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ANALYSIS OF GEARING-UP DEVICES FOR HIGH-SPEED DIAMOND BIT DRILLING OF LONG EXPLORATION HOLES

In long-hole drilling the rate of penetration reduces as in this case much more energy is consumed to rotate drill strings, vibrations and oscillations of drill strings are higher, and drill pipes which often operate at rim of abilities break more frequently. All that implies that a drill string is a low-efficient transmitter of energy from drill rigs to bottomhole, especially in long-hole drilling with diamond bits.

It is possible to intensify rock drilling process by placing mechanical energy generator at bottomhole. To this effect hydraulic motors and electric drills are currently available. However, hydraulic motors need high discharge pressure, while rotation speed of drill bits and drilling fluid flow rate are difficult to control and adjust. In case of electric drills, delivery of electric energy to bottomhole is complicated.

All these constraints are overcome with the help of mechanical transmission represented by a bottomhole multiplying gear capable to speed-up diamond drill bits several times as against the drill string rotation.

Key words: *drilling tool, bottomhole, multiplying gear, hole, diamond bit, rotation speed, core drilling*
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Introductoin

Technologies of hard rock drilling enjoy booming development both in surface and underground mining these days [1]. The growing demand for such technologies is explained by the application of advanced process solutions in surface and underground mining of hard minerals, coal and rare earth metals, in civil construction, etc. [2–11]. Recently diamond bits gain an increasing popularity as they extend service life of rock-breaking tools, increase penetration rates, reduce lowering/lifting operation time and, thus, displace rolling cutter drill bits.

The research in the area of hard rock destruction by diamond tools, as well as the experience gained in drilling and natural stone processing specifies that the potential rise in the penetration rate is first and foremost connected with an increase in the diamond drill bit speed. At the same time, the higher speed of drill bits and, accordingly, drill strings, is restricted by the drilling conditions in reality. For instance, in long-hole drilling (to 1000 m and more) with a diameter of 76 of 59 mm using a diamond rock-breaking tool, the drill string rotations per minute are no more than 350 min^{-1}. In case of the higher rotation speed, the risk of drill string breakage grows, and drilling consumes much more energy to overcome the drill hole wall friction. The linear cutting speed is no more than 1–1.5 m/s, which is known to be insufficient for a diamond tool (the recommended speed for impregnated diamond drill bits is 2–5 m/s) [12].

It is even impossible to solve the problem with the increasingly popular coiled tubing as the use of very long flexible pipes needs a bottomhole hydraulic actuator which requires high discharge pressure; moreover, it is difficult to control rotation speed of drill bit and flow rate of drilling fluid. Electric drilling needs an electric power supply [13, 14].

A way out may be the use of mechanical transmission as a bottomhole speeder capable to increase diamond bit speed several times as compared with the drilling string rotation.

Analysis of gearing-up devices for high-speed diamond

The machine should be reliable and high-performance while low metal- and energy-consuming, and should comply with engineering ergonomics [15]. These require-

