Counter-rotor hydraulic unit on the basis of a nozzle jet hydro turbine

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Abstract. This article analyzes counter-rotor hydraulic units developed on the basis of jet turbines. A new design of a counter-rotor hydraulic unit developed by the authors, consisting of a jet turbine with a nozzle and a water wheel, the principle of its operation and essence is described. Based on the velocity triangle of water flowing through the reactive and active impellers, the rotational speed of the impeller was analyzed. Accordingly, analytical relationships between the energy parameters of the impellers are presented.

1. Introduction

Due to the observed limitation of underground hydrocarbon fuel reserves at the global level and their uneven distribution across countries, an acute shortage of electric and thermal energy is expected in the future. In search of a solution to this problem, scientific research is being carried out in various fields with the aim of increasing the possibilities of using renewable energy in all countries of the world. In particular, if you pay attention to the field of hydropower, you can observe a process associated with a long history. Until the beginning of the 18th century, water wheels were installed and used on sources of low and low pressure water, mainly in the industrial sector to drive various machines. In the middle of the 18th century, after it became possible to manufacture electric generators based on the laws of electric induction, work began and rapidly developed on generating electricity based on jet hydraulic turbines that operate efficiently at various high pressures, created on the basis of the theory of L. Euler [1]. Currently, there are not many water sources that can create high water pressure, in addition, the construction of large hydroelectric stations (HPPs) on them causes great damage to the environment and partly to the economy.

According to the Ministry of Energy, according to the Concept of the Republic of Uzbekistan, by 2030, the growth in electricity consumption in Uzbekistan is expected to reach 110 billion kWh. The concept provides for an increase in the share of renewable energy sources in electricity generation in stages from 11.8% to 25% by 2030. From the bottom, now 5% of them should come from solar energy, 3% from wind energy and 3.8% from hydropower [2].

If we focus on the hydropower potential for the implementation of these tasks, we can see that the main sections of water sources, such as rivers, ditches, canals and irrigation systems, have slight slopes. It is possible to build many micro hydroelectric power plants in places where water pressure can be created in these sources within 2-6 meters [3, 4].

Although there are hydraulic turbines designed to operate at high water pressures, and anti-rotary hydraulic units developed on their basis, the possibility of their effective use at low pressures has not yet been fully resolved.

For example, in [5], the increase in the water energy utilization factor is due to the fact that the water coming from the first guide vane to the first turbine hits and sets it into rotation. Further, the water reflected from the working blades enters the second guide vane and drives the second turbine. The output shaft of both turbines is connected to the input shaft of the summing gearbox, and the output shaft of the summing gearbox is directly connected to the generator. In this hydraulic unit, the water loses most of its energy when it flows out of the first impeller. The outflowing water flow enters the lower, second guide device at an angle. Since upward pressure is created when moving from the pilot to the second impeller, the efficiency of the upper impeller is reduced. In addition, the use of

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a summing gearbox and additional guide devices in the design leads to energy losses due to an increase in local resistances. At low pressures, the complex does not give satisfactory results.
The hydraulic unit [6] consists of a counter-rotor hydraulic turbine and a hydro generator.
A counter-rotor turbine has two coaxial impellers (rotor and counter-rotor) rotating in different directions, to which a stream of water is sequentially supplied. In the counter-rotor hydraulic unit, the rotor of the hydro-generator is installed on the same shaft with the rotor of the hydraulic turbine, and the counter-rotor is mounted on the counter-rotor of the hydraulic turbine. The advantages of a counter-rotor unit in comparison with traditional hydraulic units are: use at higher heads (due to the distribution of pressure between the two impellers of the hydroturbine); the ability to reduce the size and weight of the hydro-generator (since the rotor and counter-rotor of the generator rotate in opposite directions, the frequency of rotation of the rotor relative to the counter-rotor increases significantly). In this hydraulic unit, the outflow of water from the upper impeller of the hydraulic turbine occurs in a vortex mode. As a result, large energy losses are observed. This water flow enters the second impeller from below through the second guide vane. In this case, upward pressure is created. Also, due to the local resistance of the second guide vane and the impeller, energy is lost. As a result, the efficiency of this system will be low, and at heads of 2-10 m, the complex does not give the desired result. The counter-rotor hydraulic unit is a significant complication of the design of the unit and its regulation, as well as difficulties in extracting power from the rotating counter-rotor of the hydro generator. Therefore, this design has not been applied in practice.
In [6], a nozzle jet hydraulic turbine was improved that effectively works with low-pressure water sources by installing an internal guide device. The results of the experiment showed that with a water pressure of 2 meters and a water flow rate of 200 l / s, the efficiency of the hydroturbine was 76.3%.
When these turbines operate in dynamic equilibrium under load, the absolute flow rate of the water leaving the turbine is high, but their kinetic energy is not utilized. To reuse the kinetic energy of such a water flow, a new design of a counter-rotor hydraulic unit has been developed.

