Development of a mathematical model and analysis of the dynamics of an electromechanical perforator with an impact-rotary mechanism

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Abstract. The object of the study is an electromechanical hammer drill with a rotary-impact mechanism. When studying the dynamics of an electromechanical hammer drill with a rotary-impact mechanism, methods of mathematical modeling and comparative analysis were used. A calculation scheme has been developed and, on its basis, the equations of motion of mechanical transmission elements and working tools of an electromechanical hammer drill with a rotating impact mechanism have been obtained. The solution of the resulting equations is carried out using the Runge-Kutta numerical method using the Mathcad computer program. When analyzing the dynamics of the elements of a rotary hammer with an impact-rotating mechanism, the value of the moment of resistance on the part of the rotating mechanism is taken as a variable parameter, since during the operation of the rotary hammer, with increasing drilling depth, the magnitude of the moment acting on the part of the tool increases. The dependences of the angular velocities of the elements of the hammer drill are given, which make it possible to select rational design parameters of the links of the electromechanical hammer drill using an impact-rotary mechanism.

1 Introduction

Developing new and improving the designs of existing rotary hammers is a complex task and requires solving issues related to its performance, design reliability, and the choice of rational operating modes. During operation, manual hammer drills experience significant dynamic loads. These loads are caused by the operation of the hammer drill's impact and rotary systems, which lead to significant dynamic processes in the hammer drill mechanisms and, accordingly, require research into their dynamics. The study of the dynamics of the actuators of an electromechanical hammer drill with a rotating impact mechanism, as a single system of interacting elements, is carried out using the results of a preliminary study of the dynamics of individual elements of this system, i.e. motor, impact mechanism and tool rotation system.

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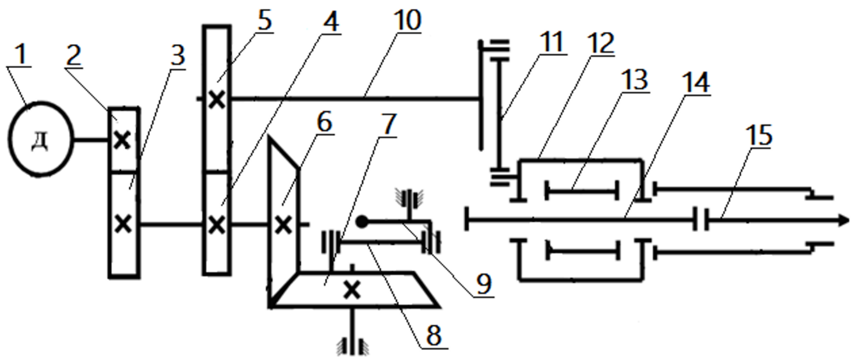
2 Materials and methods

When drawing up a generalized mathematical model of a given hammer drill, an important role is played by the correct simplification and drawing up of a calculation diagram. Therefore, when simplifying the design scheme, it is necessary to take into account the mass-inertia characteristics, compliance and elastic parameters of the hammer drill elements. A generalized mathematical model of a rotary hammer with an impact-rotary mechanism is compiled on the basis of its kinematic diagram.

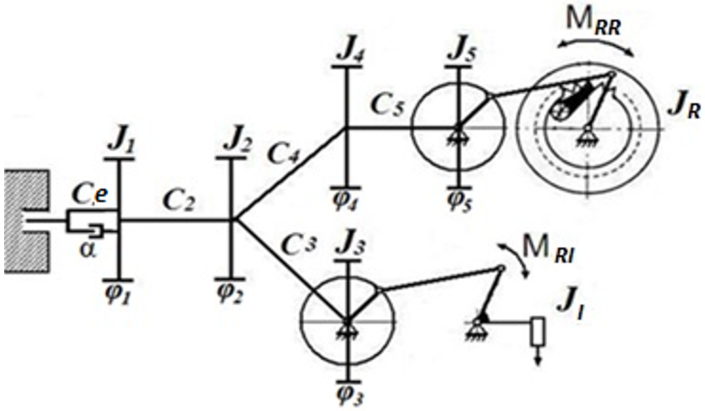
In a hammer drill with an impact-rotary mechanism (Figure 1), cylindrical helical gears 2 and 3, cylindrical spur gears 4,5 and bevel spur gears 6, 7 are used to transmit motion from the motor armature shaft 1 to the actuators (impact and rotary mechanisms). As can be seen from the figure, the torque of the wheel 2 (together with the electric motor shaft) is transmitted through the spur gear 3 to the gear shaft 4. Through the gear shaft 4, the rotational movement is distributed into two parts: through the bevel gear 6 to the impact mechanism, consisting of a crank 7 , connecting rod 8, rocker arm 9 and to a system that ensures rotation of the tool and consists of a crank shaft 10, connecting rod 11 and rocker arm 12. The rotation of the tool 15 at a certain angle and the transmission of engine torque to the tool is transmitted through a transmission mechanism based on a four-bar hinge and a ratchet mechanism 13. The engine torque is transmitted along one line to the tool for rotation at a certain angle, and along the other line through an impact mechanism, the shock impulse is transmitted to the tool in the form of a shock wave, and thus the shock-rotary mode of operation of the hammer drill is ensured [1,2,3].



