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Mini-Hydropower Plant Based on Lenyov Hydrobelt and Volume-Sectional Hydraulic Engine

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Abstract: The use of the energy of small watercourses with the help of small hydropower plants is one of the promising directions for the development of renewable energy. This article presents the designs of two different hydraulic engines, each of which has its own advantage. Therefore, the task of calculating the real parameters of the design of a mini-hydropower plant based on Lenyov hydrobelt has been solved. Theoretical calculations were validated numerically by the finite volume method and computational fluid dynamics modeling; both methods gave similar results. According to the results of calculations, this design based on the Lenyov hydrobelt with the capacity of 16 kW is advisable to place in a river with a flow velocity of at least 4.5 m/s. The article also presents an alternative type of developed mini-hydropower plant, -a volume-sectional hydraulic engine. The proposed rotary-type positive displacement hydraulic engine can operate at low pressure on a flat surface. The advantage of the hydraulic engine is the sectional operation of several working chambers. It was established that a high water velocity and a large volume of passing water was not required. The total force acting in the hydraulic engine is 5430.19 N. Due to the use of conical inlet channels, the water flow velocity was increased and the water flow became directional. The frequency of rotation of the hydraulic engine shaft at a river flow velocity of 4 m/s was 60.43 rpm. The received power in these modes was 22.25 kW.

Keywords: mini-hydropower plant; Lenyov hydrobelt; Navier-Stokes equations; finite volume method; flow modelling; water energy use; volume-sectional hydraulic engine

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1. Introduction

The use of the energy of small water streams using small hydropower plants (mini-HPP) is one of the promising directions for the development of alternative energy. The Ministry of Investment and Development of the Republic of Kazakhstan listed small damless hydropower plants based on the Lenyov hydrobelt in the category of most economically efficient green technologies of power production [1].

Among existing types of turbines, we propose to consider a new mini-hydropower plant design based on the Lenyov hydrobelt, which uses the original method of energy production from any kind of water flow (rivers, tides, sea waves, etc.). It uses natural flow without prior conversion (construction of dams, canals, or pressure pipes).

The Lenyov hydrobelt uses two shafts with many working blades, which allows for concentrating mechanical energy on the chain (cable, halyard, tape, etc.) and eliminates the need to manufacture blades, shafts and other elements of special strength and accuracy.

In Russia, individual attempts have been made to develop various mini-hydropower plants based on the Lenyov hydropower unit. At the same time, data on the water energy use by the Lenyov hydrobelt are very contradictory. It is stated that when the water velocity is $1 \text{ m}\cdot\text{s}^{-1}$, the output power is 11 kW from an area of 1 m^2 , while the power of the stream itself is only about 1 kW. It is stated that the water velocity after passing through the Lenyov hydrobelt increases 2–2.5 times. However, scientifically proved data on the efficiency of the use of Lenyov hydrobelt is absent in the literature [2,3].

The Lenyov hydrobelt belongs to the hydrokinetic type of turbines (HK systems). HK systems are a class of zero-head hydropower in which energy is extracted from the kinetic energy of flowing water, as in the case of a wind turbine, rather than the potential energy of falling water. These plants can be installed in fast flowing rivers or streams (with a velocity greater than 3 m/s).

The advantage of extracting hydrokinetic energy is that there is no need to build dams or reservoirs. Energy is provided by the natural flow of water in a river, irrigation canal, etc. The aim of the research is to create such a hydrokinetic generator for extracting energy from a fluid stream, which is improved in comparison with generators known up to now. HK turbines are relatively simple designs with no reservoir or spillway requirements. Tests have shown that the environmental impact is minimal, and the simplicity of these devices ensures low installation and maintenance costs. Due to this simplicity, these systems can be of value in rural or remote areas [4,5].

Development of HK systems can be divided into stages, from the traditional period after which the concept of the waterwheel was transferred to the recent empirical period, characterized by the modernization of pilot plants and experiments using computational fluid dynamics (CFD). The technology is currently in a period of growth for HK initiatives embodied in river, tidal and wave energy systems. A factor influencing the development of HK systems is also the rapid development of geographic information system (GIS) technologies with an assessment of the energy potential of small rivers in the regions.

Improving the performance of individual parts of hydropower plants will certainly improve the overall performance of the plant to a more economical and efficient one. Model testing is one of the most important and common approaches for the performance evaluation of hydraulic turbines. However, revolutionary growth of computational facilities in the recent years made the computational fluid dynamics (CFD) the state-of-the-art technique for the designing and evaluation of water turbines. Moreover, it is a time and cost effective approach. The CFD approach is complementary to model testing as it leads to a cost-effective solution, greatly reducing the cost of experimentation. Popularity of the CFD in fluid machinery applications can be realized with the fact that the most widely used turbine with power capacities up to 10 MW is being designed and manufactured based only on numerical CFD analysis [6,7].

Validation and verification of the computational results is an essential practice in CFD analysis of hydraulic turbines. Verification and validation techniques are used to estimate the simulation errors and establish a reliability on computational approaches. Despite certain limitations, CFD is being used extensively because of increased reliability and the capability of shortening the design cycle through controlled parametric study. In fact, the growth of computational approaches in recent times has made the virtual fluid domains more realistic in order to get the minor details of flow, which are not possible in the experimentation [6–8].

The second type of alternative design, which we propose for consideration, is a volume-sectional hydraulic engine. A rotary-type positive displacement hydraulic engine with rotary motion can operate at low head on a flat surface.

The proposed volume-sectional hydraulic engine with a parallel supply of the working fluid can be used for mini-hydropower plants of diversion and channel types. Each type of hydraulic turbine corresponds to a certain range of pressure and flow rate of water and has its own predominant field of application. The main direction in the creation of modern automated micro- and mini-hydropower plants is the use of unregulated hydraulic

engines and increased requirements for devices for generating electricity and stabilizing its parameters.

Development of an efficient mini-HPP design is an alternative to centralized energy supply for rural areas and areas with limited transmission capacity of power transmission lines. It will increase the proportion of electricity generated from renewable energy sources.