2. System Specifications

2.1The essence of the counter-rotary hydraulic unit

The proposed counter-rotary hydraulic unit contains a reactive and active impeller mounted on shafts that are coaxially connected through bearings. A jet turbine with a nozzle presented in [7] was used as a jet hydraulic turbine of a counter-rotor hydraulic unit.

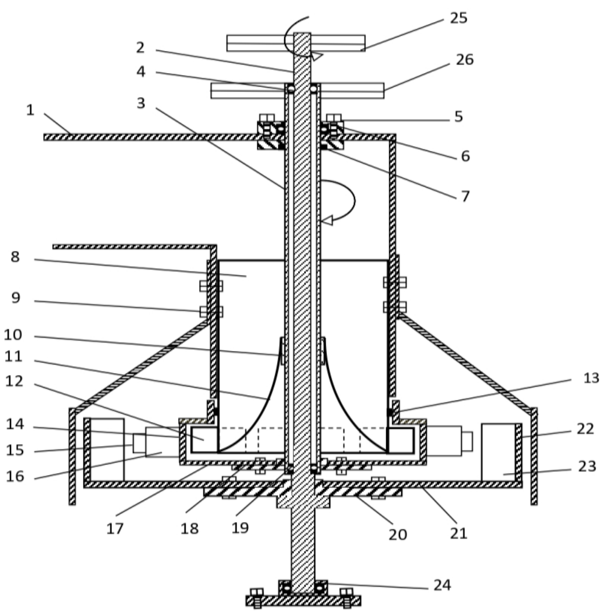


Fig. 1. General schematic view of a counter-rotor hydraulic unit: 1-inlet channel; 2-inner shaft; 3-inner shaft; 4-bearing between the shaft; 5-fixing bolts of the bearing housing; 6-bearing; 7- stuffing box; 8- guide vane; 9-bolts; 10-upper hydraulic bearing ; 11-inner coaxial cone; 12-guide impellers; 13-rubber gland; 14-cylinder impeller; 15-confuser nozzle for water outlet; 16-tap channels; 17-disk base; 18-support with nozzles; 19-cone bearing; 20-disk with trunnion, for fixing the active impeller; 21-disk active impeller; 22- inner and outer rings; 23-blades; 24-case bearing; 25-pulley of the active impeller; 26-reactive impeller pulley

Fig. 1 highlights the jet impeller fixed to the outer shaft 3 contains an internal guide vane made in the form of a cylinder 8, guide vanes 12 which are fixed between the inner coaxial cone 11, which ensures uniform distribution and compression of water along the inner perimeter of the cylinder 8, the lower end of the cylinder having a rectangular hole for directed outflow of water to the nozzle inlet, the impeller made in the form of a cylinder 14. At the same time, the channels 16 for the outflow of water are located on the same horizontal plane with the bottom of the working cylinder, which has an outlet confuser 15, which makes it possible to perpendicularly direct the water flow emerging from the nozzle of the jet to the tangent plane drawn by the point of the center of the arc of the concave and vertically mounted round-cylindrical blade 23 of the active working wheel, which is fixed on the inner shaft 2 of the hydraulic unit. The active impeller is attached to the inner shaft of the disk with a trunnion 20, and the shaft of this impeller is attached to the platform with housing bearings 24. An external shaft of the reactive impeller is inserted above the disk of the active impeller through bearings 4, 19.

In this hydraulic turbine, the relative velocity of the water jet as it exits the nozzle increases in proportion to the water pressure. Usually, for an active Pelton impeller, a jet of water at this speed is created by special nozzles. In the proposed hydraulic unit, the jet of water coming out of the nozzle has the same characteristics, that is, from the nozzle of the impeller, the water jet is ejected at high speed. To use the kinetic energy of the outgoing water jet, an active impeller with curved and vertically mounted blades was developed.

A jet of water exiting the nozzle at high speed strikes perpendicular to the surface of the blades of the active impeller at a short distance. In this case, the blades are located in the same horizontal plane as the nozzle. The number of water jets equal to the number of nozzles falls on the blades of the active impeller. As a result, under the action of pair forces, torques are created that cause the active impeller to rotate in the opposite direction relative to the reactive impeller. The speed of water after impact is almost equal to zero, it moves down under the action of gravitational forces.

