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Results of Experimental Tests of a Nozzle Picohydroturbine with a Guiding Device

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ABSTRACT: In this paper, the advantages and disadvantages of a high-speed nozzle jet turbine in low-pressure water sources are studied. On the basis of theory and experiments, an internal guiding device for this hydraulic turbine was developed and improved. According to the results of experimental tests carried out on the pico-model of the developed hydroturbine, at a water flow rate of 200 *l/s*, and at H=2m, the speed was 191.2 rpm, which is 22.7% more than the analogue. Also, in the model under consideration, the theoretically calculated efficiency of the hydraulic turbine is η_{hyd}=86.2%, according to the results of the experiment, the efficiency of the model was η_{hyd}=82.8%. The results of comparison with analogue are presented.

KEY WORDS: hydroturbine, water source, nozzle, guide vane, electric power, impeller, low pressure.

I. INTRODUCTION

It is well known that the global demand for electricity is growing. Therefore, in the field of hydropower, which is one of the sources of renewable energy, for the effective use of low-pressure water sources, the creation and improvement of hydroturbines that work effectively in low-pressure and flowing water sources is one of the priority tasks.

All over the world, as well as in the Republic of Uzbekistan, there are many places where it is possible to build micro hydroelectric power plants that create a water pressure of up to 3-4 meters. However, hydro turbines operating at high efficiency on such low-pressure water sources are not well developed [1, 2].

II. RELATED WORK

Based on the study of existing low-pressure jet turbines based on various literature and electronic resources, it was found that among them, the nozzle-jet turbine created in [3] is efficient and fast.

This jet turbine contains an impeller rigidly fixed at the bottom of the cylinder with guide impellers, channels for the outflow of water and a stator with reflectors. With mudflow to increase efficiency by increasing reactive efficiency and simplifying the design, the impeller housing is made in the form of a cylinder, blades and channels for the outflow of water, located on the same horizontal plane of the bottom of the working cylinder. Channels for the outflow of water are rectangular pipes located perpendicular to the inner radius of the impeller, having an outlet pipe that allows directing the flow of water leaving the nozzle perpendicular to the tangent plane drawn by the arc center point of a concave and vertically mounted round-cylindrical reflector.

This design has the following disadvantages: when the impeller rotates, the water column in the inlet channel, due to the rotation of the guide vanes, causes the water column directed towards the nozzle to rotate around the shaft, which acts in a tangential direction relative to the radius of the cylinder at each point inside the cylinder. As a result, due to the perpendicularity of the portable speed of the working cylinder relative to the absolute speed of the water, the possibility of water flow towards the nozzle is sharply reduced. Due to the change in the direction of the relative flow velocity, there is a loss of energy and pressure due to an increase in hydraulic and local resistance.

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III. OBJECT AND METHODS OF RESEARCH

To eliminate these shortcomings, the guide vanes were removed from the base of the working cylinder. To direct the water flow to the nozzle and exclude the rotational movement of the water column relative to the vertical axis, as well as the effect of the weight of the water column on the impeller in the vertical direction, an internal guide device is installed.

Let υ_2 be the average value of the vertical velocity of the water flow going vertically from the supply cylinder 1 to the guide device before entering the channels formed by concave impellers 4 fixed vertically between the annular disk 2 and the lower disk 3 in the radial direction. Water flows in a horizontal direction when it is distributed through the 5 channels from a vertical position, a 90° turn causes energy loss in the flow corresponding to the turn, as well as pressure loss due to friction. In order not to lose energy due to expansion during the transition of the water flow from the supply cylinder of the guiding device through the channels to the nozzle, the cross-sectional surface of the supply cylinder with a diameter d_2 must be equal to the sum of the surfaces inlet to the channels S_2. If the channels are located periodically, symmetrically along the periphery of the cylinder, and all their parameters are considered homogeneous:

$$\frac{\pi d_2^2}{4} = a(\pi d_2 - kb), \qquad a = \frac{\pi d_2^2}{4\pi d_2 - 4kb} \tag{1}$$

where, k is the number of impeller; b is the sum of the thickness of the impeller and the width of the window;