The aim of the study is to assess the efficiency of a mini-hydropower plant based on the Lenyov hydrobelt using mathematical tools and computer simulation, determining the real water energy use, as well as the adequacy of the hypotheses about 2–2.5 times increase in water velocity after passing through the Lenyov hydrobelt. The other aim is to develop an alternative type of mini-HPP based on a multi-section hydraulic engine with a parallel supply of working fluid, while water is simultaneously supplied to several working sections.

2. Mini-HPP Based on Lenyov Hydrobelt

2.1. Design of Mini-HPP

To achieve the goal, the study was performed according to the flowchart presented in Figure 1. Field tests are planned as the next step of the study after theoretical validation of the calculations.

Figure 1. Flowchart of the study.

The studied hydrobelt consists of rectangular blades set at 45° to the river flow. Blades are interconnected by a chain to obtain a closed loop rotated by an incoming stream. A frame with a chain can be mounted on floats that are attached to anchors or shores (Figure 2).

The chains transmit the force through the asterisks (impellers) to the two vertical shafts, from which the mechanical energy of the moving medium is transmitted through the flexible coupling and the intermediate shaft to the shafts of the electric generators. The installation shafts are rigidly fixed to the frame of the hydropower unit by rolling bearings. The frame has side and blind bottom walls that are $2/3$ closed, which does not prevent additional water from flowing through the top and $1/3$ of the side walls of the hydropower

unit. The position of the blades in relation to the main flow is regulated by fixed guides for the chain and moving guides for the larger side of the blade.

Figure 2. General view of Lenyov hydrobelt: 1—plate; 2—drive belt-chain; 3—«asterisks»; 4—housing construction; 5—electric generator.

2.2. Calculations

Classical methods of theoretical calculations of fluid mechanics were used as the basic research methods. The finite element method, a numerical method for solving partial differential equations of integral equations, was also used. The problems of hydrodynamics were solved by the system of Navier–Stokes equations.

When calculating the behaviour of a fluid in a region limited by the model (this region may or may not have inlets and outlets connecting it to a fluid outside the computational domain), the so-called fictitious regions method is used, i.e., formally, a computational grid is constructed in a parallelepiped region covering the model with the fluid inside [9,10]. Calculations are carried out only in cells that fall into the computational domain, that is, into a space filled in accordance with the statement of the problem by the fluid medium and solid (if it calculates heat transfer). In the cells outside the calculated area, calculations are not performed. This approach allows for the calculation of flows in very complex channels without complicating the algorithm for solving the problem.

A flat grid of profiles is a set of identical airfoils or plates periodically located in the plane (Figure 3). In [11], a brief theory of the grid of profiles, and a detailed theory of straight grids, both single and double, are presented.

Figure 3. Diagram of the grid plate.

All sources for the calculation of the Lenyov hydrobelt set forth a theory of potential currents, but even at a low velocity, the flow behind the grid is vortex.

In contrast to the flow around a single plate, in an infinite distance in front of and behind the grid of plates the speeds are generally different both in magnitude and in direction. The grid plate not only changes the velocity, but also turns it.

In Figure 3, we denote the velocity of water far from the grid as ϑ_∞ , then near the grid it will be $\vartheta_\infty - \vartheta_1$, and behind the grid $\vartheta_\infty - \vartheta_2$. Then, ϑ_1 and ϑ_2 are the lost velocities, i.e., $\vartheta_\infty - \vartheta_1$ is the flow velocity at infinity in front of the grid and $\vartheta_\infty - \vartheta_2$ is the flow velocity at infinite distance behind the grid. Let us consider the current tube in the plane of the drawing. The tube is formed by any two lines of current shifted relative to each other in the direction of the axis of the grid at the distance equal to the step t . The entire flow through the grid can be divided into such current tubes, since the flow has the property of periodicity with a period equal to a step (Figure 4).

Figure 4. Current tube in the plane of the drawing.

Let us apply the Euler form change momentum theorem [12]:

$$\rho \left[\frac{\partial \vec{\vartheta}}{\partial t} + \left(\vec{\vartheta} \nabla \right) \vec{\vartheta} \right] = -\text{grad } P + \vec{R} \quad (1)$$

where $\vec{R}$ is the main force of plate pressure on the flow; ρ is the water flow density; P is pressure—the force acting on a liquid in an arbitrarily chosen volume; $\vec{\nabla}$ is introduced differential operator. For the current tube with side t and unit depth in the case of steady flow, we have

$$(P_1 - P_2) \vec{t} + \rho \left(\vec{t} \times \vec{\vartheta}_1 \right) \vec{\vartheta}_1 - \rho \left(\vec{t} \times \vec{\vartheta}_2 \right) \vec{\vartheta}_2 - \vec{R} = 0 \quad (2)$$

where $(P_1 - P_2)$ is difference in pressures in front and behind the plane of the grid.

Taking the selected current tube as the control surface and its cross sections at infinite σ_1 and σ_2 parallel to the axis of the grid and equal in length to the step.

The vector $\vec{t}$ is directed perpendicular to the axis of the grid. Consequently, $\left(\vec{t} \vec{\vartheta} \right)$ is the product of the projection of speed and direction perpendicular to the axis by step, and this is the volume flow of water in 1 second. Hence $\left(\vec{t} \vec{\vartheta}_1 \right) = \left(\vec{t} \vec{\vartheta}_2 \right)$ or $\vartheta_{1t} = \vartheta_{2t}$.