The active impeller transmits rotational motion to a separately installed generator through a pulley 25 mounted on an internal shaft. The rotational movement of the nozzle-jet turbine is transmitted to the second generator through a pulley 26 mounted on an external shaft, or it is possible to transfer the rotational movement of the shaft to the generator rotor of the first turbine.

3. Methodology

3.1 Energy parameters of the reaction impeller

To calculate the reactive force F created by water leaving the jet impeller nozzle, the change in the momentum of the incoming and outgoing water in it and the force acting on the nozzle in a perpendicular direction relative to the radial direction are determined:

$$F = \rho S_3 \upsilon_3^2 \left(\cos\beta + \sqrt{\frac{S_3}{NS_4}\left(1 - \frac{S_3}{S_4}\right) + 1 - \frac{1}{2}\left(\xi_{S6} + \xi_2\right)} \right); \tag{1}$$

This design force is the power generated by a single nozzle, and the total reactive power is determined by multiplying it by the number of nozzles. In this case, the absolute velocity of the water jet leaving the nozzle is calculated using the following expressions [8]:

$$\upsilon_2 = \frac{\varphi_0}{R_2^2}\sqrt{2gH\left(R_1^4 - R_1^2 R_2^2 + R_2^4\right)}; \tag{2}$$

$$\upsilon_3 = \frac{\ell_k m \upsilon_2 \sin\alpha_1 \cos(\alpha_2 - \beta)}{2R_2} + \upsilon_2 \cos(\alpha_2 - \alpha_1); \tag{3}$$

$$\upsilon_4 = \upsilon_3 \sqrt{\frac{S_3}{S_4}\left(\frac{S_3}{S_4} - 1\right) + 1 - \frac{1}{2}\left(\xi_{SK} + 1.25\right)}; \tag{4}$$

$$\zeta_{SK} = 0{,}125\lambda \cdot \left(1 - \left(\frac{S_4}{S_3}\right)^2\right);$$

Angular speed of the impeller:

$$\omega_z = \frac{\upsilon_4}{r_c} - \frac{2S_3\upsilon_3^2\left(\mathrm{Cos}\beta + \frac{\upsilon_4}{\upsilon_3}\right)}{\pi \cdot r_c^3} \quad (5)$$

Here υ_3, υ_4 is the water velocity at the entrance to the nozzle; υ_2 is the water velocity at the exit from the guide vane; S_3, S_4 - respectively, the surface of the nozzle inlet and the diameter of the outlet nozzle holes; α_1 and α_2 are the angles of entry and exit of water in the guide vane; λ is the Darcy coefficient ; β is the installation angle of the guide vanes relative to the radial direction; R_2 is the radius of the guide vane cylinder; r_c - distance from the axis of the impeller to the point of the center of the nozzle outlet [9, 10].

It can be seen from formulas (1)-(5) that the reactive force generated in the nozzle is determined by the speed of the water leaving the nozzle, and also depends on the installation angle β of the guide vane relative to the radial direction. The geometric shape of the nozzle ensures the angle of impact of the water jet against the inner walls of the nozzle and the direction of the water back to the exit window of the nozzle, so that the directed flow of water creates a reactive force due to a smooth turn into the nozzle [11, 12, 13]. In accordance with the pressure and flow of water in these hydraulic turbines, the geometric shapes , sizes and number of nozzles must change. Ensuring smooth turning and obtaining the greatest jet force and efficiency of the impeller is related to the shape of the nozzle. Therefore, determining the shape of the nozzle plays a special role in obtaining the maximum value of the difference between the impulses of the incoming and outgoing mass of water per unit time.

3.2 Determining the geometric shape of the jet impeller nozzle

To determine the geometric shape of the nozzle, let's look at the organizational parts of the impeller design. If the number of nozzles on the impeller is equal to m, then each nozzle is located on the arc $A_7 N_1$ corresponding to the central angle $2\pi/m$ in the impeller cylinder. On the rest of the arc $A_6 N_1$, the nozzle is located horizontally along the arc (Fig. 2). If n guide vanes are placed in each nozzle, then the angle between them is equal to $\alpha = \frac{2\pi}{mn}$. The geometric shape of the nozzle is determined by the points A_0, A_1, ... A_n, equal to the number n of the blades. By drawing interpolation lines through these points, the geometric shape of the nozzle is formed in the horizontal plane. Therefore, the number of guide shovels is assumed to be large. In fact, the total number of blades in a diverter is in the range of 4-8, depending on the size of the turbine.

