a series of specially profiled curvilinear (spiral) confusing nozzle channels placed around the circumference in front of the guide vane. They provide the necessary moment of momentum for good operation of the hydraulic turbine for heads up to 80–100 m and a uniform flow supply to the runner along the circumferential direction and along the height of the guide vane. In this case, the vanes of the nozzle

diaphragm also play the role of stay vanes, which provide reliable support for the hydraulic unit capsule.

In the Fig. 4 a meridional section of the horizontal direct-flow bulb hydraulic unit is shown; in the Fig. 5 the nozzle confusor channel is presented in plan.
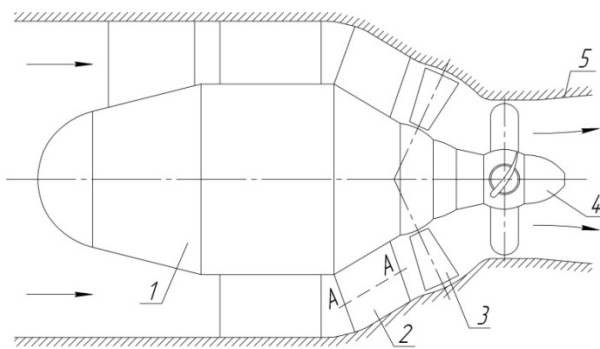

Fig. 4. Horizontal direct-flow bulb hydraulic unit
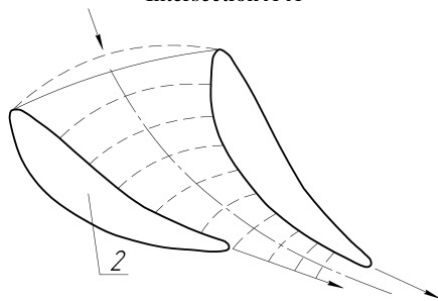
Intersection *A-A*

Fig. 5. Nozzle confusor channel

The horizontal direct-flow bulb hydraulic unit includes a metal capsule 1, in which an electric current generator, bearings, a thrust bearing, auxiliary equipment are located; a nozzle inlet diaphragm 2; guide vanes 3; rotary-vane runner of axial type 4; a draft tube of straight axis type 5.

The horizontal direct-flow bulb hydro unit consists of hydro turbine and electric generator equipment [12].

The hydraulic turbine is the drive of the electric current generator, converting the water flow energy into the mechanical energy of rotation of the rotor of the generator and works as follows. The flow of water with a certain head and flow rate is supplied to the area of the nozzle inlet diaphragm 2. In the confuser, specially profiled nozzle channels located around the circumference in front of the guide vane, the meridional and circumferential components of the flow velocity increase.

At the same time, the necessary moment of the flow momentum and uniformity in the circumferential direction and height of the guide vane are created for optimal operation of the hydraulic turbine, providing high energy-cavitation performance of the hydraulic turbine. Next, the flow enters through the vane channels of the wicket gate 3, which is a control device, into the interblade channels of the axial-type runner 4.

The rotation of the runner blades is carried out due to the control system in accordance with the cate-blade relationship on the opening of the wicket gate vanes, ensuring minimal energy loss [11]. Next, the flow with minimal losses is diverted to the downstream pool through a straight-axis draft tube 5 [12].

Thus, the use of direct-flow horizontal hydraulic units for higher heads (up to 80–100 m) makes it possible to extend the advantages of these units (such as higher throughput, providing greater turbine power with the same runner diameter, higher energy-cavitation performance and operational reliability) to the above-mentioned heads.

Conclusions. 1. The use of twin bulb hydraulic units in special capsules (bulb hydraulic units) or in a concrete pier makes it possible to obtain hydraulic units with a wide range of operation in terms of heads and flow rates for heads of 50–250 meters and higher, increase the average operating efficiency by 5–15 % and provide reliable operation.

2. The use of nozzle diaphragms as elements that create the moment of momentum necessary for optimal operation of the hydraulic turbine allows the use of direct-flow (bulb) hydroelectric units for higher heads (80–100 m).

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