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ISRA (India)	= **6.317**	**SIS** (USA)	= **0.912**	**ICV** (Poland)	= **6.630**
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JIF	= **1.500**	**SJIF** (Morocco)	= **7.184**	**OAJI** (USA)	= **0.350**

SOI: 1.1/TAS DOI: 10.15863/TAS
International Scientific Journal
Theoretical & Applied Science

p-ISSN: 2308-4944 (print) **e-ISSN:** 2409-0085 (online)

Year: 2023 **Issue:** 06 **Volume:** 122

Published: 09.06.2023 **http://T-Science.org**

Issue **Article**

Gulrukhsor Sobirjonovna Zakiriyayeva
Navoi State Mining and Technological University
Master student of the Department of Mechanical Engineering Technology, Navoi, Uzbekistan

Rashid Olimovich Muminov
Navoi State Mining and Technological University
Docent of the department Mechanical Engineering, Navoi, Uzbekistan
rashid_81@mail.ru

RESEARCH AND SELECTION OF THE PARAMETERS OF THE ROTARY-FEEDING MECHANISM OF A QUARRY DRILLING MACHINE

***Abstract**: The article analyzes the design, kinematic and power parameters of the rotational-feeding mechanism of the drilling rig.*

***Key words**: Drilling rig, rotary feed mechanism, design, kinematic and power parameters.*
***Language**: English*
***Citation**: Zakiriyayeva, G. S., & Muminov, R. O. (2023). Research and selection of the parameters of the rotary-feeding mechanism of a quarry drilling machine. *ISJ Theoretical & Applied Science, 06 (122)*, 64-68.
***Soi**: http://s-o-i.org/1.1/TAS-06-122-13 ***Doi**: crossref https://dx.doi.org/10.15863/TAS.2023.06.122.13
***Scopus ASCC**: 2200.*

Introduction

Roller drilling machines are the most common in the preparation of rock mass for excavation in open-cast mining, due to their versatility, which ensures efficient drilling of wells in a wide variety of mining and geological conditions.

Currently, the most popular machines operating at mining enterprises are machines 2SBSH - 200, 2SBSH - 200N (see Fig. 1), SBSH - 250MN. When drilling blastholes in complex structural rock masses, one of the main drawbacks inherent in the cone drilling method is the increased vibration of the drilling string, which forces the machinists to operate the machines at modes that are underestimated compared to rational ones.

Vibration causes the formation of fatigue cracks and breakage of structural elements, leads to failure of the equipment installed on the frame of the machine, has a harmful effect on maintenance personnel and increases the cost of maintaining the machines. With an increase in the power-to-weight ratio and dynamic loading of the drive, energy losses also increase. For example, according to the authors of [1], with strong vibrations of a drilling rig, the share of energy expended to create a useful torque is 30-50%. As a result, a significant part of the installed drive power of the machine remains underused.

One of the main reserves for increasing the efficiency of drilling cone rigs is the intensification of drilling modes, which is significantly hindered by vibration and dynamic loads that occur during drilling. There are various devices for reducing vibrations and dynamic loads in the elements of drilling rigs of both spindle and cartridge schemes: an automatic control system for drilling modes according to the level of vibrations, over chisel and over rod shock absorbers, drill string stabilizers.

Figure 1. Quarry drilling rig 4SBSH - 200 - 40 with cartridge-type VPM.

The use of these devices helps to reduce vibrations and loads in the elements of the drilling rig, however, these devices have not found wide application due to low efficiency and reliability. Their use is mainly aimed at reducing the level of vibration in the vertical plane and there are practically no devices that reduce the horizontal vibrations of the machine.

A further increase in the efficiency of the extraction of mineral raw materials is possible on the basis of the technical re-equipment of the extractive industries of the national economy. Increasing the productivity and reliability of mining machines, in particular drilling rigs, has determined the need to increase their power-to-weight ratio and improve technical and economic indicators.

The tasks set can be solved primarily on the basis of improving the drive of machines, and in some cases by creating fundamentally new designs of drives [3].