Assuming the flow at infinity to be irrotational, the Bernoulli theorem can be applied [12,13]:

$$P_1 - P_2 = 0.5\rho(\vartheta_2^2 - \vartheta_1^2) \quad (3)$$

Replacing the squares of the moduli of velocities by the squares of the vectors of these velocities, we obtain

$$P_1 - P_2 = 0.5\rho \left(\vec{\vartheta}_2 + \vec{\vartheta}_1 \right) \left(\vec{\vartheta}_2 - \vec{\vartheta}_1 \right) \quad (4)$$

Let us introduce the average vector velocity

$$\vec{\vartheta}_m = 0.5 \left(\vec{\vartheta}_2 + \vec{\vartheta}_1 \right) \quad (5)$$

and the velocity of flow deviation, characterizing grid deviation

$$\vec{\vartheta}_d = \vec{\vartheta}_2 - \vec{\vartheta}_1 \quad (6)$$

Then we will have

$$P_1 - P_2 = \rho \vec{\vartheta}_m \times \vec{\vartheta}_d \quad (7)$$

Equation (2) will be rewritten in the form:

$$\vec{R} = \rho \left(\vec{\vartheta}_m \times \vec{\vartheta}_2 \right) \vec{t} + \rho \left(\vec{t} \vec{\vartheta}_1 \right) \left(\vec{\vartheta}_1 - \vec{\vartheta}_2 \right) = \rho \left(\vec{\vartheta}_m \vec{\vartheta}_d \right) \vec{t} - \rho \left(\vec{t} \vec{\vartheta}_m \right) \vec{\vartheta}_d \quad (8)$$

Because

$$\vec{t} \vec{\vartheta}_1 = \vec{t} \vec{\vartheta}_2 = \vec{t} \vec{\vartheta}_m; \quad \vec{t} \vec{\vartheta}_d = \vec{t} \left(\vec{\vartheta}_2 - \vec{\vartheta}_1 \right) = 0 \quad (9)$$

In Equation (8), the expression on the right is the decomposition of the double vector product. Consequently,

$$\vec{R} = \rho \vec{\vartheta}_m \times \left[\vec{t} \times \vec{\vartheta}_d \right] \quad (10)$$

In Equation (10), according to Equation (16), $\vec{t} \perp \vec{\vartheta}_d$ and so $\vec{t} \vec{\vartheta}_d$ is perpendicular to the flow plane and $\vec{\vartheta}_m$. Then

$$R = \rho \vartheta_m \vartheta_d t \quad (11)$$

Equation (11) determines the magnitude of the vector of pressure force on the plate, and Equation (10) shows that the vector $\vec{R}$ is perpendicular to the vector $\vec{\vartheta}_m$.

It is possible to write the vector $\vec{R}$ in the form similar to Zhukovsky formula for a single profile [13]:

$$\vec{R} = \rho \vec{\vartheta}_\infty \times \vec{k} \Gamma \text{ and } R = \rho \vartheta_\infty \Gamma \quad (12)$$

where $\vec{k}$ is unit normal vector to the plane; Γ is contour velocity circulation.

To do this, the unit vectors of the grid axis $\vec{a}$ and the normals to the plane of the drawing $\vec{k}$ should be introduced, directing them so that $\vec{t}, \vec{a}, \vec{k}$ form a trihedron aligned with the right coordinate system. Noting that $\vec{\vartheta}_d = \vec{\vartheta}_2 - \vec{\vartheta}_1$ is parallel to the axis of the grid, we can derive $\vec{\vartheta}_d = (\vartheta_{2a} - \vartheta_{1a}) \vec{a}$, and therefore, $\vec{t} \vec{\vartheta}_d = \vec{t} \vec{a} (\vartheta_{2a} - \vartheta_{1a}) = \vec{t} (\vartheta_{2a} - \vartheta_{1a}) \vec{k}$.

To determine the velocity circulation along the contour composed of sections σ_1 and σ_2 and two current lines shifted by the step, the integral $\int \vartheta_l dl$ should be taken in the positive direction shown in Figure 3. Given that $\sigma_1 = \sigma_2 = t \cdot 1$, we get

$$\Gamma = (\vartheta_{2a} - \vartheta_{1a}) t \quad (13)$$

since the integrals over current lines shifted by a step are equal in magnitude and opposite in direction. So,

$$\vec{t} \times \vec{\vartheta}_d = \vec{k} \Gamma \text{ and } \vec{R} = \rho \vec{\vartheta}_m \times \vec{k} \Gamma \quad (14)$$

In our opinion, this conclusion is not correct. The fact is that liquids and gases transfer pressure in all directions and, therefore, according to Equation (3), the force $\vec{R}$ should be perpendicular not to the direction of flow, but to the plate. The disadvantage of this solution is also that it gives a paradoxical result: when the plate is installed perpendicular to the flow, according to (14) ($\Gamma = \pi\theta_a \times c \sin \theta$), the force R will be maximal. In fact, the force acting on the plate along the OY axis will be zero.

Even at low velocities, the flow is vortex behind the plate. For this reason, the model of potential fluid flow around the plate is applicable only when the angles are no more than 20° . In our case, the angles are close to 45° , which is proved by the calculations below.

The task of wrapping the plate at 90° to the flow was considered by G. Kirchhoff [14]. Later, Reilly solved this task for any angle. Reilly's formula for the force perpendicular to the plate is [13,15]:

$$R = \frac{\pi\rho\theta_\infty^2 \sin \alpha}{4 + \pi \sin \alpha} \quad (15)$$

where ρ is liquid density; α is the angle between the velocity vector at infinity in front of the plate and the plane of the plate. In this case, the force of interest that moves the plate in the direction perpendicular to $\vec{\theta}_\infty$ will be equal to

$$F_y = \frac{\pi\rho\theta_\infty^2 \sin \alpha \cos \alpha}{4 + \pi \sin \alpha} \quad (16)$$

Analysis of Equation (16) shows that the maximum will be at the angle of $39\text{--}40^\circ$. Although in [13] it is stated that Equation (15) at small angles gives the force F_y 4 times lower than the theory of N.Ye. Zhukovsky. It is used below unchanged, since the division of the denominator by 4 does not change the dependence of F_y on the angle α .

A similar result can be obtained in another way. In [16], the problem of the cylinder vortex resistance was considered and the drag coefficient was obtained, $l_x = 1.24$. It is known that by a conformal mapping it is possible to map the area outside the plate to the area outside the circle with radius R (Figure 5).

Figure 5. Plate mapping on a circle.

Map has the form of

$$Z = \frac{l}{4} \left(\xi + \frac{1}{\xi} \right) \quad (R = 1) \quad (17)$$

where Z and ξ are planes; Z is linear fractional function (conformal mapping of the region outside the plate to the region outside the circle of radius R), i.e., Z is obtained from ξ from the linear fractional substitution (transformation); ξ is fractional linear transformation.