1-universal commutator motor; 2,3,4,5,6,7 - gears; 8-connecting rod of the impact mechanism; 9- rocker arm of the impact mechanism; 10 – crank shaft of the rotary mechanism; 11- connecting rod of the turning mechanism; 12- rocker arm of the rotary mechanism; 13- ratchet mechanism; 14- waveguide; 15-tool.

Fig. 1. Kinematic diagram of a hammer drill with a rotary-impact mechanism.

According to the kinematic diagram of the hammer drill (Figure 1), its design diagram has been compiled, which, after simplifications (based on partial systems), has the form shown in Figure 2. Considering the motor shaft, gearbox elements and tool as the characteristic elements, we arrive at a model of the hammer drill in the form of a five-mass system with equivalent elastic connections between masses.



J_1, J_2, J_3, J_4, J_5, J_I, J_R - are the moments of inertia, respectively, of the motor armature shaft, pinion shaft, impact mechanism crank, gear wheel, rotary mechanism crank, impact mechanism and rotary mechanism; *Mri, Mrr* - moments of resistance, respectively, of the impact and rotary mechanisms; *Ce,* C_3*,* C_5 – stiffness, respectively, of the engine shaft, gears and splined connection of the wheel with the intermediate shaft; C_2*,* C_4 – total stiffness, respectively, of the gears and splined connection of the wheel with the intermediate shaft, gears; α – damping coefficient.

Fig. 2. Final simplified design diagram of an electromechanical rotary hammer drill.

The equation of motion of the armature of a universal commutator motor, i.e. the first mass of the design scheme will be written by the following equation [1,4,5]:

$$J_1\dddot{\phi}_1 = \Big[k_1 \cdot i^{1-a1} \cdot U \cdot \left(J_1 \cdot \ddot{\phi}_1 + M_R\right)^{a1} - k_2 \cdot i^{2-a1} \cdot \dot{\phi}_1 \cdot \left(J_1 \cdot \ddot{\phi}_1 + M_R\right)^{a2} - \\ -k_3 \cdot i^{1-a3} \cdot R \cdot \left(J_1 \cdot \ddot{\phi}_1 + M_R\right)^{a3} - \frac{dM_R}{dt}\Big] \;, \qquad (1)$$

where: $\dot{\varphi}, \ddot{\varphi}, \dddot{\varphi}$ - are, respectively, the angular velocity, acceleration and jerk of the electric motor armature; k_1,k_2,k_3-are constant coefficients; *J*-moment of inertia of the electric motor armature; *i*-current in the armature circuit; M_R - moment of resistance of the mechanism; *U* - voltage in the armature circuit; *R* - resistance in the armature circuit; *a1, a2, a3* - the number of pairs of parallel branches.

Taking into account (1), the equations of motion of the hammer drill elements according to this design scheme (Figure 2) are written by the following system of differential equations:

$$\begin{cases} J_1\dddot{\varphi}_1 = k_1 \cdot U \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a1} - k_2 \cdot \dot{\varphi}_1 \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a2} - k_3 \cdot R \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a3} - M_R'; \\ J_2\ddot{\varphi}_2 = C_2(\varphi_1 - \varphi_2) - C_3(\varphi_2 - \varphi_3) - C_4(\varphi_2 - \varphi_4); \\ J_3\ddot{\varphi}_3 = C_3(\varphi_2 - \varphi_3) - M_{RI}; \\ J_4\ddot{\varphi}_4 = C_4(\varphi_2 - \varphi_4) - C_5(\varphi_4 - \varphi_5); \\ J_5\ddot{\varphi}_5 = C_5(\varphi_4 - \varphi_5) - M_{RR}, \end{cases} \qquad (2)$$

where: - φ_1 - φ_5 angular movement of the puncher elements;

$\dot{\varphi}_1$ - $\dot{\varphi}_5$ - angular velocity of the hammer drill elements;

$\ddot{\varphi}_1$ - $\ddot{\varphi}_5$ - angular acceleration of individual masses of the hammer drill;

M_{RI} - moment of resistance acting from the impact side mechanism;

M_{RR} - moment of resistance acting on the side of the rotary mechanism.

