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DESIGN AND ANALYSIS OF SHORT-LENGTH AND LOW FLUID RATE HYDRO TURBINES FOR HOLE DRILLING

The article describes the design and analysis of a small-size hydro turbine with a cardinally new structural diagram for hole drilling. Unlike the conventional bottom-hole hydraulic motors, the proposed design involves inversion of motion of the structural components–the stator rotates around the immobile rotor, which enables an essential increase in the energy efficiency of power fluid.

The research used the methods of inversion, linking and generality to optimize the structure of the hydro turbine.

In terms of a six-sector hydro turbine with a diameter of 196 m, the energy performance of the machine is calculated as function of the drilling length, fluID flow rate and fluID flow angle.

The developed hydro turbine is an effective, small-sized and economically expedient solution for the long-hole drilling practice, which opens prospects for the further experimental validation and introduction of the machine.

Keywords: *drilling, matrix, drillhole, fluid, hydro turbine, structural arrangement, drill bit, ejection, core*
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Introduction

Modern long-hole drilling requires approaches which enable reduction of energy consumption, minimization of accidents and enhancement of drilling equipment efficiency. One of the important trends in this sphere is the development of lower energy and smaller size bottom-hole hydraulic motors which can ensure higher efficiency at lower consumption of power fluid.

The currentness of the research is governed by the requirement of improvement of drilling parameters for oil, gas and groundwater reservoirs as series-produced bottom-hole hydromotors (propeller and turbodrills) have some critical disadvantages in this regard:

- Large size (length 3.0–20.0 m, mass 110–6000 kg);
- High consumption of power fluID (300–1500 l/min);
- Short interrepair cycle;
- Energy loss because of design features.

The research hypothesizes that transition to a brand new construction arrangement—a rotating stator around an immobile rotor—can eliminate the existing disadvantages owing to:

- Unidirectionalilty of power fluID flow and stator torque;
- Increased force lever owing to design features;
- Reduced mass and metal consumption of the machine;
- Enhanced efficiency owing to utilization of ejection effect and jet exhaust.

Meeting the goal involved:

1. Development of a structure chart of a hydro turbine with rotating stator and immobile rotor;
2. Determination of energy characteristics of the hydro turbine depending on power fluID flow rate, drilling depth and structural geometry;
3. Comparative analysis of the proposed design and series-produced hydromotors (propeller and turbodrills);
4. Marketability analysis, including manufacture cost, service life and possible scenarios of introduction.

This study aims at development of a more efficient hydro turbine to enhance the energy efficiency of drilling and to reduce the operating expenditures.

Goals and objectives

The goal is to create a short-length and low fluID consumption hydro turbine for long-hole drilling.

The objectives are:

- formation of structural configuration of hydro turbines with rotating stators and immobile rotors;
- determination of estimated energy performance of hydro turbines depending on the diameter of the stator, hole length and power fluID flow rate;
- comparison of technical and energy performance of the hydro turbines and series-produced bottom-hole hydromotors (propeller, turbodrills).

Literature review and problem solving

The modern research in the area of bottom-hole hydromotors aim to enhance energy efficiency, reduce power fluID flow rate and to improve reliability of the machines. One of the major problems of the conventional turbodrills is their considerable length (12–20 m), low efficiency and high operating cost. Turbodrills are used in long-hole drilling at the temperatures up to 200°C and higher, but they have a limited overhaul life and high hydraulic losses [1, 2].

Problems of conventional bottom-hole hydraulic motors

The energy performance of turbodrills is defined by their blade profiles, which makes them a subject of scientific research. Optimization of blade geometry made it possible to increase the rotational torque at the preserved design constraints [3–5]. Downhole reduction gearboxes effectively increase the rotational torque but their wide introduction is limited by overheating of rubber sealings during long-hole drilling [6]. The research shows that optimization of pitching angles of the stator and rotor blades allows adjusting the operating mode of a turbodrill—either as a high-speed and low-torque or as a low-speed and high-torque motor [7]. However, it is impossible to create an axial turbine which has both high rotational torque and high frequency, and this abridges application in hard rock drilling [8, 9].

