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2.5.21 Machines, aggregates and technological processes (engineering sciences)

MODERNIZATION OF DRILLING RIGS USING DRIVING PLANETARY MACHINES WITH VARIABLE PARAMETERS

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МОДЕРНИЗАЦИЯ БУРОВЫХ УСТАНОВОК С ИСПОЛЬЗОВАНИЕМ ПРИВОДНЫХ ПЛАНЕТАРНЫХ МЕХАНИЗМОВ С ПЕРЕМЕННЫМИ ПАРАМЕТРАМИ

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Abstract. Variable speed characteristics and variable gear ratios of the drive mechanisms of the working bodies of the drilling rig were obtained for a planetary mechanism with a double satellite and a rocker carrier. With the help of the obtained kinematic characteristics, the design and technological parameters of the planetary drive mechanisms of the technical means of drilling rigs will be established. Design and technological parameters will be tested with experimental studies depending on the variable speed characteristics of the planetary friction mechanism.

Key words: planetary mechanisms, boring machines, rotor, carrier, crank, rocker arm, slider, satellite, sun wheel

Аннотация. Для планетарного механизма с двойным сателлитом и водилом коромысла получены переменные скоростные характеристики и переменные передаточные числа механизмов привода рабочих органов буровой установки. С помощью полученных кинематических характеристик будут установлены конструктивные и технологические параметры планетарных механизмов привода технических средств буровых установок. Экспериментальными исследованиями в зависимости от изменяемой скоростной характеристики планетарного фрикционного механизма будут проверены конструктивные и технологические параметры.

Ключевые слова: планетарные механизмы, расточные станки, ротор, водило, кривошип, коромысло, ползунок, сателлит, солнечное колесо

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Introduction

Planetary mechanisms are finding ever more widespread use, both in transport and stationary machines. They allow transmissions to be made more compact and lighter, and have advantages in transmissions from high-speed engines to the working machine. They are widely used in lifting and transport engineering, automotive, tractor construction, in the forestry and woodworking industries, and epicyclical mechanisms are also used as a drive for working bodies, as well as pile machines and concrete mixers – boring machines.

In this article, we will consider kinematic and dynamic studies of the proposed drive mechanisms. The planetary gear with a double satellite and rocker carrier allows obtaining variable speed characteristics and variable gear ratios for the drive mechanisms of technical means of drilling well oil and gas.

In rotary drilling (fig. 1), rock destruction occurs as a result of the simultaneous action of load and