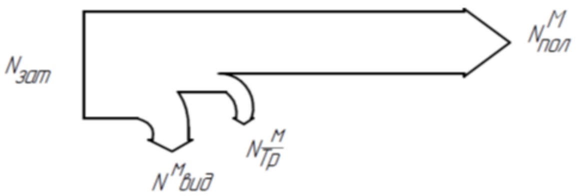
As a result of the research conducted in it was found that the best performance has a volumetric hydraulic drive with high-torque hydraulic motors. The wide hydrofication of drilling rigs opens up the possibility of a qualitative improvement in their dynamic and energy characteristics. An important property of a volumetric hydraulic drive is the possibility of using elastic-damping devices in its hydraulic system, mainly pneumohydraulic accumulators, which can significantly change (correct) the dynamic characteristics of the entire drilling rig. Such a correction of its properties is possible not only during the design process, but also during the adjustment or operation of the machine in various modes of drilling blast holes.

Based on the analysis of the results of work in the field of studying the dynamics of drives, a fundamentally new design of the hydromechanical transmission of the drive of the executive bodies of mining machines is presented, including a differential mechanism with a volumetric hydraulic brake [3].

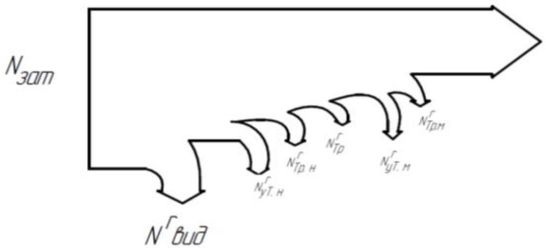
The proposed design (with a power of 50 kW) passed bench tests, as a result of which it was found that it most fully satisfies modern requirements for transmissions of mining machines and at the same time allows you to save the advantages characteristic of a volumetric hydraulic drive, which is based on double energy conversion - mechanical into hydraulic (pump - hydraulic motor), as well as to get new advantages over the traditional hydraulic drive, namely:

- no double conversion of energy;
- direct savings in terms of the installed capacity of hydraulic machines (~ twice);
- a sharp increase in the resource of the hydraulic machine (to order) due to its operation in the braking mode. [7].

The hydraulic machine of the hydromechanical rotator performs the functions of a hydraulic, and in the case of the use of pneumohydraulic accumulators, a pneumohydraulic spring with stiffness and damping adjustable over a fairly wide range. In a volumetric hydraulic drive, with double conversion of energy in the pump and motor and its transmission through pipelines, irreparable losses for leakage and friction are formed, reaching up to 40% (Fig. 2,.a). In the proposed design of the hydromechanical rotator of the executive body of the drilling rig 2SBSH-200MN, of all the above power losses, there are only losses associated with leaks determined by gaps in the brake hydraulic machine and operating pressure. Friction power losses in braking mode, determined by relative slip, are negligible, and friction losses in the pipeline in operating mode are completely absent (Fig. 2b).



$$N^{M}_{\text{виб}} = (0{,}3 \div 0{,}4)N_{\text{зат}}\,;\ N^{M}_{\text{тр}} = (0{,}05 \div 0{,}15)N_{\text{зат}};\ N^{M}_{\text{пол}} = (0{,}3 \div 0{,}5)N_{\text{зат}}$$



$$N^{\text{г}}_{\text{виб}} = (0{,}2 \div 0{,}32)N_{\text{зат}};\ N^{\text{г}}_{\text{ут.н}} \cong N^{\text{г}}_{\text{ут.м}} = (0{,}3 \div 0{,}08)N_{\text{зат}};\ N^{\text{г}}_{\text{тр}} \cong (0{,}02 \div 0{,}05)N_{\text{зат}};\ N^{\text{г}}_{\text{тр.н}} \cong N^{\text{г}}_{\text{тр.м}} = (0{,}05 \div 0{,}1)N_{\text{зат}};\ N^{\text{г}}_{\text{пол}} = (0{,}45 \div 0{,}6)N_{\text{зат}};$$