For another thing, design features of conventional turbodrills, with rotors rotating inside stators, lead to the increase in the length, mass and fluID flow rate of the machines. So, researches pursue alternative diagrams to ensure more efficient utilization of energy of drill fluID [10].

Promising solutions: Alternative diagrams of bottom-hole hydromotors

Reviews of modern research show that one of the promising solutions is the rotating stator–immobile rotor design which:

• maximizes utilization of the potential energy of the power fluID owing to the concurrency of the directions of the power fluID and stator torque;

• creates the additional rotational torques due to jet exhaust and ejection effect;

• decreases the mass and length of the motor at the same or even increased rotational torque [10, 11].

Researches show that such designs can combine a high rotational torque and a high frequency while eliminating the major shortages of the conventional diagrams [12].

Hydromechanical solutions to enhance drilling efficiency

Alongside improvement of motor designs, the scope of investigations embraces enhancement of drilling efficiency at the cost of hydrodynamic destruction of rocks. One of the trends is the use of abrasive water jets which possess high penetrability and can essentially reduce resistance power of rocks [13]. Researches show that the use of separate water courses and the optimized direction of water flows allow increasing rock fracture rate and enhancing cutting transport efficiency [2].

A keen interest is on fluID ejectors to create additional rotational torques. Optimization of the ejector design, including curved inlets in mixing chambers, makes it possible to increase efficiency of ejectors, which offers new possibilities to enhance efficiency of hydro turbines [1, 14].

Innovative solutions on drilling tools

The current trends show that solution of problems connected with dense rock drilling requires mixed-type methods of rock disintegration, including diamond drilling tools and power fluID jetting. One of the promising technologies is the use of matrices with directional water courses to ensure a hydrodynamic effect of volumetric rock destruction, which greatly lowers the power consumption of drilling [4, 15, 16].

The current research analysis shows that the promising trends of increasing efficiency of bottom-hole hydromotors are the:

• transition to the rotating stator–immobile rotor design, which allows decreasing the length, mass and fluID rate of the machine at the preserved high power;

• optimization of geometry of turbodrill blades to increase the rotational torque and to decrease the hydraulic loss;

• use of abrasive water jets and separate water courses, which decreases the resistance power of rocks;

• use of water ejectors to increase the rotational torques, which can be integrated in the design of hydro turbines drills;

• combination of hydrodynamic and mechanical disintegration, which decreases the energy cost and increases the efficiency of drilling tools.

In this manner, the proposed process solutions make it possible to design smaller size, higher effective and economic bottom-hole motors, which opens prospects for the further research and introduction of new construction diagrams.

Methods and materials

The basis of the new structure chart of a hydro turbine is Euler's theorem on the force impact exerted by an obliquely incident fluid flow on a solid horizontal plane (**Fig. 1**). According to the theorem, if the incident flow has a width b_0 and a velocity ϑ_0, then spraying jets have, respectively, b_1, ϑ_1 and b_2, ϑ_2 (see Fig.1). In this case, the fluid flow rate equation is given by:

$$\vartheta_0 b_0 = \vartheta_1 b_1 + \vartheta_2 b_2 .$$

For finding b_1 and b_2, it is additionally assumed that $\vartheta_1 = \vartheta_2 - \vartheta_0$ and $b_0 = b_1 + b_2$, accordingly:

$$b_1 = \frac{b_0}{2}(1+\cos\propto); \quad b_2 = \frac{b_0}{2}(1-\cos\propto),$$

where $\propto$ is the incidence angle of the fluid jet.

According to the hydrodynamics laws, the rotational torque of a stator is given by:

$$M = \rho \cdot Q \cdot \vartheta \cdot r,$$

where ρ is the power fluid density, kg/m³; Q is the fluid flow rate, m³/s; ϑ s the fluid flow velocity, m/s; r is the radius of the stator, m.

