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ACCESSORIES AND DEVICES

DEVELOPMENT OF A HIGH-POWER PUMP UNIT FOR HYDRAULIC FRACTURING OF DAMS IN ISOLATED OIL AND GAS POCKETS

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The design of a mobile pump unit having a diesel ICE with an effective power up to 3000 hp, a double-flow high-multispeed mechanical gearbox, and a hydraulic pump with a maximum working pressure of 105 MPa is proposed. The development is aimed at improving operation and efficiency, increasing reliability, and simplifying design of the pump unit.

Keywords: diesel ICE, high-multispeed double-flow mechanical gearbox, high-pressure hydraulic pump.

In drilling development of hydrocarbon deposits, as much as 20% of oil and gas remains undeveloped in isolated underground oil and gas pockets. For development of such deposits, use is made of the technique of deep-penetration hydraulic fracturing of reservoir dams by high pressure of the working fluid, which connects the isolated pockets into a single pocket, the efficiency of crude extraction from which is close to 100% [1].

One of the basic tasks of creation of a mobile system of special technique for hydraulic fracturing of dams of isolated pockets (by high pressure of the working fluid) is designing high-power pump unit.

The main devices in pump units of mobile systems for hydraulic fracturing of dams of isolated oil and gas pockets are diesel ICE, gearbox, stopping brake, and hydraulic pump.

In mobile systems for deep-penetration hydraulic fracturing of oil and gas pocket dams, most common are pump units with Allison type of planetary hydromechanical gearbox, which have some demerits: inadequate efficiency (due to use of ICE with limited power and hydraulic pump with low working pressure), high power loss of the ICE when hydraulic transformer is used in the gearbox (consequently low efficiency), design complication, low reliability of the power unit, etc. Therefore, for oil and gas production it is urgent to create high-power domestic pump units of mobile systems for deep-penetration hydraulic fracturing of dams of isolated oil and gas pockets by high pressure of working fluid.

The goal of this work was to raise the output and efficiency of the pump unit, increase the reliability of the unit, and simplify its design.

For improving the characteristics of the pump unit, a promising design of the power drive that includes a double-flow high-multispeed mechanical gearbox with a stopping brake (with modern copper- and iron-based frictional metal-ceramic materials and with automatic control without disruption of the power flow), which raises the planetary reducer (with halting epicycle) at the gearbox inlet and lowering the planetary reducer (with a mounted epicycle) at the gearbox outlet is proposed.

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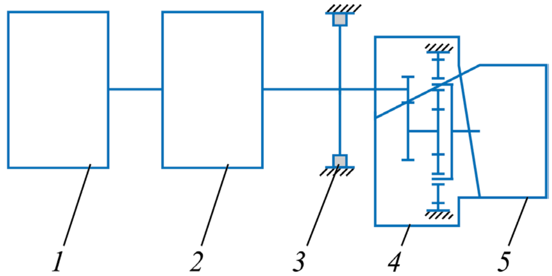


Fig. 1. Block diagram of pump unit for hydraulic fracturing of dams in isolated oil and gas pockets: *1* — ICE; *2* — gearbox; *3* — brake; *4* — cylinder-planetary reducer; *5* — hydraulic pump (total gear ratio of built-in hydraulic pump reducer $4i = 6.353$; gear ratio of cylindrical stage of the reducer $i_c = 1.16$ (inter-center distance 500 mm as per design of the unit); gear ratio of planetary stage of reducer $i_p = 5.5$).

Prerequisites for Designing the Pump Unit:

– ensuring high pressure of working fluid in hydraulic fracturing pocket;

– effective ICE power up to 3000 hp, axle rotation speed $n = 1800$ rpm;

– hydraulic pump with a cylinder-planetary reducer, maximum working fluid pressure up to 105 MPa;

– maximum hydraulic pump shaft rotation speed 300 rpm;

– hydraulic pump speed control using remote gear shifting (with roughly equal shifting steps) without power interruption during operation of the unit under stress.

The block diagram of the initial (for designing) power unit of the pump system for hydraulic fracturing of dams in isolated oil and gas pockets are shown in Fig. 1.