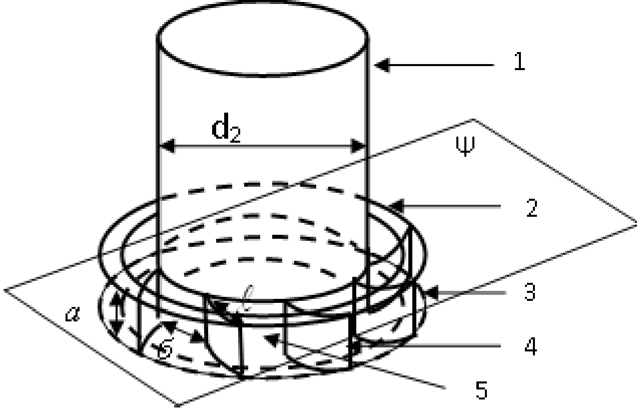

Fig. 1. General schematic view of the guide device.

We direct the OX axis along the width of the mesh profile. To obtain a neighboring profile, it is necessary to shift the grid profile by steps τ. The angle b formed by the direction with the x, 5 as the period of the grid, based on Zhukovsky's theory, we obtain the following expression for the rate of water flow exit from the outlet device [4, 5]:

$$\upsilon_3 = \frac{\Gamma}{\tau} e^{i(\alpha_2 - \beta)} + \upsilon_2 e^{i(\alpha_2 - \alpha_1)}$$

where, υ_2 is the speed of water entering the outlet of the hydroturbine:

$$\upsilon_2 = \sqrt{\frac{\frac{\upsilon_1^2}{g}\left[\frac{d_1^2}{d_2^2}\left(\frac{d_1^2}{d_2^2} - 1\right) + \frac{\alpha_1}{2}\right] - \sum h_{1i} + z_1 - z_2}{\alpha_2} \cdot 2g}$$

Here υ_1 is the speed of water entering the hydroturbine, d_1 and d_2 are the corresponding pipe diameters; - loss of water pressure in the hydraulic turbine; α_1 and α_2 are the Coriolis coefficient, for pipes its value is $\alpha_1=\alpha_2=2$ [6].

IV. RESULTS OF THE RESEARCH

The upper part of the developed guide structure is inserted from the inside into the water supply cylinder of the hydraulic turbine and secured with bolts, and the working part with blades is placed inside the impeller cylinder with a nozzle and does not participate in rotational motion.

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The pico-model of the nozzle hydraulic turbine prepared for the experiment had the following dimensions:
- length of the water supply pipe 60 mm, diameter 63 mm;
- hydraulic turbine shaft length 300 mm, diameter 18 mm;
- cylinder of the main water supply of the hydraulic turbine 210 mm long, 80 mm in diameter;

Dimensions of the organizational part of the guide device:
- feed cylinder 80 mm, diameter 76 mm;
- guide vanes 1.5 mm thick, 15 mm long, 16 mm high;
- number of guide vanes 8;
- hydraulic turbine impeller cylinder height 26 mm, diameter 120 mm;
- hydroturbine nozzle height 15 mm, water inlet diameter 20 mm, water exit diameter 14 mm;
- number of hydraulic turbine nozzles - 12;
- outer diameter of the hydraulic turbine stator 180 mm;
- stator reflector blade size in radial direction 8 mm, height 20 mm;
- hydraulic turbine pulley diameter 210 mm;
- weight of the hydraulic turbine 2.6 kg

The water flow through the guide device passes to the impeller, and the impeller rotates in accordance with the speed of water exit from the nozzle. Rotational motion is transmitted to a specially prepared generator rotor with a gear ratio of 1:4 through a pulley mounted on a shaft attached to the impeller. Below are the organizational parts of the picohydroturbine and its assembled state (Fig. 2).

However, as a result of the research, it turned out that there are practically no research stands for experimental testing of the picohydroturbine model developed based on the results of the research, and the sizes of the existing ones are very large. The equipment used at the stand is outdated. Only one or two types of hydraulic turbines can be tested in them, and the possibility of changing the pressure and water flow is limited [7-9].

Fig. 2. An experimental model of a pico hydro turbine with an internal guiding device.

Preparing the design of a hydroturbine for experiments on these stands requires a lot of money. Therefore, a stand was created with the possibility of studying a prepared pilot sample of various picohydroturbines and a methodology for conducting experiments was developed on it (Fig. 3).

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Fig. 3. Stand for experimental testing of picohydroturbine models.