The hydro turbine capacity N is calculated in terms of the rotational torque and rotatory speed:

$$N = M \cdot \omega \cdot N ,$$

where ω is the rotary speed of the stator, rad/s; N is the capacity in W or kW.

The efficiency η is determined as a ratio of the net capacity to the hydraulic power input:

$$\eta = \frac{N}{\rho g Q H},$$

where g is the gravitational acceleration, m/s²; H is the fluid head, m.

Design procedure of machines and mechanisms uses the methods of inversion, linking and generality.

It is sometimes beneficial to change the roles of parts in assemblies, for instance, to make a driving member driven, a guiding member guided, a female member male, or an immobile member mobile. And the structure acquires new properties in each event.

Regarding the structural chart of a hydro turbine in the series-produced hydromotors, the nonrotating stator is made rotational and the rotating rotor–immobile.

This made it possible to create an additional rotational torque of the stator, to simplify the design, to decrease metal consumption and to utilize the force lever.

The packaging arrangement starts from solving the main issues connected with the selection of the kinematic and force diagrams, and determination of their most efficient mutual bracing. Linking goes from the general to the special. The method was applied in engineering of a structural diagram of a hydro turbine. Regarding the generality, the stator both creates the torque and, being made of alloy D16, functions as a friction bearing.

For the explanation of the mode of functioning and to calculate the energy characteristics, we selected the structure chart of a six-sector hydro turbine with a diameter of 196 mm (**Fig. 2**).

Mode of functioning. At the start of drilling and power fluID feed through angle holes 1 and 2 of rotor 3, the fluID flow comes onto the bottom of curved grooves 5 oriented in line of the rotation of stator 4 around rotor 3, separated by partings 6.

The buoyancy force appears and is converted into the rotational torque, and stator 4 with the rigidly connected sliding support rotates.

Then, the flow, reaching the openings between blade 7 of stator 4 and the ends of parting 6 makes a sweep, and the pressure increases at blade 7 and decreases at parting 6. Having put the hydrodynamic pressure on blade 7, the flow comes to return channels 8 in opposite direction to rotation, and, through grooves 9 touching the outer wall of blade 7 in line of rotation of stator 4 and entailed by the flow along grooves 10, is directed to blade 11 with the pressure created by the joint flows.

The other fluID flow in central groove 12 of rotor 3, at the cost of the ejector effect, entails the flow from the groove of stator 4 and, through

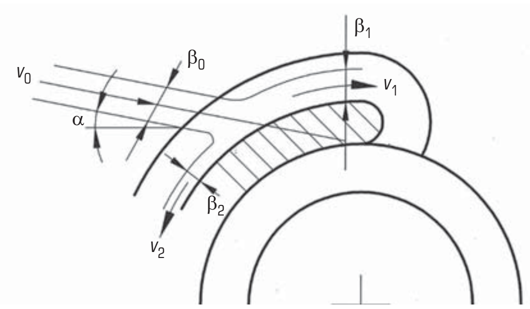

Fig. 1. Spreading of flow of power fluID in incidence on the bottom of curved grooves at an angle ∝ to horizontal plane

side openings 13 in rotor 3, comes in the expanded part of central groove 12, while decreasing the rotation interference of the stator, increasing the velocity and building up the pressure on blades 7 and 11.

In downward motion toward the bottom hole, the joint flows from central groove 12 get into jet exhaust grooves 14 made as spirals in the lower part of stator 4.

The flow makes smooth turning two times, first, along a straight line when entering the curves and, second, when exiting the curves and entering the straight line sections, at the increased pressure on the outer walls of grooves 14 along the line of rotation of stator 4.

The mud from the bottom hole and to the surface passes to ascending rotation and, affecting the curved surfaces of stator 4, generates an additional rotational torque on the outer surface of the stator.