In the developed kinematic scheme of the power unit of the pump system (Fig. 2), the diesel ICE with effective power up to 3000 hp is connected through the planetary reducer *2* having a fixed epicyclic gear and a gear ratio $i = 0.25$ (chosen because of limitation to maximum antifriction bearing rotation) with the input shaft *6* of the double-shaft automatic gearbox *3* with a summing planetary gear set [2]. In the summing planetary gear set, the main power flow is fed to the epicyclic gear *5* connected with the load shaft *27* of the gearbox. Additional power flow from the input shaft *6* of the gearbox is fed into the sun gear *4* of the summing planetary set.

The speed of the two-shaft double-flow 10-speed high-speed gearbox *3* is reversed without interruption of power flow using the friction devices with metal-ceramic disks (from copper-based MK-5 baked powdery material) [3].

One friction component in the main power flow (to epicycle *5* of summing planetary set) and one friction component in the additional power flow (to sun gear *4* of the summing planetary set) are connected for switching on the transmission in the gearbox. The total power flow is removed from the carrier *21* of the summing planetary gear set. In the power drive with switched-on transmission in the gearbox the epicyclic gear, sun gear, and carrier of the summing planetary gear set rotate in the same direction (for exclusion of parasitic circulating power in the closed kinematic circuit). For start and stop of the power drive, OT-24 disk brake with metal-ceramic dry-friction lining (made of iron-based SMK-137 baked powdery material) is used.

The gear ratio in the first gear I (Fig. 2) in the case of stopped load shaft *27* of the gearbox is determined by the equation $i_{\mathrm{I}} = i_{\mathrm{d}}(1 + K)$, for the first slowed transmission $i_{\mathrm{I\,slow}} = i_{\mathrm{d.slow}}(1 + K)$, where i_{d} and $i_{\mathrm{d.slow}}$ — gear

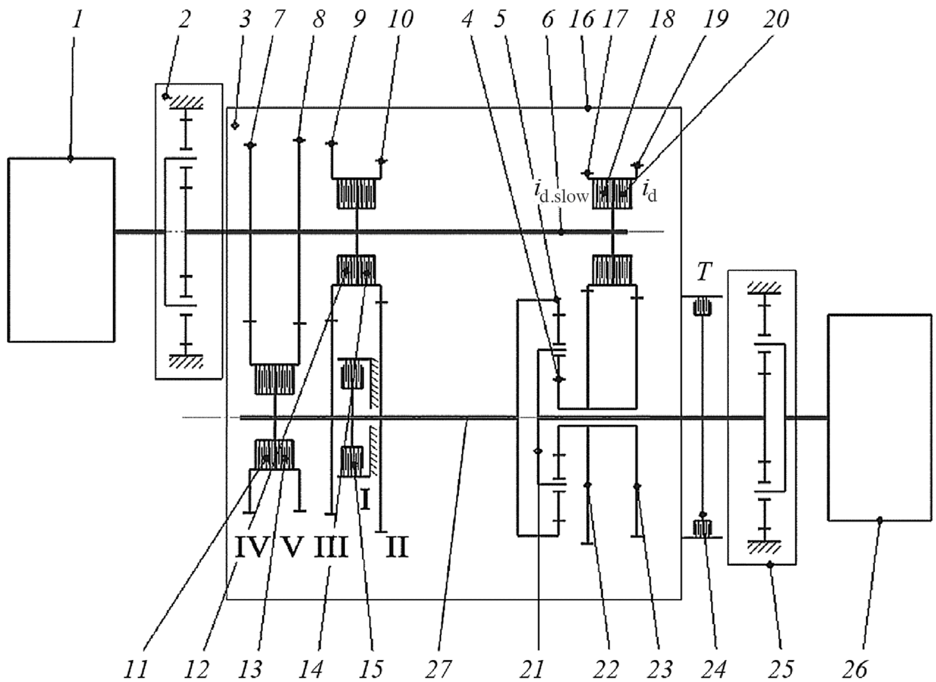


Fig. 2. Kinematic scheme of pump unit with high-speed double-shaft double-flow mechanical gearbox: *1* — diesel ICE; *2* — planetary reducer speed accelerator; *3* — 10-step gearbox; *4* — sun gear of summing planetary set; *5* — epicyclic gear of summing planetary set; *6* — input shaft of gearbox; *7–10* — cog couples with cylindrical gears of the main power flow; *7–14* — friction couples; *15* — lower gear brake; *16* — gear casing; *17, 19, 22, 23* — couplings of cylindrical gears of additional power flow; *18, 20* — friction couples of additional power flow; *21* — carrier of summing planetary gear set; *24* — stopping brake; *25* — planetary reduction gear; *26* — hydraulic pump with built-in cylinder-planetary reducer; *27* — gearbox load shaft.