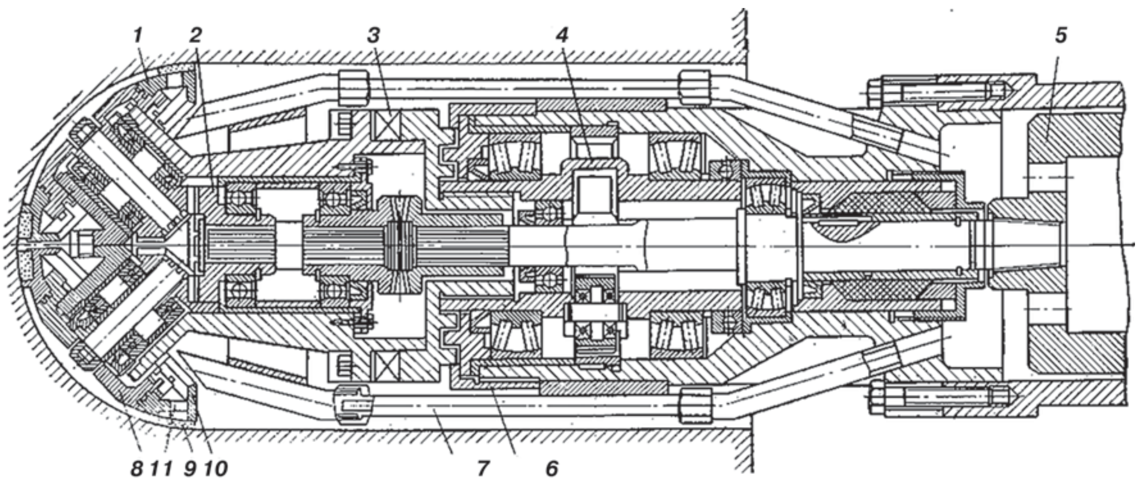

Fig. 1. Diamond epicyclic drill bit with a turbine type drive:
***1* — cutting elements; *2* — distribution gearbox; *3* — jaw coupling; *4* — epicyclic reduction gear; *5* — drive housing; *6* — tooth-type coupling; *7* — cooling fluid lines; *8* — cutting hub; *9* — diamond inserts; *10* — conic partitions; *11* — holes**

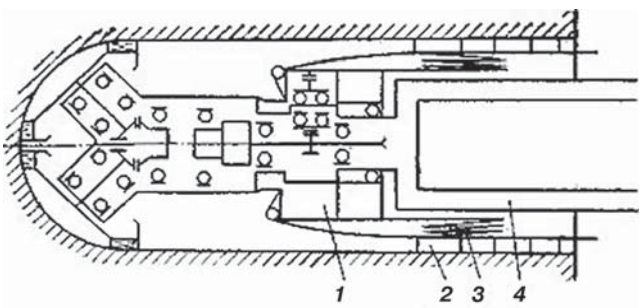

Fig. 2. Kinematic chart of diamond epicyclic drill bit:
***1* — epicyclic reduction gear; *2* — braking elements; *3* — spring reed; *4* — driving rod**

ments are valid for the mechanical transmissions that coordinate operating conditions of engine and rock-breaking tools [16].

Thus, the required mechanical transmission should provide:

1) high capacity at small diameter (less than 100 mm);
2) wide-range kinematic capabilities;
3) low cost of manufacture, assembly and repair.

First of all, it is advisable to consider application of gear transmissions as they meet the above-listed requirements to a certain extent.

The Karaganda Polytechnic Institute has designed a diamond epicyclic drill bit for drilling in rocks with the uniaxial compression strength more than 100 MPa.

The diamond epicyclic drill bit with a turbine type drive (**Fig. 1**) with the rock cutting elements made as truncated spheres with diamond inserts arranged along periphery is distinguished for the efficient cutting speed in a range from 30 to 50 m/s. This governs small cutting force and, consequently, low torque on the working shaft, which makes it possible to assemble powerful and small-size epicyclic tools. The calculations and design show that such tools can only drill holes with a diameter over 0.15 m and more.

It is impossible to use such tools in core drilling because of the center-positioned distribution gearbox intended to transmit rotation to cutting elements.

Fig. 2 shows a kinematic diagram of a diamond epicyclic drill bit powered through the epicyclic reduction carrier using a rotating rod. Such design ensures operability of the diamond epicyclic drill tool with the currently available rotary drilling equipment [17].

The required cutting speed for efficient drilling and penetration is achieved owing to fixing the epicyclic reduction gear body in the hole by the braking elements. In this case, the epicyclic reduction gear ensures relative movement and speed-up of the tool while the tool is advanced owing to rotation of the driving rod. The braking elements are expanded in the hole by the spring reeds which create forces required for the break torque with regard to a possible change in the hole diameter in case of the tool wear.