The hydro turbine is equipped with a diamond drill bit with a separate system of water courses, with divided channels to feed the power fluID and to remove the mud. The diamond drill bit has a matrix divided by water grooves into sectors, from which end groove go, which are chequerwise displaced horizontally between the sectors with mutual overlapping along the circle (see Fig. 2).

A feature of the diamond drill bits with the separate systems of water courses is creation of early bottom-hole conditions for the hydrodynamic effect of volumetric rock disintegration owing to the incident flow of power fluID with the high-rate and obstacle-free removal of mud along the circular grooves [17–21].

Results

According to Euler's theorem, the time derivative of motion vector of material points is equal to the main vector of external forces affecting the system of the points:

$$\frac{d\vec{Q}}{dt} = \vec{R}\ .$$

Because of smallness of mass forces and friction forces, we take the main vector as the resultant pressure force $\vec{R}$. The task is to determine the pressure force $\vec{P}$ of the fluid jet on an obstacle, then $\vec{R} = -\vec{P}$ and, accordingly:

$$\frac{\overrightarrow{dQ}}{dt} = \overrightarrow{-P}\ .$$

The change in the kinetic momentum $\overrightarrow{dQ}$ in the time dt is equal to the difference of second-long pulses in the final and initial section. In this regard, it is possible to state that "Projection of jet pressure on any axis of the force vector is equal to the difference of the projections of the second-long kinetic momentums in the initial and end sections of the jet on the same axis of the force vector" [22]:

$$\vec{P} = -\frac{\overrightarrow{dQ}}{dt} = M_0\overrightarrow{\upsilon_0} - M_1\overrightarrow{\upsilon_1}\ ,$$

where $M = \rho \cdot Q$ is the mass flow through a certain section of the jet per second; M_0 is the mass fluid rate of the jet in the initial section (0), kg/s; M_1 is the mass fluid rate of the jet in the initial section (1), kg/s; $\vec{\upsilon}_0$ is the average velocity vector of the flow in the initial section, m/s (in line with the jet flow in this section); $\vec{\upsilon}_1$ is the average velocity vector of the flow in the end section, m/s.

Let us discuss rotation of a six-section stator around an immobile rotor under the action of pressure fluid fed in circular grooves simultaneously along two rotor channels b_1 and b_2. In order to utilize the kinetic energy of the flow, it is necessary to make the blades move with the flow in the directions shown by the arrows in Fig. 2.

Since the fluid is fed through two channels (b_1 and b_2) 8.0 mm in diameter each at an angle α in the direction of the stator rotation, then, with regard to the angle, the flow force converted into the rotational torque is given by:
for the external channel

$$P_1 = \rho \cdot Q_1 \cdot \vartheta_f \cdot \sin\alpha,$$

for the internal channel

$$P_2 = \rho \cdot Q_2 \cdot \vartheta_f \cdot \sin\alpha,$$

where $\vartheta_f = \sqrt{2gH}$ is the incident fluid flow velocity depending on the feed height H , m/s; Q_1 is the fluid flow rate in the external channel, m^3/s; Q_2 the fluid flow rate in the internal channel, m^3/s.