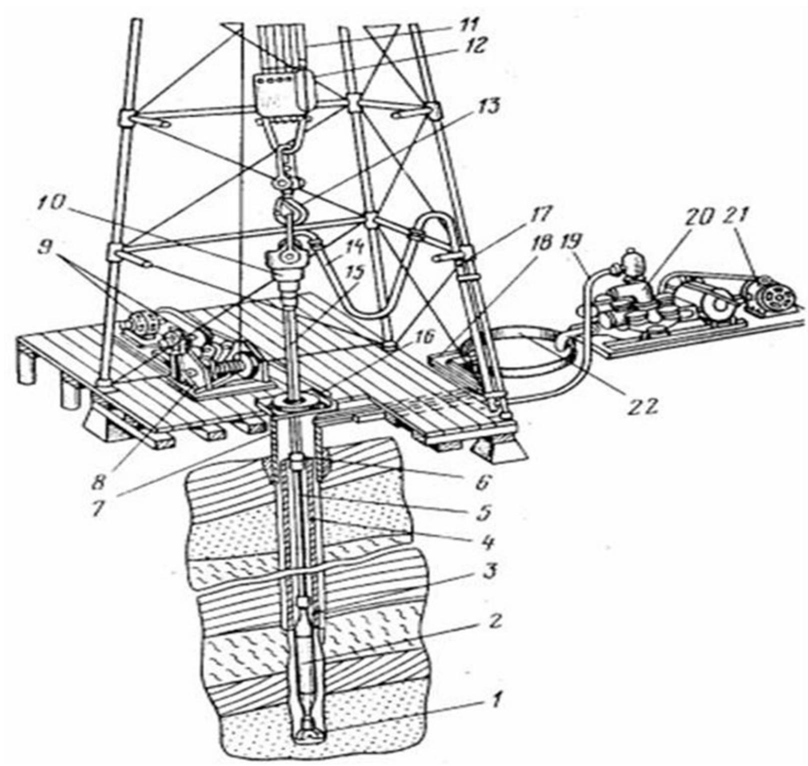

Fig. 1. Scheme of rotary drilling of wells: 1 – bit, 2 – motor, 3 – spaces, 4 – horizons, 5 – drill pipes, 6 – sub, 7 – pipe (pipe string), 8 – winch, 9 – motors, 10 – switch, 11 – wire rope, 12 – crownblock (not shown), 13 – hook, 14 – flexible drill hose (sleeve), 15 – kelly, 16 – rotor, 17 – riser, 16 – tank system (not shown), 19 – high – pressure pipeline, 20 – pump, 21 – engine, 22 – receiving tanks of mud pumps

torque on the bit. Under the action of the load, the bit penetrates into the rock, and under the influence of the torque it cleaves it. During rotary drilling (fig. 1), the power from the motors *(9)* is transmitted through the winch *(8)* to the rotor *(16)* – a special rotary mechanism installed above the wellhead in the center of the rig. The rotor rotates the drill string and the bit *(1)* screwed to it. The drill string consists of a leading pipe *(15)* and 6 drill pipes *(5)* screwed to it using a special sub *(6)*. Therefore, rotary drilling the bit deepening into the rock occurs when the rotating drill string moves along the borehole axis.

Object and methods of research

The object of research is the planetary mechanism. Rotor *(16)* (fig. 1) rotates according to the principle of a planetary mechanism around a central axis inside a fixed stator.

We take the radius of the crank $O_1 A_1 = R_1$, the length of the carrier $O_2B = \rho$, the wheel of the radius R, the radius $O_2A = \rho - l$, $O_2P = R$ (this is the line connecting the point of contact P with the point O_2), the length of the guide $AB = e$, the radius of the satellite $PB = r$, and ψ the angle between the radius of the carrier and the line O_2P, the carrier and the leading link by φ_1, and φ the angle between ρ and R (fig. 2). The angle of rotation of the composite carrier along ψ will be [1, 2]:

$$\psi = \phi \pm \alpha.$$

From the triangle O_1O_2A at $O_1O_2 = a$; $O_1A = R_1$ we have:

$$R_1^2 = a^2 + (\rho - l)^2 + 2a(\rho - l)cos\psi, \qquad (1)$$

where $\pi - \psi = \angle O_1O_2A$ is an obtuse angle.

Making transformations of expression (1), we have:

$$\rho = \sqrt{R_1^2 - a^2 sin\psi} - acos\psi + l. \qquad (2)$$

On the other hand, R is determined from the triangle O_2BP on the basis of the cosine theorem:

$$r^2 = R^2 + \rho^2 - 2\rho \cdot R \cdot cos\alpha.$$

Expressing with respect to R, we have:

$$R = \sqrt{r^2 - \rho^2 sin^2\alpha} + 2\rho cos\alpha. \qquad (3)$$

Substituting the value (2) into (3), we have:

$$R = \sqrt{r^2 - \left[\sqrt{R_1^2 - a^2 sin^2\psi} - acos\psi + l\right]^2 sin\alpha} +$$
$$+\left[\sqrt{R_1^2 - a^2 sin^2\psi} - acos\psi + l\right] \cdot cos\alpha.$$

The skew angle is determined by the expression:

$$ctg\alpha = \frac{R_1 + \dfrac{\sqrt{(\rho)^2 + [d\rho / d\psi]^2}}{r} \cdot \rho}{d\rho / d\psi}.$$