In breakage of sandstone type rocks with the uniaxial compression strength of 80–100 MPa, the energy of rock destruction by the diamond tool makes 226–360 MJ/m^3 [18]. This means that, given installed power, the currently available drill rigs with the proposed diamond tool allow 2–4 times higher penetration rate at the same material inputs. Comparatively low cost of the impregnated diamond tool and its high durability ensures saving. Penetration rate is higher, forces of cutting and feed are lower, while the weight of the drilling equipment is considerably lighter, which reduces its cost and improves maneuverability.

The discussed diamond epicyclic drill bits have complex design and can only be used in the holes with a diameter of 0.15 m and more, whereas application of a bottomhole speeder is more of interest at this time for the high-frequency diamond drilling of small-diameter holes as in the latter case the cutting speed is too low for a diamond tool (i.e., when the hole diameter is decreased at the unchanged rotation speed, the diamond drill speed is lowered too). Furthermore, in the design of a diamond epicyclic drill bit with the adjustable axial load, it is impossible to create the thrust force of the braking elements on the hole walls, which limits their application.

When designing a bottomhole speeder for the small-diameter high-frequency drilling [19], a wave gear was at first

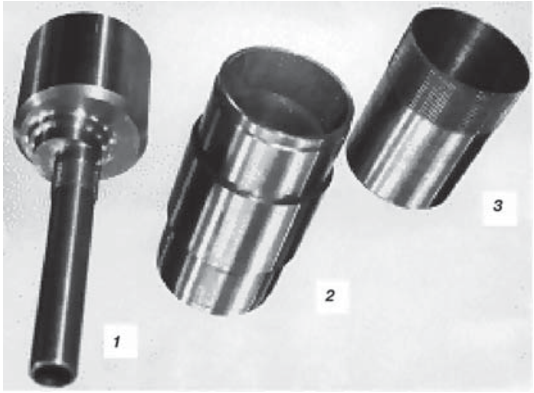

Fig. 3. Wave gear parts:
***1* — wave former; *2* — rear gear group; *3* — flexible gear**

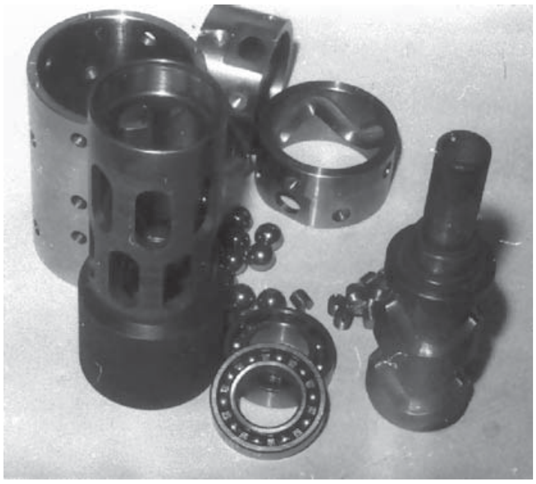

Fig. 4. Breadboard model multiplying gear parts

examined. The information about such gears first appeared in 1959 [20]. In USA and Japan, batch production of general-duty wave gears is set up. Russia also produces wave gears in series, starting from the diameter of 50.8 mm (diameter of pitchline of flexible element) [21, 22]. The wave gear is advantageous over the epicyclic gear for the: symmetric design and, as a consequence, small loads on shafts and supports; wider range loading; long endurance; mush smaller size at the same torque on the output shaft [23].

Using a dedicated program, the breadboard models of wave gears were designed and manufactured for the experimental drilling.

Fig. 3 demonstrates the parts of the wave gear subjected to the benchmark testing. At the output shaft rotation of 24 min^{-1} (high speed shaft rotation of 1920 min^{-1}), the flexible gear 3 broke by cracking between the gear teeth.

The accomplished theoretical and experimental research has revealed disadvantages that prevent application of wave gears as the bottomhole speeder, namely: high bottom limit of transmission ratio of flexible steel gear; dependence of the capacity on the flexible gear strength; difficulty and high accuracy of manufacture of flexible gears, which complicates repair.

