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The Study of the Effect of the Parameters of Elastic Coupling on the Hacker of Motion of the Rocker Arm of the Crank and Beam Mechanism

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ABCTRACT: In the article the scheme of a design and a principle of work of the crank-rocker mechanism is resulted. Dynamic and mathematical model of motion of the machine unit is given, taking into account the mechanical characteristics of the drive, the parameters of elastic coupling. On the basis of the analysis of the solution of the problem, the laws of rolling a rocker arm are obtained, taking into account the rigidity of the elastic coupling. Graphical dependencies of the parameters are obtained on the basis of the analysis which can be recommended for the necessary values.

KEYWORDS: Crank and rocker mechanism, elastic coupling, law of motion, angular velocity, rigidity, dissipation, resistance, dependencies, values.

I.INTRODUCTION.

Improving the quality of products while reducing costs is an important national economic problem. the leading role in solving this problem is the task of reducing the cost of materials for the production of a unit of production by improving the design of mechanisms and machines. The main direction in eliminating the shortcomings is to identify the reasons for the low effectiveness of the impact of working mechanisms on the processed material and the development of methods and mechanisms. Of the mechanisms with lower pairs, the mechanism of the hinged quadruple [1].

II. LITERATURE SURVEY

In the known mechanism there are four links: a crank making a complete revolution around a fixed axis, a rocker making rocking motion, the connecting rod performs a planar-parallel complex motion, the post (body) is a fixed link. The mechanism allows obtaining different laws of motion of the connecting rod and rocker in the plane, as well as their points, which mainly depend on the ratio of the lengths of the links. But, in this crank and rocker mechanism, shock phenomena occur in kinematic pairs at the extreme positions of the links, which leads to a decrease in the life of the mechanism, especially at high speed modes of operation of the mechanism. In addition, this mechanism does not allow the necessary correction of the movement of the connecting rod and rockerarm [2, 3].

III. METHODOLOGY

To eliminate shock phenomena in kinematic pairs in the extreme positions of the connecting rod and rocker, and also to implement the necessary corrections of the laws, a new flat mechanism is developed for their motion. The recommended crank and rocker mechanism comprises a crank, a connecting rod, a rocker arm and a rack (casing), connected by kinematic pairs (hinges), the kinematic pair (hinge) of the beam being connected to the stand by means of a rubber cushion, and also rubber shock absorbers installed in the casing located along the to both sides of the rocker with the possibility of interaction with the rocker in its extreme positions (Fig. 1). This design allows for the attenuation of the peak values of the reaction forces in the kinematic pair (hinge) of the beam with the stand and allows smooth

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transient modes of oscillation of the beam in its extreme positions, leading to a significant increase in reliability and operating life of the mechanism.