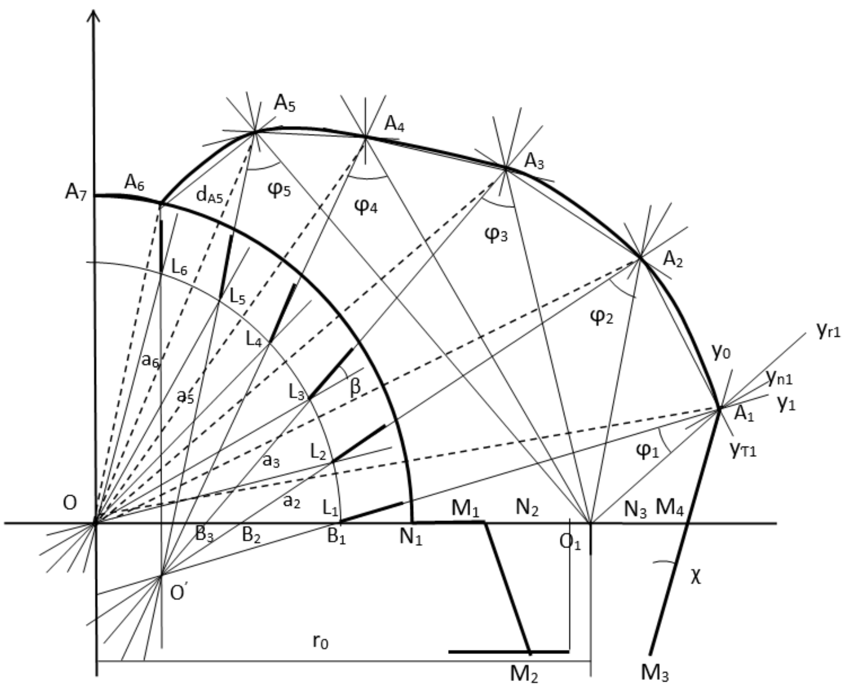


Fig. 2. Method for determining the geometric shape of the nozzle

The blades of the guide device are located at an angle β to the radial direction at the points L_1, L_2, ..., L_n. We form triangles L_I A_I O_1, L_2 A_2 O_1, L_n A_n O_I so that the water flow enters nozzle in the direction of the blade and reflected from the inner wall of the nozzle to the nozzle outlet. If we choose the width of the water outlet channel as $M_2 M_3 = x = 2\ell_0$, and the width of the water inlet to the mixer as $M_1 M_4 = 4\ell_0$, then $N_1 M_1 = M_1 N_2 = N_3 M_4 = \ell_0$. If the taper angle of the nozzle confuser $\angle M_3 = \chi$, then the lengths $N_3 M_3$ and $M_3 M_4$ are calculated in a simple way.
The impeller nozzle consists of two parts - the water supply channel $A_6 H_1$ and the water outlet channel $M_2 M_3$, which also performs the function of a confuser. Top view of the confuser forms a trapezoid $M_1 M_2 M_3 M_4$. Let the cycle frequency of the impeller of the hydraulic turbine ω and the power arm $OO_1 = R_0$ in a water source with water flow Q and head H.
Let's construct the equation of the straight-line y_1 in the direction of the stream from the point L_1, and also draw a straight-line y_0 along $M_3 M_4$. We mark point A_1 at their intersection. This point is located at the greatest distance in the radial direction of the nozzle and determines the height of the nozzle in the radial direction. Considering the laws of geometric optics for a water flow of small diameter, let the reverse flow from point A_I go to point O_1, and we determine the equation y_{rI} in this direction. We will assume that relative to the normal line y_{nI} drawn to the nozzle wall at point A_1, the angles of descent and return of the water flow are equal. The normal line is perpendicular to the tangential line y_{tI} passing through this point. The intersection point of these lines is the X_{A1}, Y_{A1} coordinate of the point A_I on the XO Y plane. To find the coordinates X_{A2}, Y_{A2} of the next point A_2 is determined by equating the tangent line y_{tI} with the equations of the straight line y_2 passing through the line A_2 B_2. Thus, the coordinates of the next points in the series are determined. An interpolation line is drawn through certain coordinates of the point, the geometric shape of the nozzle is formed, and its dimensions are determined (Fig. 2).
We determine the coordinates of the starting points of the geometric figure according to Fig. 2:

$$\begin{gathered} x_{L1} = R_2;\ y_{L1} = 0; \\ x_{N1} = R_S;\ y_{N1} = 0; \\ x_{M1} = R_0 - 2l_0;\ y_{M1} = 0 \\ x_{M2} = R_0 - l_0;\ y_{M2} = -l_0 / tg\chi; \\ x_{M3} = R_0 + l_0;\ y_{M3} = -l_0 / tg\chi; \\ x_{M4} = R_0 + 2l_0;\ y_{M4} = 0 \end{gathered} \tag{6}$$