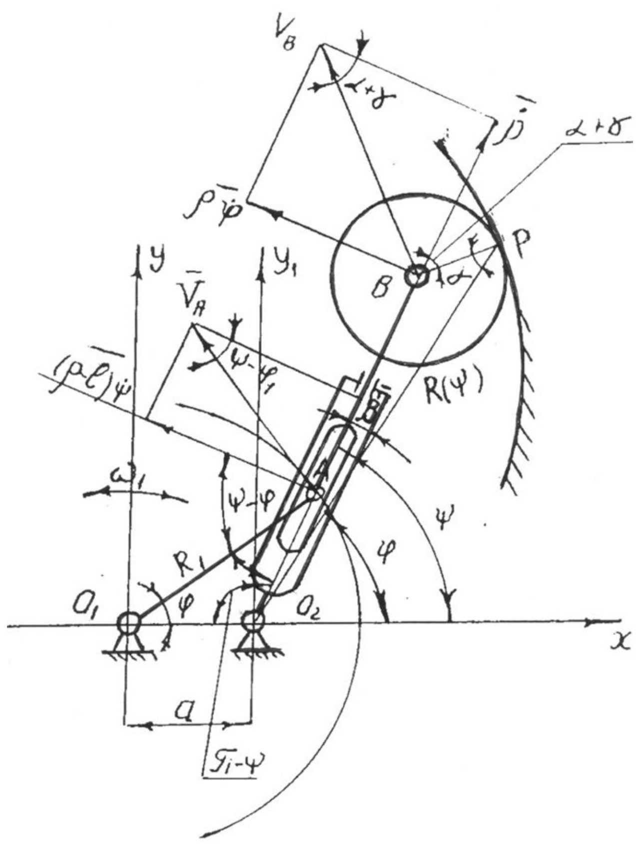

Fig. 2. Diagram of friction mechanism with a composite carrier: 1 – crank, 2 – rocker, 3 – slider, 4 – guide, 5 – satellite, 6 – wheel

The variable gear ratio of the investigated mechanism is determined by the formula [3, 4]:

$$U_{CH} = \frac{\omega_C}{\omega_H}. \qquad (4)$$

Gear ratio between the satellites by the driving link:

$$U_{C1} = \frac{\omega_C}{\omega_1}. \qquad (5)$$

The angular velocity of a composite carrier is determined by the angle of rotation ψ:

$$\omega_H = \frac{d\psi}{dt}.$$

The angular velocities of the satellite are determined:

$$\omega_C = \frac{\upsilon_B}{r}, \quad \omega_1 = \frac{\upsilon_A}{R_1}, \qquad (6)$$

where υ_B, υ_A are the linear velocities of the center of the satellite and point A of the leading link, which is determined by the formulas [5, 6]:

$$\upsilon_B = \sqrt{\left(\frac{d\rho}{dt}\right)^2 + \left(\frac{d\psi}{dt}\rho\right)^2},$$

$$\upsilon_A = \sqrt{\left(\frac{d\rho}{dt}\right)^2 + \left[\frac{d\psi}{dt}(\rho - l)\right]^2}. \qquad (7)$$

Expressions (4) and (5) taking into account (6) and (7) have the forms:

$$U_{CH} = \frac{\sqrt{\left(\frac{d\rho}{dt}\right)^2 + \left[\frac{d\psi}{dt}\rho\right]^2}}{r\frac{d\psi}{dt}}, \qquad (8)$$

$$U_{C1} = \frac{R_1}{r}\sqrt{\frac{\left(\frac{d\rho}{dt}\right)^2 + \left(\frac{d\psi}{dt}\rho\right)^2}{\left(\frac{d\rho}{dt}\right)^2 + \left[\frac{d\psi}{dt}(\rho - l)\right]^2}}, \qquad (9)$$

where the relative speed of the composite carrier is:

$$\dot{\rho} = \frac{d\rho}{dt} = \dot{\psi} \cdot a\sin\psi\left[1 - \frac{a\cos\psi}{\sqrt{R_1^2 - a^2\sin^2\psi}}\right]. \qquad (10)$$

The final gear ratio of the planetary friction mechanism with a composite telescopic rocker carrier (8) and (9), taking into account the formulas (2) and (10), will be written in the form of the gear ratio of the satellite link [7, 8]:

$$U_{CH} = \frac{1}{r}\sqrt{\left[a\sin\psi\left(1 - \frac{a\cos\psi}{\sqrt{R_1^2 - a^2\sin^2\psi}}\right)\right]^2 + \left[\sqrt{R_1^2 - a^2\sin^2\psi} - a\cos\psi + l\right]^2},$$