ratio of the drive to sun gear *4* of the summing planetary gear set; K — parameter of summing planetary set of the gearbox (internal gear ratio).

The gear ratios in the second–fifth (II–V) gears in the gearbox are determined by the equation:

$$i_{\mathrm{II-V}} = i_{\mathrm{d}} \cdot i_{K_i} \frac{(1+K)}{i_{K_i} + i_{\mathrm{d}} K},$$

where i_{K_i} — gear ratio of the cylindrical coupling of the gear of the main power flow to the epicyclic gear *5* (Fig. 2) of the summing planetary set in the ith transmission of the gearbox.

The gear ratio in the second slow–fifth slow transmissions in the gearbox is determined by the equation:

$$i_{(\mathrm{II-V})\,\mathrm{slow}} = i_{\mathrm{d.slow}} \cdot i_{K_i} \frac{(1+K)}{i_{K_i} + i_{\mathrm{d.slow}} K}.$$

The coefficient of moment distribution between the epicyclic and sun gears of the summing planetary gear set of the gearbox in the ith gear (proportion of power from the ICE transmitted by the main flow) is determined by the equations:

$$\beta = \frac{i_{\mathrm{d}} K}{i_{K_i} + i_{\mathrm{d}} K}, \qquad \beta_{\mathrm{slow}} = \frac{i_{\mathrm{d.slow}} K}{i_{K_i} + i_{\mathrm{d.slow}} K}.$$

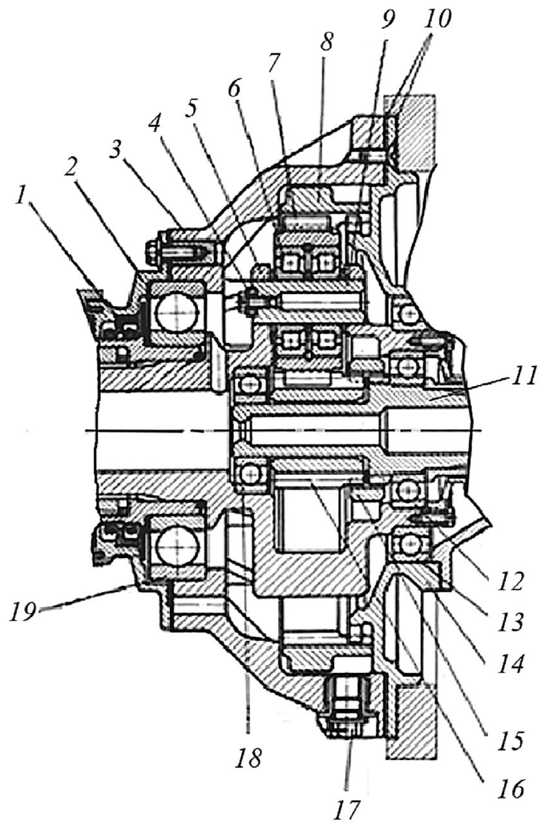


Fig. 3. Block diagram of planetary reducer: *1* — cup packing; *2* — cover; *3* — casing; *4* — plug; *5* — carrier; *6* — bearing; *7* — satellite; *8* — epicycle; *9* — axle; *10* — lining; *11* — shaft; *12* — bearing; *13* — bearing; *14* — casing; *15* — bushing; *16* — sun gear; *17* — bolt; *18*, *19* — bearings.

In the cited equations, the gear ratio i is the ratio of the rotation speed of the driving components of the drive to the rotation speed of the driven components of the drive.