Slope for the segment $M_3 M_4$ to determine the point A_1:

$$k_0 = \frac{Y_{M2}}{X_{M4} - X_{M3}};$$

Slope corresponding to segment $L_1 A_1$: $k_1 = tg\beta$

$OB_1 = b_1$:

$$b_I = \frac{R_2 \sin\beta}{\sin((I-1)\alpha + \beta)};$$

$_1$ coordinate :

$$\begin{gathered} X_{A1} = \frac{k_1 R_2 - k_0 X_{M4}}{k_1 - k_0}; \\ y_{A1} = k_1(x_{A1} - R_2); \end{gathered} \tag{7}$$

Let us define the coordinates of the following points in a general form:

$$k_1 = tg((I-1)\alpha + \beta); \qquad k_{RI} = \frac{y_{AI}}{x_{AI} - r_0}; \qquad k_{\tau I} = -\frac{1}{k_{NI}};$$

$$\varphi_I = arctg(k_{RI}) - arctg(k_I);$$

$$k_{NI} = tg(\frac{\varphi_I}{2} + arctg(k_I));$$

$$y_{AI} = k_I(x_{AI} - b_I);$$
$$x_{AI} = \frac{k_I b_I - k_{\tau(I-1)} x_{A(I-1)} + y_{A(I-1)}}{k_I - k_{\tau(I-1)}};$$
$$y_{A(n+1)} = R_s in((1 - \frac{p}{100})\frac{2\pi}{m});$$
$$X_{A(n+1)} = R_s Cos\frac{2\pi}{m}; \qquad Y_{A(n+2)} = R_s Sin\frac{2\pi}{m};$$
$$X_{A(n+2)} = 0 \tag{8}$$

R_s - distance from the center of the impeller to the edge of the working cylinder:
where h_k is the height of the guide vanes in the radial direction.

4. Results of mathematical modeling

The Low Re ke interface for turbulent flow was used in the hybrid tooling environment COMSOL Multiphysics 6.0. In this case, single-phase flows were modeled at high Reynolds numbers for the stationary operation of the hydroturbine. The physical interface satisfies the requirements for incompressible and compressible flows with sufficient accuracy at low Max values (typically less than 0.3).

In the geometric shape and dimensions of the nozzle obtained by this method, the momentum conservation equations, the Navier-Stokes equations, and the mass continuity equations for the turbulent flow of water flow in the nozzle through the Low Re ke interface were solved. The effects of turbulence were modeled using the two-parameter ke AKN model with appropriate limitations. The calculations were carried out using the turbulent flow dynamics (CFD) of fluids in the Reynolds-averaged Navier-Stokes dynamic hybrid model (RANS) [14, 15]. In the stationary state, the parameters of the water flow in the nozzle were studied using equations that include geometric nonlinearity in nozzles of two different geometric shapes. In this case, the following Low Re ke interface equations for turbulent flow were solved in the COMSOL Multiphysics 6.0 hybrid instrumentation environment:

$$\begin{cases} \rho(\vec{u}\cdot\nabla)\vec{u} = \nabla\cdot\left[-p\vec{I} + \vec{K}\right] + \vec{F} \\ \rho\nabla\cdot\vec{u} = 0 \\ \vec{K} = (\mu + \mu_T)\left(\nabla\vec{u} + (\nabla u)^T\right) \\ \rho(\vec{u}\cdot\nabla)k = \nabla\left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\nabla k\right] + P_k - \rho\varepsilon \\ \rho(\vec{u}\cdot\nabla)\varepsilon = \nabla\left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\nabla\varepsilon\right] + C_{\varepsilon 1}\frac{\varepsilon}{k}P_k - C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k}f_\varepsilon(\rho,\mu,k,\varepsilon,J_w) \\ \nabla G\cdot\nabla G + \sigma_w G(\nabla\cdot\nabla G) = (1 + 2\sigma_w)G^4 \end{cases} \tag{9}$$