In this system of equations, the moment of resistance acting from the impact mechanism is determined by the formula [1-8]:

$$M_{RI} =\rightleftarrows\rightleftarrows J_y \cdot \ddot{\phi}_y \cdot u_{43}, \text{ N·m}, \tag{3}$$

where: $\ddot{\phi}_i$ - angular acceleration of the impact mechanism rocker arm;

J_i- moment of inertia of the rocker arm of the impact mechanism;

u_{43} - an analogue of the angular velocity of the rocker arm of the impact mechanism.

The moment of resistance on the part of the rotating mechanism is determined by the formula [1.4]:

$$M_{RR} = J_r \cdot \ddot{\phi}_r \cdot u_{65} + \frac{M_{FR}}{u}, \text{ N·m}, \tag{4}$$

where: $\ddot{\phi}_n$ - angular acceleration of the rocker arm of the turning mechanism;

J_n- moment of inertia of the rocker arm of the rotary mechanism;

u_{65} - analogue of the angular velocity of the rocker arm of a rotary mechanism;

M_{FR}- the moment of friction of the tool on the processed medium; gear ratio of the rotating mechanism.

It should be noted that under the condition, (is the angle of movement of the rotating mechanism), the moment of friction of the tool against the processed medium M_{FR} takes on a given value, which ensures the working stroke of the ratchet mechanism, ensuring the rotation of the tool at a certain angle. Provided M_{FR} = 0, i.e. in this range, the ratchet mechanism reverses, and it takes its original position.

Taking into account the values $\ddot{\phi}_i$ and $\ddot{\phi}_r$

$\ddot{\phi}_i = \ddot{\phi}_3 \cdot u_{43} + \dot{\phi}_3^{\,2} \cdot \dot{u}_{43}$ and $\ddot{\phi}_r = \ddot{\phi}_5 \cdot u_{65} + \dot{\phi}_5^{\,2} \cdot \dot{u}_{65}$ we have

$$M_{RI} =\rightleftarrows\rightleftarrows J_i \cdot u_{43}(\ddot{\phi}_3 \cdot u_{43} + \dot{\phi}_3^{\,2} \cdot \dot{u}_{43}), \text{ N·m}. \tag{5}$$

$$M_{RR} =\rightleftarrows\rightleftarrows J_r \cdot u_{65}(\ddot{\phi}_5 \cdot u_{65} + \dot{\phi}_5^{\,2} \cdot \dot{u}_{65}) + \frac{M_{FR}}{u}, \text{ N·m}. \tag{6}$$

We rewrite the system of differential equations (2) taking into account equations (3) – (6) in the form:

$$\begin{cases} \ddot{\varphi}_1 = \dfrac{k_1 \cdot U \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a1} - k_2 \cdot \dot{\varphi}_1 \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a2} - k_3 \cdot R \cdot (J_1 \cdot \ddot{\varphi}_1 + M_R)^{a3} - M'_R}{J_1}; \\ \ddot{\varphi}_2 = \dfrac{C_2(\varphi_1 - \varphi_2) - C_3(\varphi_2 - \varphi_3) - C_4(\varphi_2 - \varphi_4)}{J_2}; \\ \ddot{\varphi}_3 = \dfrac{C_3(\varphi_2 - \varphi_3)}{J_3 + J_y \cdot u_{43}^{\,2}} - \dfrac{J_y \cdot u_{43} \cdot \dot{\varphi}_3^{\,2} \cdot \dot{u}_{43}}{J_3 + J_y \cdot u_{43}^{\,2}}; \\ \ddot{\varphi}_4 = \dfrac{C_4(\varphi_2 - \varphi_4) - C_5(\varphi_4 - \varphi_5)}{J_4}; \\ \ddot{\varphi}_5 = \dfrac{C_5(\varphi_4 - \varphi_5)}{J_5 + J_n \cdot u_{65}^{\,2}} - \dfrac{J_n \cdot u_{65} \cdot \dot{\varphi}_5^{\,2} \cdot \dot{u}_{65} + \dfrac{M_{FR}}{u}}{J_5 + J_n \cdot u_{65}^{\,2}}. \end{cases} \tag{7}$$