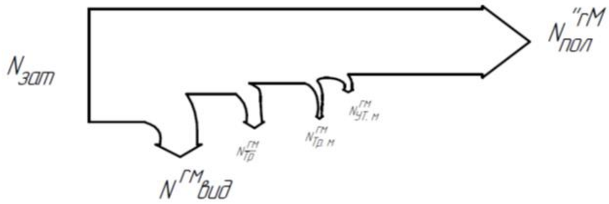


Figure 2. Power balance of the rotator drive during the working process a) electromechanical drive, b) electrohydraulic drive, c) electrohydromechanical drive.

$$N^{\text{гм}}_{\text{виб}} = N^{\text{г}}_{\text{виб}} = (0{,}2 \div 0{,}32)N_{\text{зат}};\ N^{\text{гм}}_{\text{тр}} = N^{\text{м}}_{\text{тр}}(0{,}05 \div 0{,}15)N_{\text{зат}};\ N^{\text{гм}}_{\text{трм}} < 0{,}005N_{\text{зат}};\ N^{\text{гм}}_{\text{утм}}(0{,}03 \div 0{,}08)N_{\text{зат}};\ N^{\text{гм}}_{\text{пол}} = (0{,}45 \div 0{,}715)N_{\text{зат}}\,.$$

The similarity of the dynamic characteristics of a hydromechanical transmission and a volumetric hydraulic drive is achieved, for example, by the identity of the volumes of fluid under operating pressure.

The design of the IMP2.5 hydraulic motor, which is one of the basic models of the size range developed at the Federal State Unitary Enterprise NNTsGP "IGD im. A.A. Skochinsky" [1, 2] provides for a radial arrangement of piston groups, each of which consists of two pistons, in the transverse holes of which the ends of the traverse are inserted. Rollers are installed on the traverse, on which clips are put on. The traverse is made of equal strength, has smooth transitions, due to which stress concentrators are eliminated in the areas of its greatest loading. [6]

The force from the pressure of the working fluid on the plunger is perceived by the traverse, and is transmitted through the rollers and clips to the profiled guide.

As a conjugation, limiting the resource of the radial-piston hydraulic machine IMP2.5, a plunger-cylinder friction pair is taken, as a result of which wear, the efficiency of the entire hydraulic machine can sharply deteriorate [2].

The specific work in the friction pair [4] "plunger-cylinder" is:

$$A = NT,\ \text{N/m} \qquad (1)$$

where N is the specific friction power, N/ms; T - interface resource, s.

The maximum specific friction power in the interface of the resource-limiting hydraulic machine is determined by the formula:

$$N = V_{\max}\, f\, [\sigma_{см}],\ \text{N/ms} \qquad (2)$$

where: max $V_{\max}$ - average maximum sliding speed of the plunger relative to the cylinder, m/s; f is the coefficient of friction in the plunger-cylinder pair; $[\sigma_{см}]$ - allowable contact pressure in the hydraulic machine piston group, N/m^2.

For one design of a hydraulic machine capable of operating in various modes (pumping, motoring, braking, and others) up to the maximum allowable wear of the interface limiting the life of the machine, the following equality is true:

$$N_б\, T_б = NT \qquad (3)$$

where: $N_б$ and $T_б$ - respectively, the specific friction power and the resource of the hydraulic machine during its operation in the base mode.

The "motor" mode is taken as the basic mode of operation of the IMP2.5 hydraulic machine.

According to the Federal State Unitary Enterprise NNTsGP "IGD im. A.A. Skochinsky" [2], the IP2.5 motor can develop power up to 60 kW at a nominal pressure of the working fluid (25 MPa) and a nominal speed (60 rpm); the time between failures to the limit state, characterized by a decrease in the total efficiency by 15%, is 5000 hours for it. (Time to first failure 3000 hours).