Here:

$$\ell_w = \frac{1}{G} - \frac{\ell_{ref}}{2}$$
$$\mu_T = C_\mu \rho \frac{k^2}{\varepsilon} f_\mu(\rho,\mu,k,\varepsilon,\ell_w) (10)$$
$$P_k = \mu_T\left[\nabla\vec{u} : \left(\nabla\vec{u} + (\nabla\vec{u})^T\right)\right]$$

Table 1. parameters of the jet turbine

Name	Expression	value	Description
dens	1000[kg/m^3]	1000 kg/m³	density of water
visc	1e-5[Pa*s]	1E−5 Pa s	viscosity
versus	6[m/s]	6m/s	water inlet velocity to the hydro turbine
Re	2*R0*vs*dens/visc		Reynolds number
Q	0.2[m^3/s]	0.2 m³/s	water consumption
H	2[m]	2 m	water pressure
omega_z	6.6[rad/s]	7.6rad/s	reactive impeller cyclic frequency
bm	0.004[m]	0.004m	wall thickness
alpha1	15[deg]	0.2618rad	the angle of entry of the water flow into the diverter
alpha2	20[deg]	0.34907rad	the angle of exit of the water flow from the diverter
beta	15[deg]	0.2618rad	the angle of installation of the guide shovel relative to the radial direction
lamb	0.05	0.05	Darcy coefficient
g	9.81[m/s^2]	9.81 m/s²	free fall acceleration
n	8	8	amount of steering gear shovels
fi0	0.95	0.95	coefficient of water input to the hydro turbine
hi	15[deg]	0.2618rad	confusing angle
R1	(Q/(3.14*fi0*(2*g*H)^0.5))^0.5	0.10346 m	supply radius cylinder
R2	0.98*R1	0.10139 m	the radius of the cylinder of the guide device
myu	3	3	the ratio of the length of the guide shovel to the radius of the guide device
delta	0.02[m]	0.02m	the distance between the guide shovel and the working cylinder
hk	0.036[m]	0.036m	the radial height of the guide vane
m	4	4	nozzle quantity
Rs	hk+delta+bm+r2	0.15139 m	impeller radius cylinder
p	10	10	the percentage of nozzle solidification area, %
alpha	2*pi/m*n[deg]	0.21932rad	the central angle between the spades
v2	(fi0/R2^2)*(2*g*H*(R1^4 - R1^2*R2^2 + R2^4))^0.5	6.0774 m/s	water flow velocity at the inlet to the guide vanes
Lk	R2/myu	0.033796 m	guide spade length
v3	(Lk*m*v2*sin(alfa1)*cos(alfa2 - betta))/(2*R2) + v2*cos(alfa2 - alfa1)	7.0989 m/s	the speed of the water flow at the entrance to the nozzle
L	Q/((1 - p/100)*Rs*v3*2*3.14/m)	0.13171 m	guide shovel height
S3	2*3.14*Rs*(1 - p/100)*L/m	0.028173 m²	water inlet surface of the nozzle
S4	S3/2.5	0.011269 m²	outlet surface of the nozzle
eps_k	0.125*lamb*(1 - (S4/S3)^2)	0.00525	coefficient of energy loss in the nozzle
sig	(d*(d - 1) + 1 - (eps_k + 1.25)/2)^0.5	2.0304	
v4	v3*sig	14.413 m/s	the speed of water flow from the nozzle

For the pressure exerted by the liquid on the inner wall of the nozzle, the following relations are used:

$$\left[-p\vec{I}+\vec{K}\right]\vec{n}=-p'_0\,\vec{n},$$

$p'_0 \le p_0$, (eleven)

For the static and osmotic pressure of the nozzle in the boundary condition, the following Lagrange equations were used:

$$\nabla k\cdot\vec{n}=0.\ \nabla\varepsilon\cdot\vec{n}=0.\nabla G\cdot\vec{n}=0$$

The velocities of water entering the nozzle were assumed to be uniform, and the following results were obtained to change the velocity of water in the nozzle (Fig. 3). In this case, the obtained parameters of the jet turbine are presented in Table 1.

5. Results and Discussions

For the case when the geometric shape of the jet turbine nozzle is rectangular, the flow of water through it was modeled according to the parameters given in Table. 1, based on mathematical modeling (Fig. 3).

On fig. It can be seen from Fig. 3 that around the walls of such a nozzle, located on the side of the water outlet channel, there is a mass deficit, as a result of which vortex flows are formed. In this area P<0. The average value of the absolute velocity at the nozzle outlet is small and scattered. The reactive force that occurs in the nozzle is very small, as it is determined by the difference in the momentum of the water flow entering and exiting the nozzle.