$$U_{C1} = \frac{R_1}{r}\sqrt{\frac{\left[a\sin\psi\left(1 - \frac{a\cos\psi}{\sqrt{R_1^2 - a^2\sin^2\psi}}\right)\right]^2 + \left[\sqrt{R_1^2 - a^2\sin^2\psi} - a\cos\psi + l\right]^2}{\left[a\sin\psi\left(1 - \frac{a\cos\psi}{\sqrt{R_1^2 - a^2\sin^2\psi}}\right)\right]^2 + \left[\sqrt{R_1^2 - a^2\sin^2\psi} - a\cos\psi\right]^2}},$$

where U_{CH}, U_{C1} accordingly, variable gear ratios between the rocker carrier satellite and the driving link satellite.

Based on expressions (10) taking into account $\omega_H = \omega_1\left(1 + \alpha^1\right)$ and $\dot{\rho} = \frac{d\rho}{dt}$, where $\alpha^1 = \frac{d\alpha}{d\psi}$, we obtain a generalized formula for determining the variable gear ratio of the investigated mechanism.

Gear ratio satellite-rocker pair is:

$$U_{CH} = \frac{\sqrt{\left[\omega_1\left(1 \pm \alpha^1\right)\rho\right]^2}}{r\omega_1\left(1 \pm \alpha^1\right)}, \qquad (11)$$

gear ratio satellite-leading link:

$$U_{C1}=\frac{R_1}{r}\sqrt{\frac{\left[\omega_1\left(1\pm\alpha'\right)\rho\right]^2+\left[\rho\right]^2}{\left[\omega_1\left(1\pm\alpha'\right)\cdot\left(\rho-l\right)\right]^2+\left(\dot{\rho}\right)^2}}, \quad (12)$$

where $\alpha'=\frac{\partial\alpha}{\partial\psi}$.

If the carrier length and angle are constant, we obtain the Willis formula from (11) and (12). Taking into account the constancy of the "mismatch" angle, we have the following approximate formula for determining the gear ratio:

$$U_{CH}=\frac{\sqrt{\left(\rho\right)^2+\left[d\rho/d\psi\right]^2}}{r},$$

$$U_{C1}=\frac{R_1}{r}\sqrt{\frac{\left(\rho\right)^2+\left[d\rho/d\psi\right]^2}{\left[\rho-l\right]^2+\left[d\rho/d\psi\right]^2}}.$$

The given basic kinematic dependences of the studied mechanisms make it possible to solve kinematic problems.

For dynamic analysis, consider the function of the transfer mechanism.

Results

We project the geometric parameters of the mechanism under study onto the coordinate axes OX and OY (fig. 3):

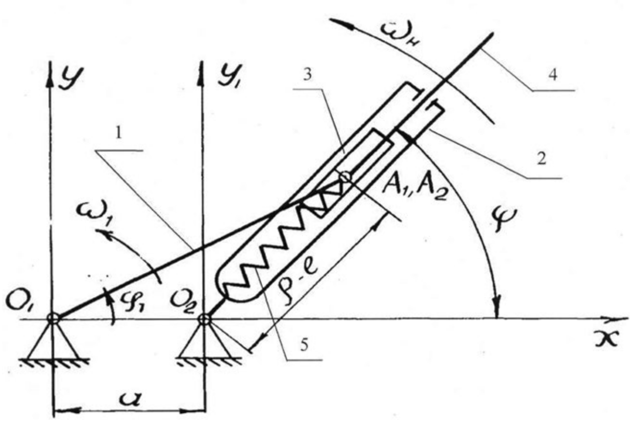

Fig. 3. Calculation scheme for determining transfer functions: 1 – crank, 2 – rocker, 3 – slider, 4 – guide, 5 – spring

The sum of the projections on the X-axis:

$$\sum_{i=1}^{i=n}X_i=0,\ R_1\cos\varphi_1-\left(\rho-l\right)\cos\psi-a=0.$$

The sum of the projections on the Y-axis:

$$\sum_{i=1}^{i=n}Y_i=0,\quad R_1\sin\varphi_1-\left(\rho-l\right)\sin\psi=0.$$

From the expressions for zero sums of projections, we determine:

$$\rho-l=\frac{R_1\cos\varphi_1-a}{\cos\psi}=\frac{R_1\sin\varphi_1}{\sin\psi}, \quad (13)$$

or

$$F\left(\varphi_1,\psi\right)=R_1\cos\varphi_1\sin\psi-a\sin\psi-$$
$$-R_1\sin\varphi_1\cos\psi=R_1\sin\left(\varphi_1-\psi\right)-a\sin\psi. \quad (14)$$