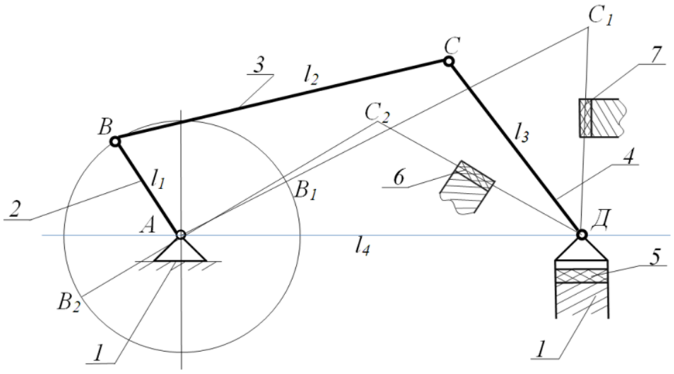

Fig 1. Scheme of the crank and beam mechanism

The crank-rocker mechanism consists of a rack 1 (body), a crank 2, a connecting rod 3 and a rocker 4 connected together by hinges A, B, C, D. The hinge D of the rocker 4 is mounted on the stand 1 by means of a rubber cushion 5, and on both sides of the rocker arm 4 with the possibility of interaction in the extreme position of the rocker mounted rubber (elastic) shock absorbers 6 and 7 connected to the rack 1 (see Figure 1).
The crank-rocker mechanism works as follows. Crank 2 receives rotational motion from the drive motor (not shown in the figure). Accordingly, the movement from the crank 2 is transmitted to the connecting rod 3, then to the rocker arm 4. The movement of the mechanism takes place in the plane. At the extreme positions of the crank 2, connecting rod 3 and rocker 4, impact phenomena occur in the kinematic pairs A, B, C, D between links 2, 3, 4 and 1. During impacts, the rubber cushion 5 deforms and absorbs the peak values of the reaction forces of the hinge D of the rocker 4. In the extreme positions C1 and C2 of the rocker 4, the maximum values of the accelerations arise, which leads to additional withdrawal forces. At the same time, due to the interaction of the beam 4 in these positions with rubber shock absorbers 6 and 7, some absorption of forces occurs and results in a smooth transition of the mode of motion of the mechanism.
Equations of motion of the machine assembly with the recommended mechanism, analysis of the motion of the rocker arm. The movement of the machine assembly taking into account the moment of the asynchronous electric motor in the form of a dynamic characteristic and working bodies is described by the following system of equations [4]

$$\frac{\omega_0 - \dfrac{d\varphi_1}{dt}}{\omega_0} = \frac{S_k M_\partial}{2M_k} + \frac{\dot{M}_\partial}{2\omega_c M_k}$$

$$J_1 \frac{d^2\varphi_1}{dt^2} = M_\partial - u_{21} M_1$$

$$J_2 \frac{d^2\varphi_2}{dt^2} = M_1 - u_{32} M_2$$

$$J_3 \frac{d^2\varphi_3}{dt^2} = M_2 - c_1\varphi_1 - в_1 \frac{d\varphi_3}{dt} - M_{cn}$$

where ω_0- angular velocity of ideal idling, 1/s; S_k - critical or maximum slip; M_∂ - the driving moment of an asynchronous electric motor, Nm; M_k - the critical maximum moment of the electric motor in the static mode (or tilting), Nm; ω_c - angular frequency of the network, 1/s; J_1, J_2, J_3- moments of inertia, respectively, of the machine aggregate, kgm^2; angular velocities according to the masses of the machine aggregate, 1/s; M_1 and M_2 - moments of resistance, respectively, of the machine aggregate, Nm.

IV. RESULTS

The numerical solution of the problem was carried out with the following values of the parameters: N=0,35 kW, n_1=1430 rpm, l_1=(4,5-8,2) mm, l_2=(12,5-17,5) mm, l_3=(7,5-11,5) mm, c_1=(1,2-1,8)·10^2 Nm/rad, J_3=(2,4-3,2)·10^{-3} kgm^2.