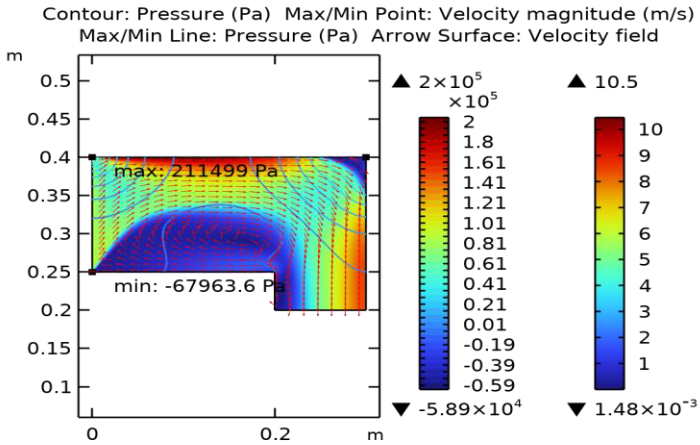


Fig. 3. The outflow of water from a nozzle of a rectangular geometric shape

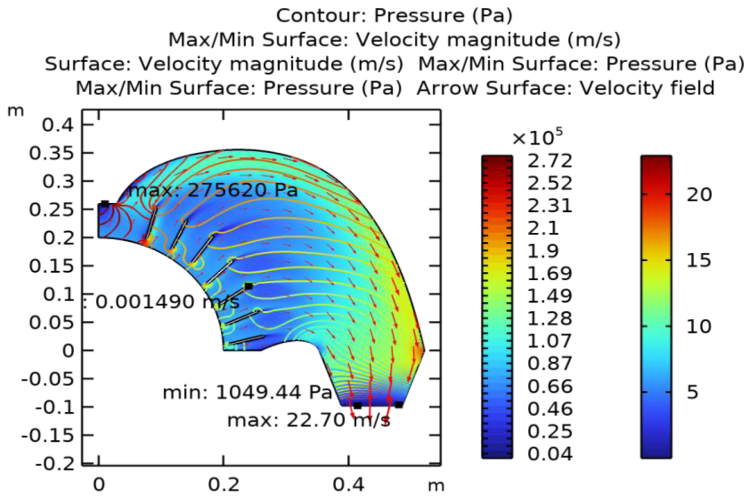
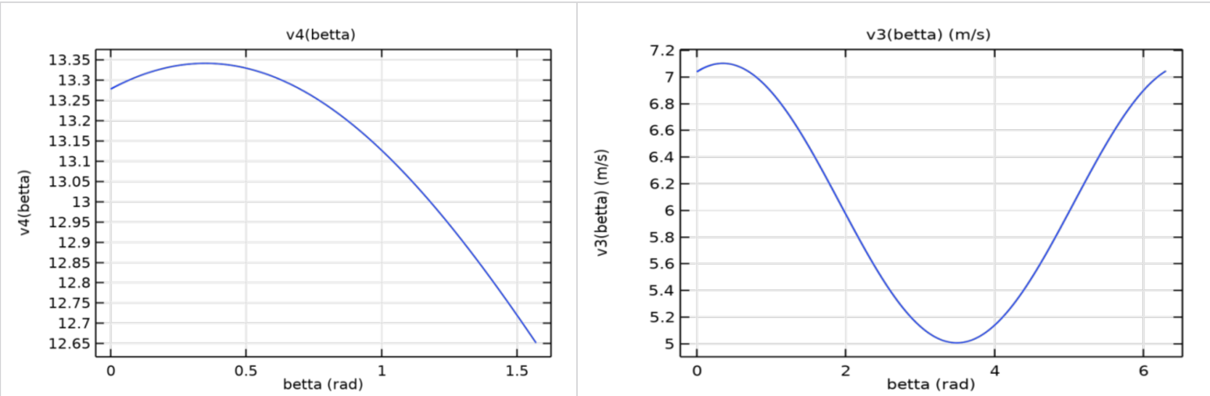


Fig. 4. Behavior of water flow in a nozzle of geometric shape, made by the proposed method

The water flow in a nozzle with a geometric shape created using expressions (6)-(8) was simulated in the hybrid tool environment COMSOL Multiphysics 6.0 on the Low Reke interface for turbulent flow (Fig. 4).
It can be seen from Fig. 4 that there is no shortage of water mass in this nozzle. At all points of the nozzle, a uniform compression of the water flow is observed. Compared with the rectangular nozzle, the pressure generating the reactive force is 1.3 times greater, the absolute velocity of the water exiting the nozzle is 3.2 times greater, and the water jet exits the nozzle at an absolute velocity of 22 m/s without splashing. In an experimental test, the efficiency of this hydroturbine was 82%.