Then the total derivative has the form:

$$\frac{dF\left(\varphi_1,\psi\right)}{dt}=\frac{\partial F}{\partial\varphi_1}\frac{d\varphi_1}{dt}+\frac{\partial F}{\partial\psi}\frac{d\psi}{dt},$$

where $\Pi'=\frac{\partial F}{\partial\varphi_1}$ and $\Pi'=\frac{\partial F}{\partial\psi}$ are the transfer function of the crank or $\Pi'=\frac{R_1\cos\left(\varphi_1-\psi\right)}{R_1\cos\left(\varphi_1-\psi\right)-a\cos\psi}$.

Taking into account (14), the transfer function is equal to:

$$\Pi'=\frac{\partial\psi}{\partial\phi_1}=\frac{\frac{\partial F}{\partial\phi_1}}{\frac{\partial F}{\partial\phi_1}}.$$

Then the angular velocity of the link, taking into account (13) and (14), is equal to:

$$\omega_2=\Pi'\omega_1=$$
$$=\omega_1\left[R_1\cos(\phi_1-\psi)/(R_1\cos(\phi_1-\psi)-a\cos\psi)\right].$$

The transfer function of the link, taking into account (14), has the form [9, 10]:

$$\Pi''=\frac{\frac{\partial^2 F}{\partial \phi_1^2}+2\frac{\partial^2 F}{\partial \phi_1\,\partial \psi}\Pi'+\frac{\partial^2 F}{\partial \psi^2}\left(\Pi'\right)^2}{\frac{\partial F}{\partial \psi}}, \quad (15)$$

where second partial derivatives:

$$\frac{\partial^2 F}{\partial \phi_1^2}=-R_1\,sin\left(\phi_1-\psi\right),\quad \frac{\partial^2 F}{\partial \phi_1\,\partial \psi}=R_1\,sin\left(\phi_1-\psi\right),$$

$$\frac{\partial^2 F}{\partial \psi^2}=R_1 sin\left(\varphi_1-\psi\right)+a sin\psi. \quad (16)$$

Equation (15), taking into account (14) and formulas (16), has the form:

$$\Pi''=\frac{1}{R_1\,cos\left(\phi_1-\psi\right)-a cos\psi}\cdot$$

$$\cdot\left[R_1\,sin\left(\phi_1-\psi\right)+2R_1\,sin\left(\phi_1-\psi\right)\right]\cdot$$

$$\cdot\left[\frac{R_1\,cos\left(\phi_1-\psi\right)}{R_1\,cos\left(\phi-\psi\right)}+a cos\psi+R_1 sin\left(\phi_1-\psi\right)+a sin\psi\right]\cdot$$

$$\cdot\left[\frac{R_1\,cos\left(\phi_1-\psi\right)}{R_1\,cos\left(\phi_1-\psi\right)+a cos\psi}\right]^2.$$

Then the angular acceleration of the link of the composite carrier has the form:

$$\varepsilon_2=\Pi''\omega_1^2=\frac{\frac{\partial^2 F}{\partial \phi_1^2}+2\frac{\partial^2 F}{\partial \phi_1\,\partial \psi}\Pi'+\frac{\partial^2 F}{\partial \psi^2}\left(\Pi'\right)^2}{\frac{dF}{d\psi}}\omega_1^2.$$

Conclusions

A kinematic analysis of the models was carried out, on the basis of which dynamic dependencies between the design and technological parameters of the drive mechanisms of technological machines were established.

Analytical models of systems of nonlinear equations of motion of elements of planetary mechanisms in drilling rigs have been compiled. The transfer functions of the interaction of a pair of satellite-rocker and a pair of satellite-leading link, formulas for the angular velocities and accelerations of the rocker and the leading link were obtained.

Based on the experiments, the design and technological parameters of the drive planetary mechanisms of drilling rigs will be established depending on the variable kinematic characteristics obtained in the article.

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