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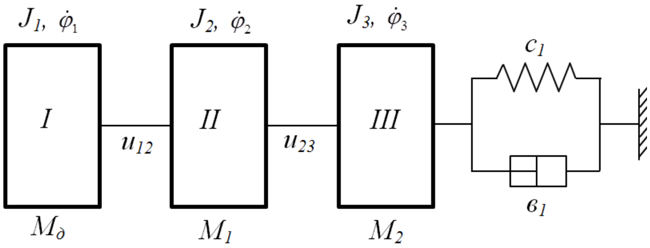
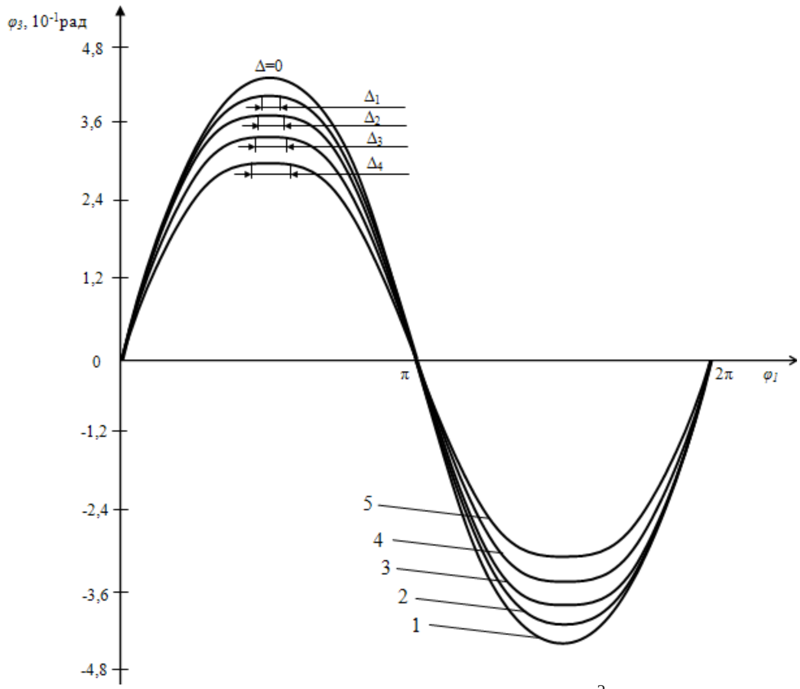


Fig 2. The design scheme of the three-mass machine assembly of the crank and beam mechanism

In Fig. 3 shows the regularities of rocking motion of the rocker, taking into account the effect of the rigidity of the elastic bonds established in the extreme positions of the rocker in the body of the mechanism. An analysis of the regularity of the rocker's motion shows that the amplitude of the oscillation in the absence of elastic bonds reaches 0,46 rad, and under the influence of an elastic coupling with a stiffness coefficient of $1,2{\cdot}10^2$ Nm/rad, the amplitude is reduced to 0,41 rad.In this case, in fact, there appears a protracted rocker beam within (0,1-0,12) π of the crank mechanism turning.
With an increase in the stiffness coefficient of the elasticity of the bonds to $3,0{\cdot}10^2$ Nm/rad, the value of the rocker endurance in the extreme positions reaches (0,16-0,21) π of the crank angular displacement. It is important to determine the time of the rocker's standstill when it interacts with the elastic restraints of the mechanism installed in the body. In Fig. 4 shows the graphical dependences of the change in the time of the rocker stand in the extreme positions from the variation in the rigidity of elastic bonds. Analysis of the graphs shows that with an increase in the elastic stiffness coefficient from $0,9{\cdot}10^2$ Nm/rad to $3,4{\cdot}10^2$ Nm/rad and at ω_1=$1,8{\cdot}10^2$ rad/s, the time of the rocker extension in the extreme positions increases from 0,06 s to 0,095 s at M_{cn}=0, and at M_{cn}=1,8·10 Nm, this time is reduced to 0,045 s.Therefore, in order to increase the time of the rocker's endurance in the extreme positions, it is advisable to increase the rigidity of the elastic bonds.



1-at without elastic coupling; 2- withc_1=$1,2{\cdot}10^2$ Nm/rad;
3-with c_1=$1,8{\cdot}10^2$ Nm/rad;4-with c_1=$2.4{\cdot}10^2$ Nm/rad;
5- withc_1=$3.0{\cdot}10^2$ Nm/rad
Fig 3. Regularities of the rocker's motion when the coefficient of rigidity of the elastic coupling is changed