Equation (17) displays a plate that is perpendicular to the flow. If instead of l we put $l \sin \alpha$, then the processing of Equation (17) maps the plate to the same circle.

Then $R = 1.24 l \sin \alpha \frac{\rho\theta_\infty^2}{2} = 0.62 l \rho\theta_\infty^2 \sin \alpha$ and $F_y = 0.62 l \rho\theta_\infty^2 \sin \alpha \cos \alpha$.

In this case, the angle between the plate and the flow, giving a maximum of F_y is 45° . According to Equation (16), the coefficient at $\rho\vartheta_\infty^2$ is 0.25.

The next step is obtaining the ratio between the plate length and the grid spacing. Let's consider the flow of water, taking into account the motion of the grid. The interaction of water with the grid leads to the fact that the water velocity varies near the grid. If we denote the water velocity away from the grid by ϑ_∞ , then near the grid it is $\vartheta_\infty - \vartheta_1$ where ϑ_1 is the lost velocity (Figure 6).

Figure 6. Lost and relative velocities.

Further, the grid moves upward with the speed U , and the flow acquires the velocity U_1 , directed downwards (Figure 6). The water velocity relative to the plate will be $U + U_1$, and the modulus of the relative velocity is determined by Equation (18):

$$W = \sqrt{(\vartheta_\infty - \vartheta_1)^2 + (U - U_1)^2} \quad (18)$$

The total force acting on the grid along the y axis is:

$$F_y = 0.5 n l b W^2 C_y \sin \mu \quad (19)$$

where μ is the angle of attack of the relative velocity. The same force acts on water, which acquires the speed U_1 along the y axis. The force $n t b \rho \vartheta_\infty U_1$ acts on the entire grid, which is equal to Equation (19), where b is the plate length, l is the plate width, t is the grid spacing, n is the number of plates. From here

$$U_1 = \frac{C_y W^2 l}{2 t \vartheta_\infty} = \frac{l}{4 t} \frac{W^2}{\vartheta_\infty} \quad (20)$$

where C_y is determined from Equation (16) ($C_y = \frac{2\pi \sin \alpha \cos \alpha}{4 + \pi \sin \alpha}$). When obtaining $C_x = 0.5$ at 45° , similarly we get

$$\vartheta_1 = \frac{l}{4 t} \frac{W^2}{\vartheta_\infty} \quad (21)$$

Using Equations (20) and (21), for W we get

$$W^2 = \frac{\vartheta_\infty^2 + U^2}{1 + 0,5 l t^{-1} - 0,5 l t^{-1} U \vartheta_\infty^{-1}} \quad (22)$$

Substituting Equation (22) into (19), we can determine the power developed by the grid:

$$N = nlbpW^2U \sin \mu \quad (23)$$

If divide this equation by the flow velocity, the coefficient of water energy use can be found:

$$\xi = \frac{2l}{t} \frac{W^2}{\vartheta_\infty^2} \frac{U}{\vartheta_\infty} \sin \mu \quad (24)$$

Since $\mu = \alpha - \beta$ then $\sin \mu = \sin \alpha \cos \beta - \cos \alpha \sin \beta$.

At $\alpha = 45^\circ$, $\sin \alpha = \cos \alpha = 0.5\sqrt{2}$. Then $\sin \mu = 0.5\sqrt{2}(\cos \beta - \sin \beta)$ or $\sin \mu = 0.7 \left[\frac{(\vartheta - \vartheta_1)}{W} - \frac{(U + U_1)}{W} \right]$.

After substituting U_1 , ϑ_1 and W from the Equation (20) and taking into account that $l/t < 1$ and $U/\vartheta_\infty < 1$, we get:

$$\xi = \frac{1.4}{t} \frac{l}{\vartheta_\infty} \frac{U}{\vartheta_\infty} \left(1 + \frac{U^2}{\vartheta_\infty^2} \right)^{1/2} \left(1 + \frac{l}{2t} \right)^{-3/2} \left[1 - \frac{U}{\vartheta_\infty} - \frac{l}{2t} \frac{U}{\vartheta_\infty} - \frac{l}{2t} \frac{U}{\vartheta_\infty} \right]^{-1/2} \quad (25)$$

Complexity of the dependence of ξ from U/ϑ_∞ and l/t makes it difficult to analyze (25). In this light the values of ξ were calculated at $l/t = 0.4; 0.6; 0.8; 0.9; 1.0$ and at different values of U/ϑ_∞ . Results of the calculation are presented in Figure 7.

Figure 7. Dependence of ξ from U/ϑ_∞ and l/t .

The dependences obtained in Figure 7 allow to draw the following conclusions:

- the ratio of the width of the plate to the distance between the plates (grid step) must be equal to 0.9 ($l/t = 0.8$) to achieve a maximum ξ ;
- maximum ξ is reached at the ratio of the grid speed to the flow velocity equal to $U/\vartheta_\infty = 0.4$;
- the calculated maximum value $\xi = 0.186$, at the velocity of $\vartheta_\infty = 4$ m/s with a square meter area, 400 watts of energy can be obtained.

In front of the grid, the flow velocity varies. The water is decelerated and its velocity becomes $\vartheta_\infty - \vartheta_1$. Now it is necessary to analyze in detail how the water velocity changes after passing through the first and the second grids. After passing through the first grid, the water velocity will change, it will acquire the velocity $\vartheta - \vartheta_2$ and U_2 .

Before the second grid, the flow is decelerated again and its velocity will be $\vartheta - \vartheta_2 - \vartheta_3$ and $U_2 - U_3$ (the “−” sign is because the second row plate is angled to the flow at 45°). After passing through the second grid, the velocity will change again. Below, we define these velocity changes (Figure 8).

Figure 8. Water velocities before the first and second grid.

The force acting on the grid along the OY axis is: $F_y = 0.5C_y lb\rho W^2$.

The same force acts on water and creates the velocity U_1 . Then it is equal to $ntb\rho\vartheta_{\infty}U$.

Thus $U_1 = \frac{C_y W^2 l}{2t\vartheta_{\infty}} \sin \mu$.