Block diagram of the input planetary gear reducer that enhances the ICE speed is shown in Fig. 3 (at the gearbox exit a similar reducer is installed as a speed limiter when the epicycle of the planetary gear is at rest).

Block diagram of the high-speed double-shaft double-flow mechanical gearbox is shown in Fig. 4 (gear ratio of the gearbox for transmissions: $i_{\mathrm{I}} = 3.7$; $i_{\mathrm{II}} = 3.0$; $i_{\mathrm{III}} = 2.4$; $i_{\mathrm{IV}} = 1.8$; $i_{\mathrm{V}} = 1.6$; $i_{\mathrm{VI}} = 1.4$; $i_{\mathrm{VII}} = 1.2$, $i_{\mathrm{VIII}} = 1.1$; $i_{\mathrm{IX}} = 1.0$; $i_{\mathrm{X}} = 0.85$ [4].

Also developed is a block diagram of disk stopping brake [5] for a high-capacity power drive (Fig. 5) that is mounted on the gearbox casing.

Based on the proposed kinematic scheme (Fig. 2) a 3D model of the power drive of the pump unit of the mobile system has been developed for deep-penetration hydraulic fracturing of oil and gas pockets (Fig. 6).

In the developed model (Fig. 6), the gears are shifted with stopping of the load shaft of the gearbox (first and first slow gear) using toothed clutch. A three-dimensional model of power drive is required for tentative solution of many technological issues for making the proposed power drive, for example, for determining mass-centering and inertia characteristics of the drive.

The developed gearbox is a high-speed (up to 8000 rpm on gearbox components depending on the anti-friction bearing operation conditions) and is therefore compact, the friction components of the gearbox are controlled with reduced load indices for ensuring reliable operation of the gearbox in combination with high-power ICE.

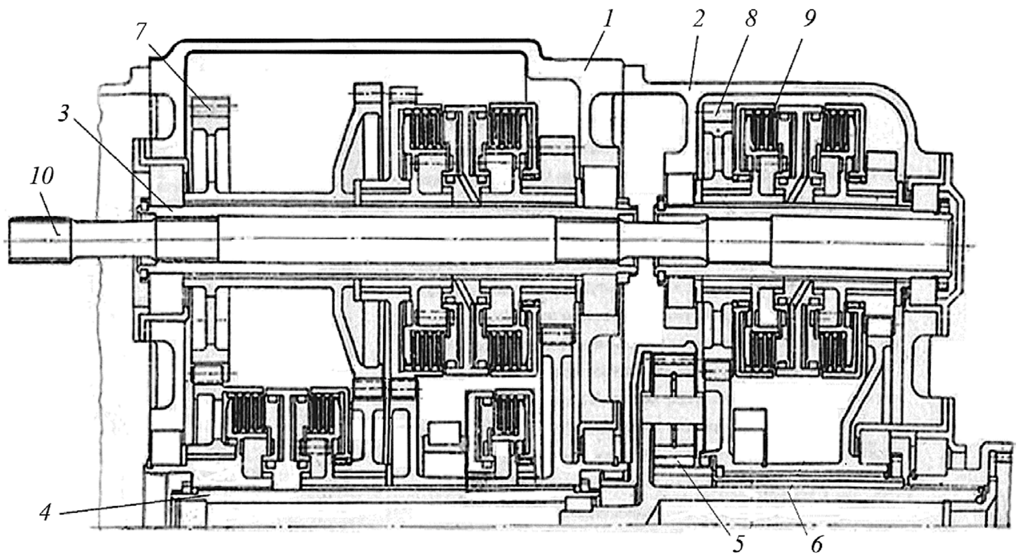


Fig. 4. Block diagram of high-speed double-shaft double-flow mechanical gearbox: *1* — crankcase; *2* — casing; *3* — intermediate shaft; *4* — load shaft; *5* — summing planetary set; *6* — driven shaft; *7* — gear transmission of main power flow; *8* — gear transmission of additional power flow; *9* — friction clutch; *10* — torsion shaft.