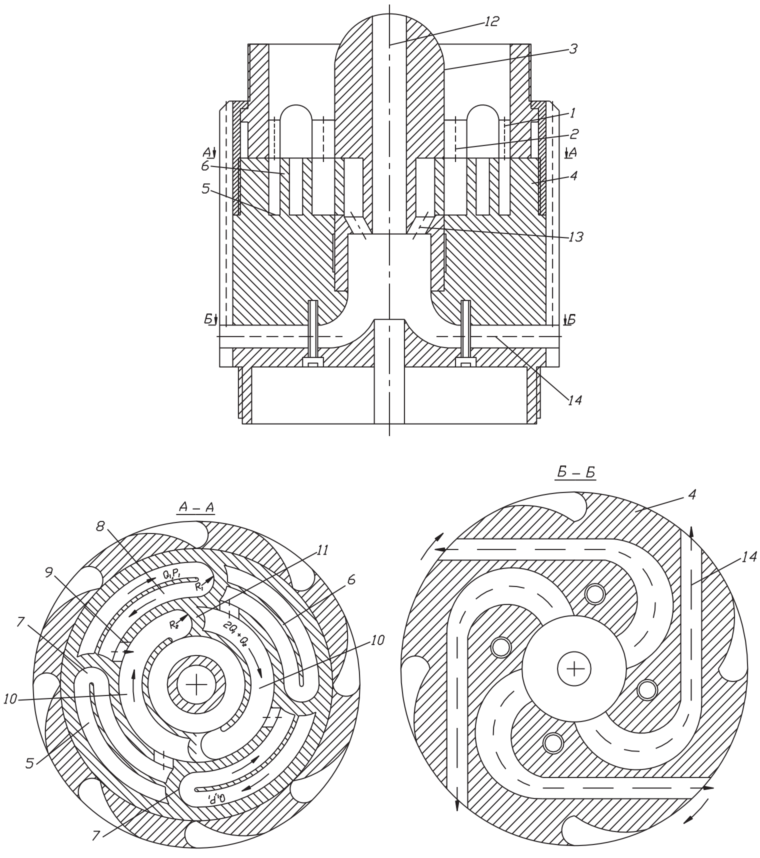

Fig. 2. Structure chart of six-sector hydro turbine with a diameter of 196 mm

The semicylindrical geometry of the blade with the curved surface conditions the fluid flow rotation at an angle of 180°. This provides the doubled dynamic flow impact Q_1 and Q_2 on it.

Initially, at the end of the parting, the flow, while placing the force impact on the semicylindrical blades on the external chamber with a radius R_1 and, according to the diagram of the hydraulic machine, the dynamic impact $2Q_{1+}Q_2$ on the blades of the external chamber with a radius R_2 , with participation of the force P_1 and P_2, puts the stator into rotation (see Fig. 2).

We project the vector equation on the flow axis and obtain the dynamic fluid pressure on the blades of the external and internal chamber, respectively:

$$P_{b1} = 2 \cdot \rho \cdot Q_1 \cdot \vartheta \ ;$$

$$P_{b2} = 2 \cdot \rho \cdot (2Q_1 + Q_2) \cdot \vartheta,$$

where $\vartheta = \vartheta_f - \vartheta_b$ is the incident flow velocity on the blades, m/s; $\vartheta_b = \dfrac{\pi \cdot D \cdot n}{1000 \cdot 60}$ is the blade velocity, m/s; D is the stator diameter, m; n is the rotational frequency, min^{-1}. It is assumed that n=500 min^{-1}.

The impact torque of the flow on the bottom of the external chamber is given by:

$$M_1 = P_1 \cdot h_1 \ ,$$

where h_1 is the force lever, m.

The impact torque of the flow on the bottom of the internal chamber is:

$$M_2 = P_2 \cdot h_2 \ ,$$

where h_2 is the force lever, m.

The torque of the external chamber blade is:

$$M_{b1} = P_{b1} \cdot \frac{\pi R_1^2}{2} \cdot \ell_1 \ ,$$

where ℓ_1 is the blade height, m.

The torque of the external chamber blade is:

$$M_{b2} = P_{b2} \cdot \frac{\pi R_2^2}{2} \cdot \ell_2 \ ,$$

where ℓ_2 is the blade height, m.

The pressure difference in the jet exhaust in the channels in the lower part of the stator is:

$$\Delta P_p = \frac{\vartheta_p^2}{\mu^2} \cdot \frac{1000}{2g} \ ,$$

where $\vartheta_p = \dfrac{4 \cdot (2Q_1 + Q_2)}{\pi \cdot b^2}$ is the jet exhaust velocity in the lower part of the stator, m/s; 4 is the number of channels; b is the width of channels, m; $\mu = 0.448$ is the flow rate through the inlet opening, this is a product of the jet velocity coefficient and the jet compression coefficient at the outlet of the openings.