Since the force according to the Reilly Equation (26) is equal and directed perpendicular to the plate, $F_y = \frac{\pi\rho\vartheta_{\infty}^2 \sin \alpha \cos \alpha}{4 + \pi \sin \alpha}$, $C_y = \frac{2\pi \sin \alpha \cos \alpha}{4 + \pi \sin \alpha}$, and at 45° $C_y = 0.5$.

$$U_1 = \frac{l}{4t} \frac{W^2}{\vartheta_{\infty}} \quad (26)$$

The fluid is inhibited by the plate and the resistance force acts on the plate

$$F_x = 0.5C_x lbW_1^2 \quad (27)$$

On the other hand, this force creates the velocity ϑ_1 and $0.5C_x lbW_1^2 = nt b\rho\vartheta_{\infty}\vartheta_1$.

Hence at $C_x = 0.5$, since $C_y = \frac{2\pi \sin^2 \alpha}{4 + \pi \sin \alpha}$.

$$\vartheta_1 = \frac{l}{4t} \frac{W^2}{\vartheta_{\infty}} \quad (28)$$

Now U_2 and ϑ_2 should be calculated. The formation of vortices can be taken into account by the so-called added mass [16]:

$$m_2 = \frac{\rho S_2 \vartheta_2}{2} \quad (29)$$

where S_2 is plate area.

The velocity of vortex moving towards the flow is $\vartheta_2/2$. Then the flow velocity behind the plate is $\vartheta - 0.5\vartheta_2$ and the force perceived by the grid is equal to

$$F_x = \rho S_2 (\vartheta - 0.5\vartheta_2) \vartheta_2 \quad (30)$$

The mass of the flow involved in the interaction with the grid is

$$m_1 + m_2 = \rho S_2 (\vartheta - \vartheta_2) + \frac{\rho S_2 \vartheta_2}{2} = \rho S_2 (\vartheta - 0.5\vartheta_2) \quad (31)$$

Then the energy conservation law can be written in the form:

$$0.5(m_1 + m_2)\vartheta_{\infty}^2 = F(\vartheta - \vartheta_1) + 0.5(m_1 + m_2)(\vartheta - \vartheta_2)^2 + 0.5m_2\vartheta_2^2 \quad (32)$$

where the expression on the left is the energy of the flow incident on the grid, the first term on the right is the energy perceived by the grid, the second is the energy carried away by the flow, the last term corresponds to the energy consumed for the formation of vortices.

Taking into account that $\frac{m_2}{m_1+m_2} = \frac{\vartheta_2}{2\vartheta_\infty+\vartheta_2}$, we get

$$\vartheta_2 = \frac{2\vartheta_1\vartheta_\infty}{\vartheta_\infty + \vartheta_1} \quad (33)$$

Approximately, we can assume that $\vartheta_2 \approx 2\vartheta_1$, considering the low value of $\vartheta_1/\vartheta_\infty$.

On the other hand, the impulse $nt\vartheta_\infty U_1$ lost by water in 1 second must be equal to the force acting on water $F = 0.5 n l C_y W^2 \sin \mu$, $S = l \sin \mu$ is the area through which water passes to the plate ($b = 1$), where μ is the angle of attack of relative velocity. Since $\mu = 45^\circ - \beta$ (Figure 9), then

$$\sin \mu = \frac{\sqrt{2}}{2} (\cos \beta - \sin \beta), \text{ where } \sin \beta = \frac{U + U_1}{W}, \cos \beta = \frac{\vartheta_\infty - \vartheta_1}{W} \quad (34)$$

Figure 9. Determination of the relative velocity of the flow incident on the plate.

Considering that $W/\vartheta_\infty \approx 1$, we can assume that $\sin \mu = 0.5$. Then at $C_x = 0.5$

$$\vartheta_1 = \frac{l}{8t} \frac{W_1^2}{\vartheta_\infty} \quad (35)$$

Similarly, in the projections on the axis OY we get $nt\vartheta_\infty U_1 = 0.5 l C_y W^2 \sin \mu$ and

$$U_1 = \frac{l}{8t} \frac{W_1^2}{\vartheta_\infty} \quad (36)$$

Considering that $W_1^2 = (\vartheta_\infty - \vartheta_1)^2 + (U + U_1)^2$, Equations (35) and (36), we obtain a biquadratic equation for W , the solution of which is

$$W_1^2 = 1.06\vartheta_\infty U W = 1.03\vartheta_\infty \quad (37)$$

Then $\vartheta_1 = U_1 = 0.106\vartheta_\infty$.

Since $\vartheta_2 = 2\vartheta_1$, and $U_2 = 2U_1$, then

$$\vartheta_2 = U_2 = 0.212\vartheta_\infty \quad (38)$$

The flow past the first grid falls on the second grid. The second grid slows down the incident flow and gives it the velocity along the OY axis. When calculating the velocities U_3 , ϑ_3 and W_2 , it is necessary to take into account that in the Lenyov apparatus the second grid moves in the opposite direction. The flow falling on the second grid has absolute

velocities $\vartheta - \vartheta_2$ and U_2 (Figure 10). Therefore, the angle of attack of both absolute and relative velocities will change.

Figure 10. Absolute (a) and relative (b) velocities of the flow incident on the 2nd plate.

The absolute velocity ϑ'_∞ is the sum of $\vartheta - \vartheta_2$ and U_2 . The angle that it forms with the plate is the angle of attack for the second plate. From Figure 10 it can be seen that $\alpha = 45^\circ + \beta$. In the formulas of C_x and C_y $\sin \alpha$ and $\cos \alpha$ appears. We determined these functions using the results of calculations for the first grid.

By the formulas of trigonometry we have $\sin \alpha = \sin 45^\circ \cos \beta_2^1 + \cos 45^\circ \sin \beta_2^1 = 0.705(\cos \beta_2^1 + \sin \beta_2^1)$.

Since $\sin \beta_2^1 = \frac{U_2}{\vartheta'_\infty} = \frac{0.212\vartheta_\infty}{\sqrt{\vartheta_\infty^2 + \vartheta_2^2 - 2\vartheta_\infty\vartheta_2 + U_2^2}} = 0.26$; $\cos \beta_2^1 = \frac{\vartheta - \vartheta_2}{\vartheta'_\infty} = 0.966$.