A jet turbine with a nozzle was tested on a prepared stand.

During the experiment, the water pressure in the hydroturbine model was determined using pressure gauges, the water flow rate was determined on a scale in a graduated container, and the rotational speed of the impeller was determined using a tachometer.

Due to the small size of the picohydroturbine and pipes in the stand, corrections were made for excess energy losses due to turbulent motion and local resistance.

The energy parameters obtained as a result of pilot tests of the picohydroturbine model are presented in the table below (Table 1).

Table. 1

The results of pilot tests of the picohydroturbine model

№	Q, m^3/s	ω, lap/s	U, V	I, A	P, W	η_{HPP}, %
1	0,0151	12,1	221	1,013	223,9	75,6
2	0,0148	11,9	219	1,008	220,8	76,0
3	0,0153	12,2	225	1,018	229,1	76,3
4	0,0149	11,9	220	1,015	223,3	76,4
5	0,0147	11,8	218	1,013	220,8	76,6
6	0,0152	12,1	224	1,017	227,8	76,4
7	0,0148	11,9	218	1,014	221,1	76,1
8	0,0146	11,7	217	1,009	219,0	76,4
9	0,0151	12,1	223	1,018	227,0	76,6
10	0,0149	12,0	221	1,009	223,0	76,3
average amount	0,01494	12	220,6	1,0134	223,6	76,3

V. DISCUSSION

To determine the economic efficiency of a jet hydro turbine micro HPP with an internal guiding structure, a comparison method was used, as in other facilities [10, 11; 319, -36 p.].

From Table. 1 average efficiency of micro HPP was 76.3%. Generator efficiency η_{gen}=0.95 [12], and additional devices $\eta_{transfer}$=0.97 [13], the efficiency of a micro HPP consists of multiplying the efficiency of its parts. Then, the efficiency of the hydro turbine is determined by the following formula:

$$\eta_{hyd} = \frac{\eta_{HPP}}{\eta_{gen} * \eta_{\text{transfer}}} = \frac{0{,}763}{0{,}95 * 0{,}97} * 100\% = 82{,}8\% \quad (4)$$

The efficiency of the hydro turbine was 82.8%.

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Theoretically calculated coefficient of speed at a water flow of 15 *l/s*:

$$n_s = f\left(\frac{Q}{Q_e}\right)^{\frac{1}{2}}\left(\frac{H_e}{H}\right)^{\frac{3}{4}} = \frac{3{,}65nQ^{\frac{1}{2}}}{H^{\frac{3}{4}}} = \frac{3{,}65 \cdot 817 \cdot \sqrt{0{,}015}}{2^{\frac{3}{4}}} = 217rpm$$

The coefficient of speed obtained as a result of the experiment:

$$n_s = f\left(\frac{Q}{Q_e}\right)^{\frac{1}{2}}\left(\frac{H_e}{H}\right)^{\frac{3}{4}} = \frac{3{,}65nQ^{\frac{1}{2}}}{H^{\frac{3}{4}}} = \frac{3{,}65 \cdot 723{,}6 \cdot \sqrt{0{,}015}}{2^{\frac{3}{4}}} = 191{,}2rpm$$

In the prototype presented in [3], the water flow was 200 *l/s*, the speed coefficient at H=2m was 155.8 rpm. For a hydraulic turbine with a guide structure, it was 191.2 rpm and 22.7% more.

In the model under consideration, the theoretically calculated efficiency of the hydroturbine was η_{HPP}=86.2%, according to the results of the experiment, the efficiency of the small model was η_{hyd}=82.8%. In the prototype, the efficiency was equal to η_{hyd}=62.5%.

VI. CONCLUSION

As a result of the theoretical and practical research, the following conclusions were drawn:

- due to the installation of an internal guide structure to the nozzle jet hydraulic turbine, the moment of inertia of the impeller decreases, as a result of which its speed increases. This avoids the use of gearboxes and multistage pulleys.

- When setting the value of the angle b of the blades of the internal guide structure relative to the radial direction 76-78^0, the angle of entry of water into the blades α_1=5^0-8^0, the angle of exit from under the blades 65-68^0, the least energy loss is observed due to local resistance. By eliminating the rotation of the water column, energy losses are reduced by 13% -25% depending on the water pressure.

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