Then, the rotational torque of the stator due to the jet exhaust of flow is:

$$M_p = \Delta P_p \cdot \frac{\pi \cdot D^2}{4} \cdot h_3 \ ,$$

where h_3 is the force lever, m.

The total rotational torque of the stator with regard to 6 sections is:

$$\Sigma M_{\lim} = \left(M_1 + M_2 + M_{b1} + M_{b2} + M_p\right) \cdot 6 \ ,$$

The theoretic efficiency of the bottom-hole hydromotor is given by:

$$\eta = 2 \cdot \left(1 - \frac{\vartheta_b}{\vartheta}\right) \cdot \frac{\vartheta_b}{\vartheta} \cdot \left(1 + \cos\beta\right) \ ,$$

where $\beta = 0^0$ is the angle of jet reflected from the blades.

With regard to η and theoretical efficiency, the rotational torque of the stator is:

$$M_s = \Sigma M_{\lim} \cdot \eta \ .$$

The output capacity of the rotor hydromotor is:

$$N = \frac{M_s \cdot h}{9000.55} \ ,$$

where h = 500 min^{-1} is the rotational frequency.

The calculation input data were, mm: $R_1 = 8.0$; $R_2 = 8.0$; $h_1 = 80.0$; $h_2 = 43.0$; $\ell_1 = 30$; $\ell_2 = 30$; $D = 120$; $d_p = 20\left(\textit{rotor diameter}\right)$; $b_1 = 8$; $b_2 = 8$; $b = 10\ \left(\textit{depth}\ 10\right)$.

Figures 3 and **4** depict the calculated energy performance of the structural diagram of the hydro turbine depending on the hole length at the increase in the power fluid flow rate and in the flow incidence angle $\propto$ relative to horizontal plane.

Table 1 offers the calculation matrix for the technical and energy performances of the series-produced bottom-hole hydromotors and the structural chart of the hydro turbine.

Table 2 compares the technical and energy performance of the proposed hydro turbine and series-produced bottom-hole motors. All data are obtained from the present-day literature [1, 2, 10–13] and from the calculations performed within this research.

The data correctness was checked using the normalization methodology. The calculations had two stages:

1. Theoretical modeling in MATLAB with regard to turbulence of power fluid;
2. Verification using available data on series-produced hydromotors.

It is important to mention that the advantages are not because of errors but due to novel design principles. The major advantages are:

• The decreased power fluid flow rate by 40–50% owing to unidirectional flow;

• The increased rotational torque by 30–35% owing to utilization of jet exhaust;

• The decreased mass more than by 3 times as compared with the analogs.

Discussion

The analysis of the obtained results shows that the proposed design of the hydro turbine has the increased energy performance in longer hole drilling as against the series-produced bottom-hole hydromotors which demonstrate the decreased efficiency. The improvement is achieved due to:

Table 1. Calculation matrix

H—fluid feed height, m (hole depth)											
100				**300**				**700**			
Fluid incidence angle ∝ to horizontal plane, deg											
30°		**70°**		**30°**		**70°**		**30°**		**70°**	
Fluid flow rate, l/min											
15*	25	15	25	15	25	15	25	15	25	15	25
10**	20	10	20	10	20	10	20	10	20	10	20
*Q1—fluid flow rate in central channel of rotor; **Q2—fluid flow rate in side openings of stator.											