To calculate U_3 and ϑ_3 it is necessary to get U_2 and ϑ_2 , as well as ϑ'_∞ . According to Figure 4, this velocity is equal to

$$\vartheta'_\infty = \sqrt{(\vartheta_\infty - \vartheta_2)^2 + U_2^2} = 0.816\vartheta_\infty \quad (39)$$

The relative flow velocity for the second grid is

$$W_2^2 = (\vartheta_\infty - \vartheta_2 - \vartheta_3)^2 + (U + U_3 - U_2)^2 \quad (40)$$

U_3 and ϑ_3 play the roles of U_1 and ϑ_1 , and U_3 and ϑ_3 play the roles of U_2 and ϑ_2 . Therefore, reasoning similarly, we can write

$$\vartheta_3 = \frac{IC_x}{4t} \frac{W_2^2}{\vartheta'_\infty} \sin \mu_2; \quad U_3 = \frac{IC_y}{4t} \frac{W_2^2}{\vartheta'_\infty} \sin \mu_2 \quad (41)$$

Substituting ϑ_2 , ϑ_3 , U_2 and U_3 into Equation (40) we obtain the biquadratic equation, the solution of which is

$$W_2^2 = 0.615\vartheta_\infty^2 \text{ and } W_2 = 0.784\vartheta_\infty \quad (42)$$

Now it is possible to determine the coefficient of water energy use for the second grid:

$$\zeta_2 = \frac{W_2^2}{\vartheta_\infty^2} \frac{U}{\vartheta'_\infty} \sin \mu \quad (43)$$

To determine μ_2 , it is necessary to calculate β_2 from

$$\operatorname{tg} \beta_2 = \frac{U + U_3 - U_2}{\vartheta_\infty - \vartheta_2 - \vartheta_3} \quad (44)$$

Since W_2 has been calculated, determination of U_3 and ϑ_3 is not difficult: $U_3 = 0.06\vartheta_\infty \sin \mu_2$, $\vartheta_3 = 0.103\vartheta_\infty \sin \mu_2$. Substituting them into Equation (41), we get $\sin \mu_2 = 0.287$.

It remains to determine ζ_2 . Substituting in Equation (43) the incoming values, we get $\zeta_2 = 0.130$.

Therefore, the whole Lenyov apparatus can give only 0.316, i.e., one third of the flow energy:

$$\zeta = \zeta_1 + \zeta_2 = 0.186 + 0.130 = 0.316 \quad (45)$$

Thus, based on the calculated model, we can conclude that:

- after passing through the Lenyov unit, the water velocity slows down and is about 0.8 of the velocity before the unit. The velocity behind the unit can increase significantly, in the conditions of elevation difference, according to the formula $pg h = 0.5\rho(\vartheta_2^2 - \vartheta_1^2)$ at $h = 0.5$ m and $\vartheta_1 = 1$ m/s, the velocity $\vartheta_2 = 3.28$ m/s without viscous friction;
- from one square meter of the grid at $\vartheta_1 = 1$ m/s, one can get no more than 200 W. The flow power is $N = 0.5\rho S\vartheta_\infty^2 = 500$ W. Net power is $N_{\text{net}} = 500 \times 0.32 = 160$ W;
- to obtain a net power of 16 kW, the water flow velocity must be at least $\vartheta_\infty = 4.5$ m/s.

The calculated data indicate the need to modify the structure with regard to the derived optimal parameters of the distance between the grids, the angle of inclination, etc.

2.3. Modelling

In the COSMOSFloWorks program, the motion and heat transfer of fluid was simulated using the Navier–Stokes equations, which describe in unsteady formulations the laws of conservation of mass, momentum, and energy of this medium. The calculations were carried out using a standard CFD solver. The geometry of the hydropower plant was installed in a rectangular domain ($6000 \times 1000 \times 12,000$ mm), length of the hydropower plant was 2000 mm, and the rest of the geometric parameters were parameterized in order to find their optimal combination (Figure 11).

The following parameters were defined as variable parameters: blade width— c , blade tilt angle— α , distance between blades— a , distance between rows of blades— b ; target function is unit power (the function whose minimum or maximum is sought and must be a continuous function parameters), the blade height is constant 1000 mm, and the flow velocity increased in the range of $1\text{--}5$ m·s^{−1}. The sensitivity calculation was carried out to determine the changes in the value of the response magnitude caused by a change in the value of the design parameter. The calculation of the optimal solution was carried out by the method of direct selection of options. The radial motion of the plates was not taken into account, since the software does not support this function. It was assumed that the forces on the plates downstream and upstream are mutually compensated, and the result is zero. The problem was solved stationary, i.e., the domain did not rotate. The software used does not have such an ability. The $k\text{--}\varepsilon$ turbulence model was used for calculating the turbulence kinetic energy k and the kinetic energy dissipation rate ε . The buffer layer was not modeled; wall functions were used to calculate the wall velocity. Due to its fast convergence and relatively low memory requirements, the $k\text{--}\varepsilon$ model is very popular in solving industrial problems. The model is well suited for solving problems of external flow around bodies of complex geometric shape. For example, the $k\text{--}\varepsilon$ model can be used to model the flow near a bluff body.

Figure 11. The domain for modelling: mesh $N_x=100$, $N_y=21$, $N_z=123$, where N is number of cells.

2.4. Numerical Solution of the Problem of the Power of the Unit and Its Parameters

The problems of hydrodynamics are solved using the Navier–Stokes equations [17–19]. The results of the calculations are presented in Table 1 and Figure 12. According to this table, a power of 16 kW is achieved at a flow velocity of about 4 m/s. The optimal ratio of the blade width to the grid is $c/a = 0.8$, the distance between rows of blades $b = 3c$.

Table 1. Results of numerical calculations of mini-HPP capacity.