A) *B)*

Fig.5. Dependence of the absolute water flow rates in the hydraulic turbine on the installation angle of the guide vane relative to the radial direction: A) The absolute velocity of the water flow at the outlet of the diverter; B) the speed of the water flow at the outlet of the nozzle

From Figure 5, we can say that the absolute velocity v3(β) of the water leaving the divertor is the highest when the value of the blade installation angle relative to the radial direction β is in the range from 0 to 0.5 radians. For these values of b, the absolute velocity v4(β) of the water jet leaving the nozzle also reaches its maximum value.
A small model of a nozzle hydraulic turbine with a guide was prepared for pilot testing. The structural parts of the finished hydraulic turbine had the following dimensions:
- length of the water supply pipe 60 mm, diameter 63 mm;
- hydraulic turbine shaft length 300 mm, diameter 18 mm;
- the height of the cylinder of the main water supply of the hydraulic turbine is 210 mm, the diameter is 80 mm;
- height of the water supply cylinder of the guide apparatus 80 mm, diameter 76 mm;
- guide vanes 1.5 mm thick, 15 mm long, 16 mm high;
- number of guide shovels 8;
- hydraulic turbine impeller cylinder height 26 mm, diameter 120 mm;
- hydroturbine nozzle height 15 mm, water inlet diameter 20 mm, water outlet diameter 14 mm;
- number of hydraulic turbine nozzles - 12;
- outer diameter of the hydraulic turbine stator 180 mm;
- radial size of the stator water return blade 18 mm, vertical height 20 mm;
- hydraulic turbine pulley diameter 210 mm;
- weight of the hydraulic turbine 2.6 kg

Water enters the impeller nozzle through the guide vanes. At the outlet of the nozzle, the water reaches its maximum velocity. According to the speed of the water, the impeller rotates and the rotational motion is transmitted to the generator with a gear ratio of 1:4 through a pulley mounted on the shaft. The generator is made on the basis of the G250G3 generator.
The angle formed by the normal to the inner wall of the nozzle with guide vanes of the jet turbine will be denoted by α_1, since the distance to the impeller is very small, we can assume that the water flow enters the impeller at this angle. Let's take υ_3 as the absolute water velocity at the outlet of the diverter and determine the relationship between the linear speeds of rotation of the active and jet impeller using the velocity triangle.
In active turbines, the speed of water entering the impeller corresponds to the full value of the operating pressure; with further movement of water, the speed does not change.
As we mentioned above, in jet turbines, the speed of water entering the impeller always corresponds to only a part of the available pressure, so its speed continuously increases as it moves in sections of the impeller. As a result, the

relative velocity v_4 of the water at the outlet is greater than the relative velocity v_3 at the inlet, and since $v_2 = u_3$, the relative velocity v_3 is less than u_3. At the same time, the shape of the nozzle should be such that the water flow moving in them is not compressed in width (vertically), the water outlet should have a straight part at the end, and their curvature should not be excessive in other parts.
a_n active impeller with $\alpha_1 = 25°$, we obtain from the velocity *triangle* $u_{a1} = 0.55\,\upsilon_4$ jet velocity in front of the blade. Based on these data, to calculate the energy released in the active wheel, the moment of inertia of the impeller relative to the center of its shaft is calculated, and the cycle speed and, accordingly, the power of the active impeller are determined.

6. Conclusion

The following conclusions were obtained during the study:
- The pressure generated in the nozzle with a rectangular geometric shape is almost 2.5 times less than the pressure in the curved nozzle. This leads to a multiple decrease in their effectiveness. Therefore, for nozzle turbines, the preparation of the nozzle geometry in accordance with the direction of water flow in the proposed method is the main factor determining the efficiency of the turbine.
- When an active impeller is installed on the outgoing high kinetic energy water flow from the jet turbine, the power generation is increased by 25-40% depending on the pressures, in addition to the electricity generated by the jet turbine, according to the water pressure.
- A counter-rotary unit with a jet turbine and an active water wheel works as efficiently as two separate hydro turbines. Also, the movement of water flows in them does not affect each other. If for some reason one of the impellers stops moving, the remaining second will continue to work.

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