In the further theoretical and experimental research of the known mechanical transmissions to act as the bottomhole speeders, the radically new devices have been found—alternating roller gears. These gears are composed of intermediate members arranged so that their shafts are coaxial (**Fig. 4**). As against bearing (roller, friction) gears, the selected gears have much higher load-bearing capacity, stable transmission ratio and higher efficiency. They are much simpler than tooth gears and cover the transmission ratio range unreachable for the wave gears.

Efficiency of the breadboard model of the alternating roller speeder with a diameter of 73 mm was also tested [23]. For the tests, the required coupling parts and a cushioning spring to measure the moment M_3 on the body of the multiplying gear were manufactured.

Brief description of the test bench.

(1) Drive shaft (carrier) of the multiplying gear is rotated by the output shaft powered by asynchronous motor with the connected car gear box. (2) The loading device for the multiplying gear is the asynchronous motor with its rotor connected to the shaft (high speed) of the speeder. (3) The motor body is fixed in bearings and balanced by loads. (4) The asynchronous motor is run in the mode of dynamic slowdown. Direct current is delivered to three motor windings. (5) Oil-cooled transformer. (6) DC/AC thyristor converter. (7) Voltmeter gauge and current meter.

Test conditions

1. Input shaft speed—300 min^{-1}. 2. Output (high speed) shaft speed—300·2.33=700 min^{-1}. 3. One test duration without cooling—10 min.

Test sequence

After actuation of the input (main) shaft (carrier) of the multiplying gear and deceleration of the output (high speed) shaft by the loading motor, torques were measured on the bodies of the speeder and the loading motor, M_3 and M_1, respectively.

Efficiency of the multiplying gear was calculated from the formula:

$$\eta = M_1 / (M_1 + M_3) \cdot u,$$

where u — the transmission ratio of the speeder (the breadboard model had u = 2.33).

The test results are compiled in **Tables 1** and **2**. Plotted using the test data, the relationship between the multiplying gear efficiency and transmitted torque (**Fig. 5**) shows that the increasing transmitted torque enhances the speeder efficiency. The analysis of the data yields that ini-

Table 1. Testing data of the alternating roller gear with the transmission ratio of 2.33—Grafitol lubricant

Test No.	M_1, N·m	M_3, N·M	Efficiency
1	6	12	0.77
2	8	15	0.81
3	11	21	0.80

Table 2. Testing data of the alternating roller gear with the transmission ratio of 2.33—TAD-17m lubricant

Test No.	M_1, N·m	M_3, N·M	Efficiency
1	6	10	0.86
2	8	13	0.88
3	10	16	0.90

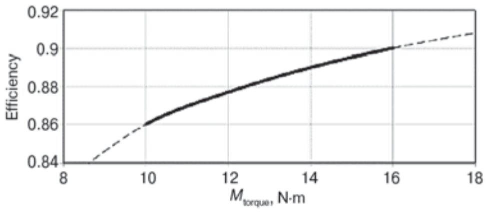

Fig. 5. Relationship of the speeder efficiency and the transmitted torque

tially, at the minimum torque on the main shaft of the alternating roller gear, the boll touches the race track at points of the manufacturing errors while under the increasing torque these contacts become more linear (along segments of a circle). This happens as a consequence of elastic deformation of metal at the loaded points and due to the running-in phenomenon.

Regarding efficiency of the speeder, as the total torque on the main shaft of the multi-row alternating roller gear grows, the torque distribution nonuniformity between the sections will decrease until the sections are loaded equally. Evidently, the efficiency will stop growing and stabilize at the maximum level.

Conclusion

The objectives met in the course of the experimental research include:

— verification of the serviceability and applicability of the alternating roller gear as a speeder;

— basic engineering solutions to manufacture a multi-row alternating roller speeder with a diameter of 73 mm (with the preserved design and operation simplicity of the device);

— validation of a full-scale operative rotary drilling speed-up gear.

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