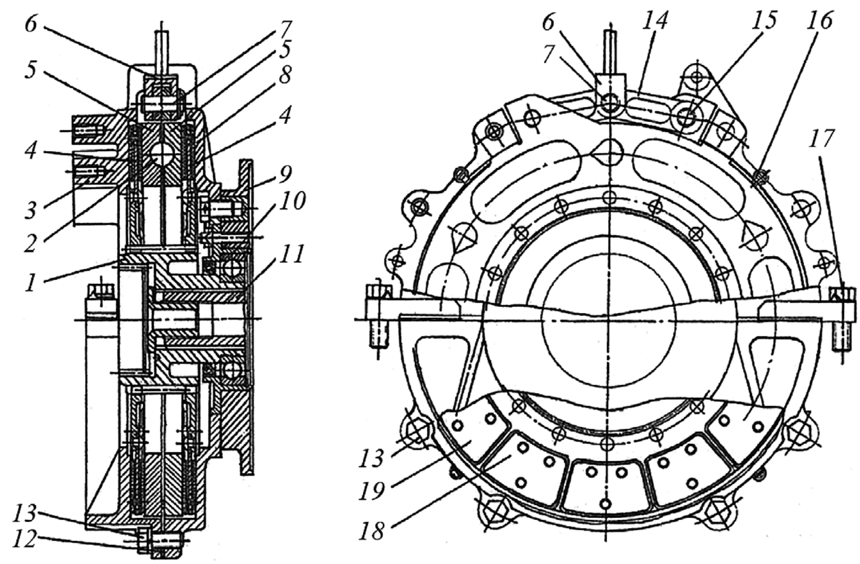
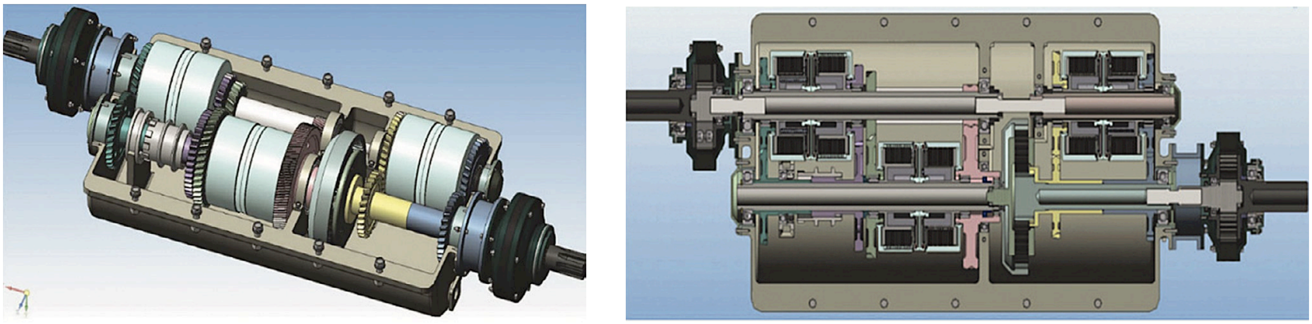


Fig. 5. Block diagram of disk stopping brake of power drive: *1* — drum; *2* — ball; *3* — cover; *4* — friction disk; *5* — press-disk; *6* — pinion; *7* — axle; *8* — casing; *9* — gearbox cover; *10* — bearing; *11* — gearbox shaft; *12* — adjusting gasket; *13* — bolt; *14* — lever; *15* — axle; *16* — spring; *17* — yoke bolt; *18* — rivet.

The developed compact mechanical power drive is structurally simple, has a high efficiency (compared to hydromechanical transmissions) and operationally reliable (thanks to the new design of the power drive, use of high-speed friction devices with new friction materials).

The developed design of the power drive as a component of the pump unit can be used to enhance the efficiency of the unit with use of ICE with power up to 3000 hp and hydraulic pump with maximum working fluid pressure up to 105 MPa. The proposed power drive is promising for further enhancement of the ICE power (for example, up to 5000 hp).



(a) (b)

Fig. 6. 3D model of power drive of pump unit with a high-speed double-shaft double-flow mechanical gearbox having planetary gear reducers: (a) — general view without the top part of the gearbox casing; (b) — sectional view in joint plane.

Use of the proposed power drive as a component of the mobile system for deep-penetration hydraulic fracturing of oil- and gas-bearing formations is promising for heightening oil and gas production efficiency.

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