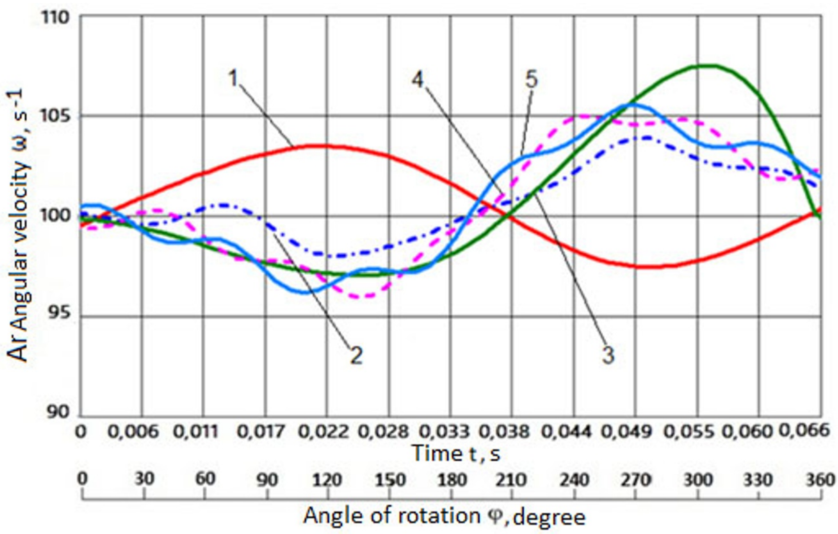
The solution of the resulting differential equations (7) is carried out using the numerical Runge-Kutta method using the Mathcad computer program. The solution of these differential equations allows us to analyze the dependencies of the kinematic values of the elements of the hammer drill and determine the rational design parameters of the impact-rotary mechanism.

3 Results and discussion

When analyzing the dynamics of the elements of a rotary hammer with an impact-rotating mechanism, the value of the moment of resistance on the part of the rotating mechanism is taken as a variable parameter, since during the operation of the rotary hammer, with increasing drilling depth, the magnitude of the moment acting on the part of the tool increases.

Figure 3 shows the dependences of the angular velocities of the hammer drill elements on the hammer drill operating cycle time t, which can be converted into the rotation angle of the crank of the impact mechanism. For one cycle of operation of the hammer drill, time t is taken, corresponding to one revolution of the crank of the impact mechanism of the hammer drill.

The change in the angular velocity of the engine, reduced to the crank of the impact mechanism (curve 1) during the operation of the hammer drill, occurs uniformly. This is facilitated by the fairly large transmission ratio of the mechanisms from the engine to the load element.



1 - angular speed of a universal commutator motor; 2- angular speed of the gear shaft; 3-angular speed of the impact mechanism crank; 4-angular speed of the gear; 5-angular speed of the crank of the turning mechanism.

Fig. 3. Dependences of the angular velocities of the hammer drill elements on time t and rotation angle φ (at $\Delta t=0{,}6\cdot10^{-4}$s and $M_R=0$)

In the diagram under consideration, the zone of relatively high angular velocity of the engine falls in the zone of reverse motion of the rocker arm of the impact mechanism, and the zone of low angular velocity of the engine falls, accordingly, in the cocking zone of the rocker arm. Also, the values of the angular velocity of the engine are related to the magnitude of the load in the impact mechanism, which depends on the parameter of the reverse stroke and cocking of the rocker arm and the increase in the drilling depth of the hammer drill. With increasing value of the moment of resistance, i.e. load, the arithmetic mean value of the angular speed of the engine decreases.

Analyzing the diagram (Figure 3), it can be noted that the values of the angular velocities of the second (curve 2), fourth (curve 4) and fifth (curve 5) elements are around the values of the angular velocity of the third element ω_3, i.e. crank of the impact mechanism, where, according to the design diagram, the connecting rod and rocker arm of the impact mechanism were brought. The angular velocity of the crank (curve 3) has minimum values in the rocker arm reverse zone and relatively high values in the rocker arm cocking zone. The return zone of the crank of the impact mechanism corresponds to the operating mode of the rotary mechanism, respectively, the cocking zone of the rocker corresponds to the idle mode of the rotary mechanism.

It should be noted that the value of the angular velocity of the crank ω3 (at M_R=0) reaches its maximum value (ω_{3max}=107,5 s^{-1}) at a crank angle of 305^0.