	Calculated Point 1	Calculated Point 2	Calculated Point 3	Calculated Point 4	Calculated Point 5
River velocity [m/s]	1	2	3	4	5
Force 1 first row of blades [N]	448.41	1794.93	4038.40	7182.68	11,223.61
Force 2 second row of blades [N]	219.41	879.57	1981.75	3521.44	5500.07
Force on the chain [N]	667.82	2674.50	6020.14	10,704.12	16,723.68
rpm on the shaft	10.91	21.82	32.73	43.64	54.55
Power [kW]	0.29	2.35	7.92	18.78	36.68

Figure 12. Simulation results at the velocity of 2.7 m/s.

The results of the estimated power of the mini-HPP, depending on the velocity of the river, are presented in Figures 13 and 14. When calculating, the speed of the blades was taken with a coefficient of 0.4 of the velocity of the river according to the following formula:

$$P = \frac{\vec{M} \times rpm}{9549} \quad (46)$$

where M is the torque, N·m; rpm is the number of rounds per minute; P is power, kW; 9549 is proportionality coefficient.

Figure 13. Simulation results at the velocity of 2.7 m/s.

Figure 14. Calculated dependence of the power of the hydrobelt on the velocity of the water flow.

The numerical integration of the equations of motion of a fluid and a grid gives results close to theoretical ones. The theoretical value of the c/a ratio is 0.8, and the numerical calculation is approximately 0.7.

It should be noted that the above numbers were obtained without taking into account the resistance and friction forces during the movement of the belt.

According to the results of calculations, this design with the capacity of 16 kW is advisable to place in a river with a flow velocity of at least 4.5 m/s (calculated point 4 in Table 1). The Mini-HPP based on the Lenyov hydrobelt can be used as an additional unit to the derivational existing mini-HPP (for the power supply of the economic unit and the needs of the mini-HPP owner).

Increasing the flow level in front of the Lenyov hydrobelt will not affect the operation of the mini-HPP turbine, since the classical mini-HPP is higher than the hydrobelt level. Therefore, the water level in front of the hydrobelt cannot be higher than the hydrobelt height.

3. Volume-Sectional Hydraulic Engine

We have developed an alternative type of mini-HPP, a volume-sectional hydraulic engine. The proposed rotary-type positive displacement hydraulic engine with rotary motion can operate at low head on a flat surface. Therefore, the advantage of the hydraulic engine is the sectional operation (several working chambers work at the same time), a high speed of water flow is not required, and a large volume of passing water is not required. From suspended particles in water, a separate housing with a protective cover hermetically protects the gears. For the rest of the turbines described above, the working cycle of each blade is short, therefore the water consumption is large, and in the hydraulic engine, the cycle is long due to the annular working chambers. It is recommended to manufacture the housing and stator from polyethylene material, and the piston and contactors, the ring drum-type rotor, gears, and the shaft rotor are made of metal materials.

The volume-sectional hydraulic engine with a parallel supply of a working fluid (simultaneously water enters several working chambers (section) consists of the following elements (Figure 15): 1—a cylindrical body with a stator 2 inside. On the outer surface of a cylindrical body 1 is installed shaft 3 with closures 4. Closure 4, which passes the piston; for this, window 5 is made, while the function of the closure is to divide the working chamber and pass the piston through the window. At the end of the cylindrical body 1 and stator 2, a rotor 6 is made, a short-drive shaft 7 is installed on the rotor. Shaft 7 of rotor 6 and shaft 3 of contactors 4 rotate synchronously through paired gears 8. Between the end surfaces of contactors 4 and the outer surface of stator 2 and the surface of the body 1 is a pivot-rotating

drum-type rotor 9. Between the outer-end surface of the annular drum-type rotor 9 and the outer surface of the stator 2 and the inner surface of the cylindrical body 1, an annular working chamber 10 is made. The outer-end agility of the annular rotor 9 of the drum-type is provided with an opening 11 for water supply. Ring rotors 9 (in the amount of 2 pieces) of a drum-type are connected to each other through a piston 12. The outer surface of the contactors is equipped with a rubber lining 13. The ends of drum type rotor 9 are rigidly connected to the rotor 6.

Figure 15. Sketch of the general view of the volume-sectional hydraulic engine.

Figure 16 shows the outer body-forming cylindrical segment, consisting of the following elements: 1-outer body-forming cylindrical segment, 2-inner cylindrical chamber-forming segment (stator-shaped), 3-shaft of contactors, 4-cylindrical contactor (separator of the working chamber), 5-window for passage of the piston, 10-annular working chamber, 11-window for water supply in the working chamber, 12-blade-shaped piston, 13-rubber cover of the contactor (serves for sealing). The principle of operation is based on the fact that water enters the throughput window 11, then water under pressure presses on the outer surface of the rubber cover 13 of the closure 4 and simultaneously presses on the piston 12, then the piston begins to rotate along the annular working chamber 10, and the waste water is discharged outside through the discharge channel, then the piston 12 passes through the window of the contactor 5 (the contactor rotates and, accordingly, the annular drum rotor rotates (pivotally)).

The volume-sectional hydraulic engine works as follows: water under pressure enters the internal cavities of the annular drum-type rotor 9, then the pressure water flows through the opening 11 into the working chamber 10; at this moment, under the action of the force of the incoming pressure water, the piston 12 rotates, the rotation of the piston 12 is transmitted ring rotors of the drum type 9, the rotation of the rotor is transmitted to the rotor 6, then to the shaft 7, respectively, then the gears 8 rotate, with the help of the paired gears 8, the rotation is transferred to the shaft 3, respectively, the contactor 4 rotates, while the previously supplied water through the channel K is discharged to the outside. When the piston 12 approaches the outlet channel and further, when the piston approaches the cylindrical closure 4, the movement of water stops. The flow of water

is resumed when the hole 11 passes through the closure 4. The essence of the hydraulic engine is in the presence of a plurality of sectional chambers 10, while the outer surface of the annular drum-type rotor is made in the form of a cylindrical segment, which allows to periodically closing the water supply channel, while periodic required water supply for the implementation of the working cycle of the hydraulic engine. Due to the sequential sectional nature of the working chambers 10 and the parallel supply of pressure water, the continuity of work is carried out, while several sectional chambers operate simultaneously during the working cycle.

Figure 16. Sketch of the body cylindrical segment.