Table 2. Comparison of technical and energy performance of series-produced bottom-hole hydromotors and the proposed structure chart of hydro turbine

Model	Index						
	Technical			Energy			
	Body diameter, mm	Length, mm	Mass, kg	Fluid flow rate, l/s	Rotational torque, N·m	Capacity, kW	Efficiency η
TB123B-195	195	8060	1440	45–50	714.882	–	–
ZTS	195	26110	4470	32–54	5700	–	–
4C2-172,00 (China)	172	3200	–	6.0	2850	–	–
Hydro turbine	196	500	55	4–6	18830	112.316	0.282

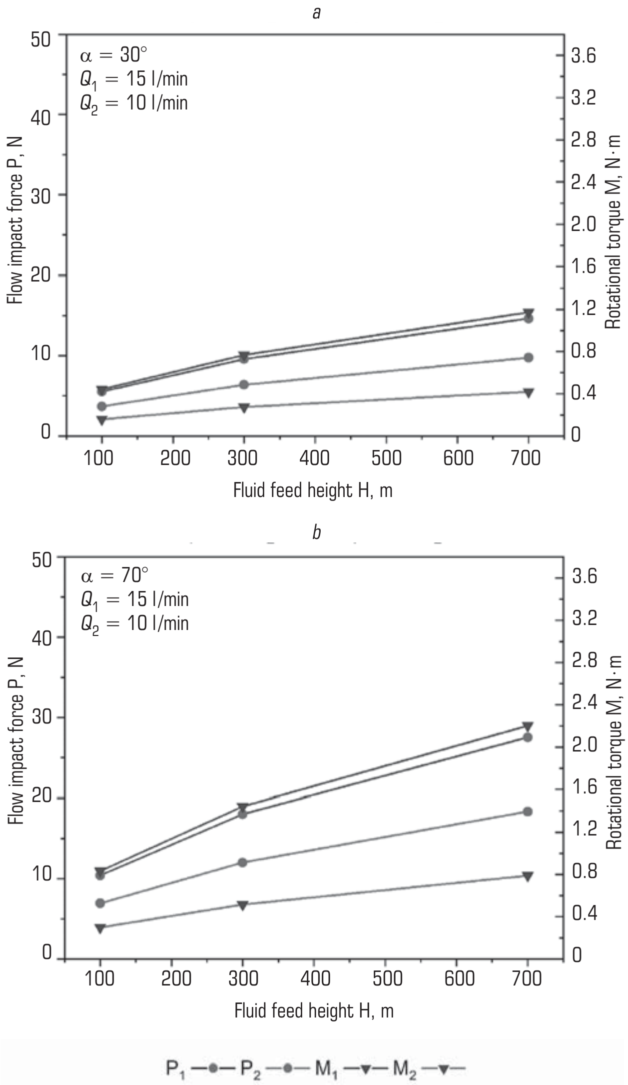

Fig. 3. Calculated energy performance of the structure chart of the hydro turbine depending on the hole length at the increase in the power fluid flow rate and in the flow incidence angle ∝ relative to horizontal plane:
a–different hole lengths and fluid flow rates at the flow incidence angle of 30°;
b– different hole lengths and fluid flow rates at the flow incidence angle of 70°