Taking into account (2) and after appropriate transformations (3) will take the form:

$$\mathrm{T} = T_б \frac{\mathrm{V}_б}{V} T, \text{ hour} \tag{4}$$

For radial plunger hydraulic machines, the sliding speed of the plunger relative to the cylinder is:

$$V = \frac{\mathrm{d}}{\mathrm{d}\varphi} \rho(\varphi) \frac{d}{dt} \varphi(t), \text{ m/s} \tag{5}$$

where $\rho(\varphi)$ - radius of curvature of the guide profile as a function of the angle of rotation of the hydraulic machine rotor; $\varphi(t)$ - angle of rotation of the hydraulic machine rotor as a function of time.

Thus, for the basic - motor operating mode IMP2,5 [5]:

$$\varphi_б(t) = \omega_{\text{н}} t, \tag{6}$$

For braking operation IMP2,5:

$$\varphi(t) = \omega_{\text{ут}} t, \tag{7}$$

Here:

$$\omega_{\text{ут}} = \frac{P}{P_{\text{н}}}(1 - \eta_{\text{г}})\omega_{\text{н}}, \text{ rad/s} \tag{8}$$

Substituting (4.5), taking into account (4.6), (4.7), (4.8) and taking into account that $\rho_б(\varphi) = \rho(\varphi)$, we get:

$$\mathrm{T} = T_б \frac{\mathrm{P}_{\text{нб}}}{\mathrm{P}_{\text{н}}} \frac{1}{(1-\eta_{\text{г}})}, \text{ hour} \tag{9}$$

where $P_{нб}$ - nominal working pressure in the basic - motor mode of operation IMP2.5, Pa; $P_н$ - design working pressure in braking mode IMP2.5, Pa; $\eta_г$ - volumetric efficiency of the brake hydraulic machine (assumed 0.92);

At $P_{нб} = P_н = 25$ MPa finally get:

$$\mathrm{T} = T_б \frac{1}{(1-\eta_{\text{г}})}, \text{ hour} \tag{10}$$

The operating time to the first failure of the brake hydraulic machine will be:

$$\mathrm{T} = 3000 \frac{1}{(1-0{,}92)} = 37500 \text{ hour}.$$

After conducting research and analyzing the selection of parameters for the rotary-feeding mechanism of a quarry drilling machine, several conclusions can be drawn:

Performance Requirements: The rotary-feeding mechanism plays a crucial role in the efficiency and productivity of a quarry drilling machine. It should be designed to meet the performance requirements of the drilling operation, such as rotational speed, torque, and feed rate. These parameters depend on factors like the type of rock, drill bit size, and desired drilling depth.

Power and Torque: The selection of an appropriate motor and power transmission system is vital to ensure sufficient power and torque for the rotary-feeding mechanism. The motor should be capable of delivering the required rotational speed and torque consistently to drive the drilling process effectively.

Gearbox Design: The gearbox used in the rotary-feeding mechanism should be designed to withstand the high torque and axial loads encountered during drilling operations. The selection of gear ratios and gear types should be optimized to provide the desired feed rate while maintaining reliability and durability.

Control System: An efficient control system is necessary to regulate the rotary-feeding mechanism accurately. It should enable precise control of the rotational speed, feed rate, and drilling depth. Advanced control algorithms and feedback mechanisms can enhance the overall performance of the drilling machine.

Material Selection: The selection of materials for the rotary-feeding mechanism components is crucial to ensure their strength, durability, and resistance to wear. Components such as gears, shafts, and bearings should be made from high-quality materials capable of withstanding the harsh operating conditions of a quarry drilling machine.

Safety Considerations: Safety should be a top priority when designing and selecting parameters for the rotary-feeding mechanism. Proper safeguards, emergency stop mechanisms, and overload protection systems should be incorporated to prevent accidents and protect both the machine operators and the equipment.

In conclusion, the research and selection of parameters for the rotary-feeding mechanism of a quarry drilling machine require careful consideration of performance requirements, power and torque, gearbox design, control systems, material selection, and safety. By optimizing these factors, it is possible to enhance the efficiency, reliability, and safety of the drilling machine, leading to improved productivity in quarry operations.

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