The projected general view of the hydraulic engine is shown in Figure 17. The design consists of a pipe 1 inside which there are working cavities, housing of closures 2, side walls 5, 6, driving gear 4 and drive gear 3.

Figure 17. General view of the volume-sectional hydraulic engine.

Figure 18 shows a sectional view of the engine. Here you can see the rotor 1 with cavities and a side hole, the rotor shaft 3, the rotor drive shaft 7, the contactor shaft 4, the contactor 2, the contactor body 5 and the rotor body 6. The design consists of six sections. The cavities in the rotor are offset relative to each other by an angle of 60 degrees. The first lower rotor and the upper contactor communicate with each other by cavities through a bypass opening located on the side of the rotor and thus form one pair of cylinders. By analogy, the design of the mini-HPP is similar to a six-cylinder internal combustion engine.

Figure 18. General view of the volume-sectional hydraulic engine.

Figure 19 shows inlet 1 and inlet 2.

Figure 19. General view of the volume-sectional hydraulic engine.

The analysis of the structure by the finite element method was carried out in the SolidWorks software. To carry out the calculation, the initial parameters were set: the speed of the water flow is $4 \text{ m}\cdot\text{s}^{-1}$. The density of water is taken as $0.99823 \text{ g}\cdot\text{mL}^{-1}$ at a temperature of $20 \text{ }^{\circ}\text{C}$. The color scale indicates the velocity of water in the structure (Figure 20). After performing the calculation in the SolidWorks software, the water velocity

at the outlet of the mini-HPP was $2 \text{ m}\cdot\text{s}^{-1}$. The speed has halved in comparison with the initial values, i.e., there was a transformation of water energy into mechanical energy of rotation of the output shaft of the mini-HPP, which in turn is connected to a generator for generating electrical energy.

Figure 20. Visualization of water flow in the volume-sectional hydraulic engine.

Figure 20 shows the principle of operation of the mini-HPP structure. Pressurized water enters the rotor cavity through the inlet holes at a speed of $4 \text{ m}\cdot\text{s}^{-1}$. Further, it exerts pressure in this cavity and thereby turns the entire rotor shaft at a certain angle. Since the design of the mini-HPP consists of six sections (cylinders) spaced from each other at an angle of 60 degrees, the cycle of water pressure on the rotor surface is repeated. The 60 degree angle was calculated mathematically to meet the cycle repeatability condition. As a result, we have a rotational movement of the rotor. Movement from the rotor through the drive gear is transmitted through the driven gear to the contactor shaft, forcing the contactors to work as valves. The contactor is made in the form of a hemisphere, inside of which there is a cut-off bar (Figure 20). The contactor is closed from the outside by a casing.

Figure 21 shows the position of the rotors in relation to the inlets. Full and partial overlap of inlet ports is shown. In these sections, the process of pressure of the shafts on the rotor cavity is stopped or partially stopped. The process is cyclical.

Figure 21. Full and partial overlap of the inlet hole of the structure.

For the power calculations of the volume-sectional hydraulic engine the flow velocity of the river is assumed to be 4 m/s. The hydraulic engine has six sections. The force at the inlet to the hydraulic engine, taking into account the intake manifolds in the form of a conical tube, was 3768.5 N. At the outlet of the hydraulic engine, the water force drops by about two times (Table 2). This is facilitated by the design of the hydraulic engine itself, as well as the loss of force due to the resistance of the rubbing parts of the structure, losses in the gears of the structure. The total force acting in the hydraulic engine is 5430.19 N. Due to the use of conical inlet channels, the water flow velocity is increased and the water flow becomes directional. The frequency of rotation of the hydraulic engine shaft at a river flow velocity of 4 m/s is 60.43 rpm. The received power in these modes is 22.25 kW.

Table 2. Results of numerical calculations of volume-sectional hydraulic engine.

Parameter	Sections 1–6
River velocity [m/s]	4
Water force at the hydraulic engine inlet [N]	3768.50
Water force at the outlet of the hydraulic engine [N]	1859.75
Total force [N]	5430.19
rpm on the shaft	60.43
Power [kW]	22.25

4. Conclusions

The results of the calculations led to the following conclusions:

- the ratio of the blade width to the distance between the blades of the hydrobelt (grid step) to achieve a maximum ζ must be equal to 0.9 ($l/t = 0.8$);
- maximum ζ is reached at the ratio of the grid velocity to the flow velocity $U/\vartheta_{\infty} = 0.4$;
- the calculated maximum value $\zeta = 0.186$ (at the velocity of 4 m/s with a grid area of 1 m², about 400 W can be obtained);

- after the passage through the Lenyov unit, the water velocity slows down and is approximately 0.8 of the water velocity before the unit. The velocity behind the unit can increase if there is a minimum height difference $h = 0.5$ m, then at $\vartheta_1 = 1$ m/s before the unit, the velocity behind the unit will increase significantly to $\vartheta_2 = 3.28$ m/s without taking into account viscous friction;
- water flow of at least 4.5 m/s is necessary to obtain a useful power of 16 kW;
- obtained patterns of velocity distribution during the passage of water flow through the hydrobelt and the turbulence zone.

Mini-HPP based on the Lenyov hydrobelt can be used as an additional unit to the derivational existing mini-HPP (for the power supply of the economic unit and the own needs of the mini-HPP owner). We assume that one of the limitations of using the proposed hydraulic engine is the low water flow velocity, which should not be less than 2 m/s; the immersion depth of the structure must be at least 0.5 m.

The alternative type of mini-HPP has also been developed; a rotary-type volume-sectional hydraulic engine with a rotary motion which can operate at low pressure on a flat surface. The advantage of the hydraulic engine is the sectional nature of its operation (several working chambers operate at the same time), a high water velocity is not required, and a large volume of passing water is not required. In this type of hydraulic engine, the working cycle is long due to the annular working chambers.

Since a system of sequentially installed mini-HPPs based on hydraulic belt as well as a sectional type of hydraulic engine, can be promising for a simple user from the point of view of relative cheapness, in the future we plan to manufacture and study prototypes of these mini-HPP in real conditions (for at least two years) along the bed of mountain and lowland rivers.

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