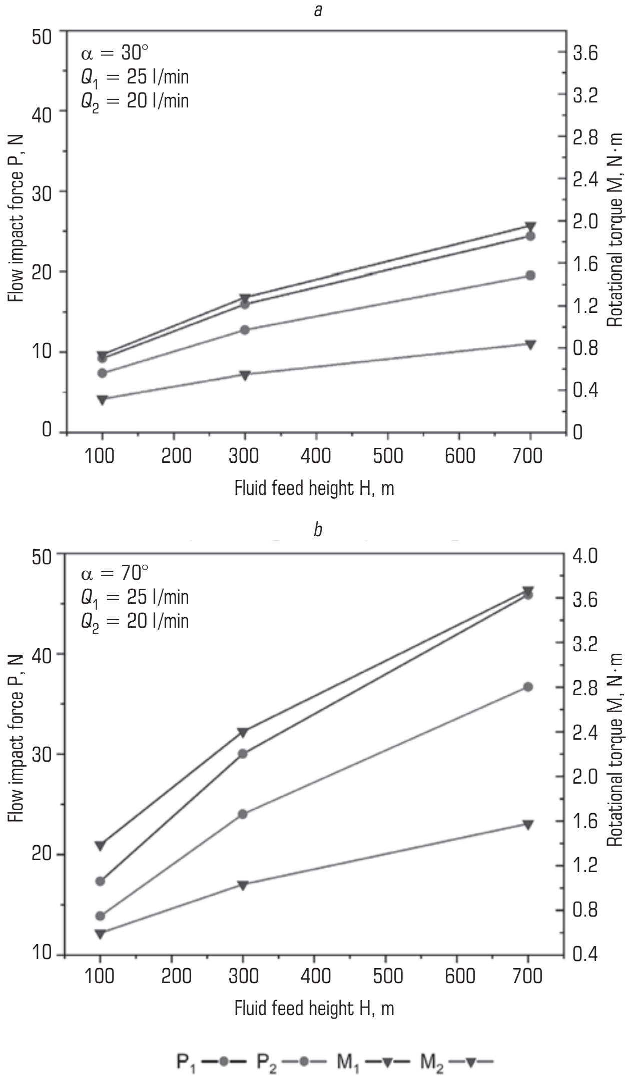

Fig. 4. Calculated energy performance of the structure chart of the hydro turbine depending on the hole length at the increase in the power fluid flow rate and in the flow incidence angle ∝ relative to horizontal plane:
a–different hole lengths and fluid flow rates at the flow incidence angle of 30°;
b– different hole lengths and fluid flow rates at the flow incidence angle of 70°

• The same direction of power fluID flow and rotational torque of the stator at the enlarged area of interaction between the flow and elements of the structure, which decreases energy loss;

• The inversion method when the rotating rotor is made immobile and the stator—mobile. This simplifies the structure, reduces metal consumption and increases the length of the force lever;

• The ejection effect when friction creates an entrained fluID flow, which increases the flow velocity and the hydrodynamic impact on the stator blades;

• The reactive jet of the fluID flow, which generates an additional rotational torque;

• The separate fluID feed and mud removal system, which reduces the rock resistance to fracture and improves the hole cleaning.

In this manner, the proposed design ensures the more efficient use of the potential energy of the fluID flow at the smaller size and lower flow rate of the machine.

The proposed hydro turbine ensures also the economic benefits, namely:

• The decreased operating cost—the decreased loss of power fluID allows reducing power consumption of the drill to 25%;

• The increased overhaul period—owing to the decreased load on the hydro turbine, its endurance increases by 30–40%.

• he decreased metal consumption—the decreased mass by 2–3 times saves the cost of manufacturing;

• The simplified assemblage and maintenance—the absence of complex reducing gearboxes facilitates operation.

The estimated cost of manufacture of the new hydro turbine is 30–40% of the cost of the series-produced turbodrill, which makes hydro turbine beneficial and competitive machine.

The proposed design opens prospects for the further research and pilot testing to prove the calculated data and to optimize the design for the real-life drilling conditions.

Conclusions

The use of the design engineering methods (inversion, linking and generality) allowed more efficient utilization of the capacities of the structure chart of hydro turbines, including the same directions of the power fluID flow and rotational torque in horizontal plane, the ejection effect and the jet exhaust converted to the rotational torque.

The result of such approach application was the increased length of the force lever, the simplified design and the decreased metal consumption, which promoted the better use of the potential energy of the power fluid.

Using the obtained results, a structure chart was developed for the small-size hydro turbines 0.5–0.6 m long, with the lower fluID rate of 90–120 l/min. As a consequence, the higher energy performance was achieved at the minimal power fluID flow rate.

The calculation of the energy performance provided by the new structure chart of the hydro turbine depending on the drilling length, and on the power fluID feed angle and flow rate proved their advantages over the series-produced hydromotors.

The hydro turbines demonstrate the higher energy performance in the longer drilling, while the series-produced bottom-hole hydromotors feature the higher power consumption in this case.

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