If we consider the technical characteristics of a universal commutator motor, then the nominal speed of the armature is 16600 rpm. (at a voltage of 220V), which corresponds to the average angular velocity of the crank of the impact mechanism (at M_R=0) ω_3=103,4s^{-1}. The arithmetic average value of the angular velocity of the crank of the impact mechanism (at M_R=0) according to the results of the mathematical model is about 102 s^{-1}. The error between the passport data of the angular velocity of the universal commutator motor driven to the crank of the impact mechanism and its value according to mathematical modeling is on average 1.5%.

When the crank rotation angle is φ=305°, its angular velocity decreases and returns to the average value. The reduction in the angular velocity of the crank is also affected by the value of the transmission ratio of the impact mechanism, which increases at the end of the cycle.

An increase in the moment of resistance on the part of the rotating mechanism leads to an increase in the amplitude of vibrations of the second, fourth and fifth elements of the hammer drill (Figure 4).

Quantitatively, the increase in the amplitude of angular velocity fluctuations is for the crank of the rotary mechanism (ω_5) without load $\Delta\omega_1\approx$2,0s^{-1}, with a load M_R=0,03Nm, $\Delta\omega_1\approx$2,2s^{-1}, with a load M_R=0,06Nm, $\Delta\omega_1\approx$5,2s^{-1}, at load M_R=0,09Nm, $\Delta\omega_1\approx$9,5s^{-1}, i.e. with increasing load, the amplitude of oscillations increases linearly.

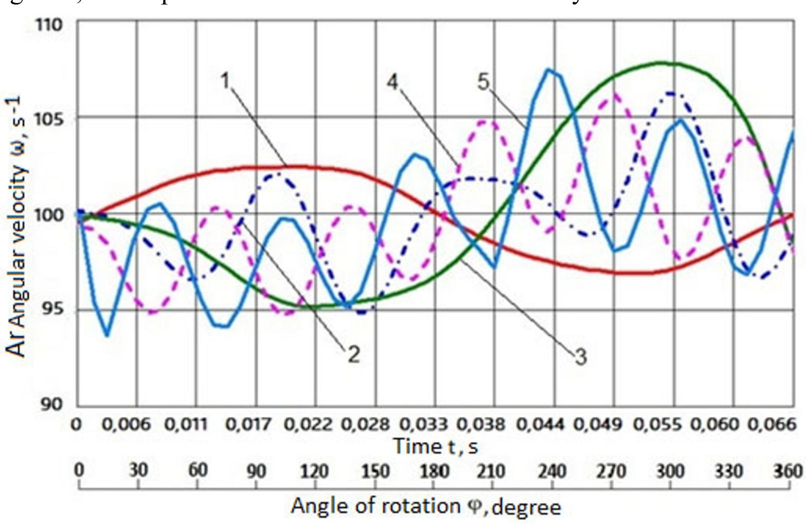


Fig. 4. Dependences of angular velocities of hammer drill elements on time t and rotation angle φ (at $\Delta t=0,6\cdot10^{-4}$s and M_R=0.09Nm).

The magnitude of the moment of resistance of the rotating mechanism changes with increasing drilling depth, i.e. it increases with increasing friction moment between the surface of the tool and the well.

An increase in the vibration amplitude of the elements, accordingly, leads to an increase in the magnitude of cyclic loads on the supports, which in the future can lead to breakdowns of the element supports. Therefore, when manufacturing parts and their supports, it is necessary to take into account additional loads caused by the presence of vibrations and their increase with increasing load.

It should be noted that the values of the angular speeds of the engine (ω_1) and the crank of the impact mechanism (ω_3) slightly depend on the loads, in turn, an increase in load leads to a slight increase in the amplitude of fluctuations in the angular speed of the crank of the impact mechanism.

The remaining elements of the hammer drill (ω_2, ω_4, ω_5) are significantly influenced by the load values.

Therefore, when improving the design of rotary hammers, it is necessary to pay attention to the reliability of the supports of the above-mentioned elements.

4 Conclusion

A generalized mathematical model of a hammer drill with a rotating impact mechanism has been developed, the solution of which will allow us to analyze the main kinematic parameters of the elements included in the mathematical model. A program for numerical solution of differential equations of a mathematical model based on the Mathcad computer program has been developed. Analysis of the obtained diagrams showed that elements 2, 4 and 5 are more susceptible to torsional vibrations, in which, with increasing load, the amplitude of angular velocity fluctuations intensively increases, which leads to external loads on the supports of these elements.

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