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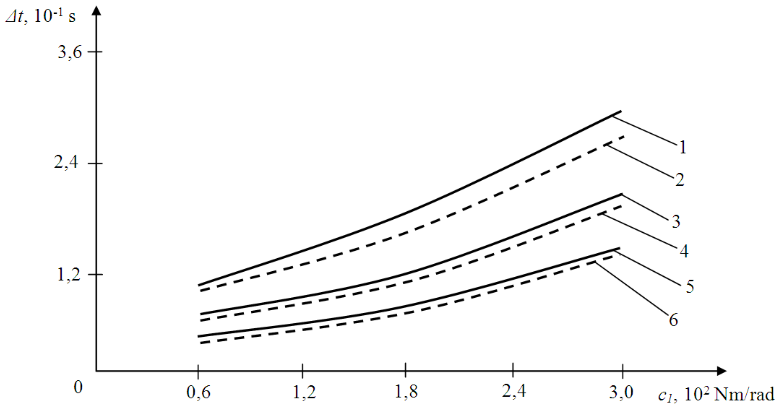
Specific values are selected depending on the technological requirements of the machine. The increase in the technological load also includes, to some extent, the change in the time of the rocker's standstill (see Fig. 4, graphs 1, 3, 5).In addition, an increase in technological resistance leads to an increase in the swing of the angular velocity of the beam (see Fig. 5).Thus, with an in crease in M_{cn} from 6,0 Nm to 70 Nm at c_1=1,2·10^{2}Nm/rad, the angular velocity of the rocker ($\Delta\dot{\varphi}_3 = \dot{\varphi}_{3\max} - \dot{\varphi}_{3\min}$) in creases from 0,72 rad/s to 2,69 rad/s, and for c_1=2.4·102 Nm/rad radius of the oscillation of the angular velocity of the rocker in creases to 1,52 rad/s.

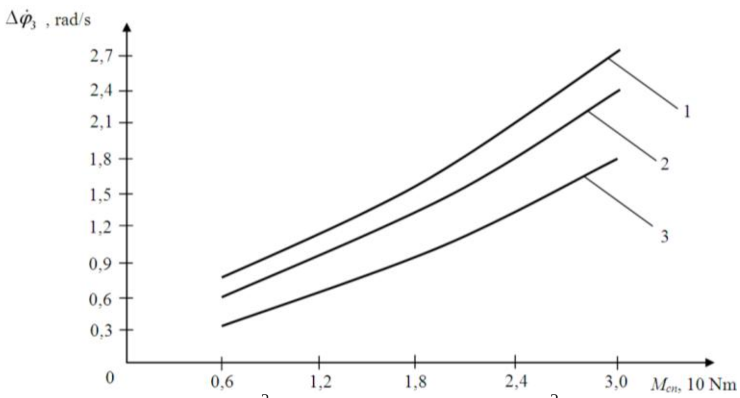
It should be noted that the swing of the angular velocity of the beam is largely dependent on the moment of inertia (see Figure 6).



1, 3, 5- taking into account M_{cn}=1,8·10 Nm;
2, 4, 6- without taking into account M_{cn};1, 2-at ω_1=2,2·10^2 rad/s;
3,4- at ω_1=1,8·10^2 rad/s;3- with ω_1=1,1·10^2 rad/s

Fig 4. Graphical dependences of the change in the time of the beam (interaction) of the beam within the limits of the position from the change in the coefficient of rigidity of the elastic coupling and the speed of the crank

The increase in the moment of inertia of the rocker from 0,8·10^{-3} kgm^2 to from 5,6·10^{-3} kgm^2 at c_1=2,4·10^2 Nm/rad the swing angle of the angular velocity of the rocker decreases from 2,1 rad/s to 0,31 rad/s by the nonlinear regularity. As the elasticity coefficient decreases, the elastic coupling decreases from 4,22 rad/s to 2,05 rad/s.Therefore, choosing the required values of J_3, $c1$, M_{cn}, l_1, l_2, l_3, we can obtain the required values of the changes φ_3, Δt, $\Delta\dot{\varphi}_3$ etc. for the corresponding technological machine.



1-with c_1=1,2·10^2 Nm/rad; 2-with c_1=1,8·10^2 Nm/rad;
3-with c_1=2,4·102 Nm/rad

Fig 5. Graphical dependencies of the sweep of the angular velocity of the beam from the change in the moment of resistance

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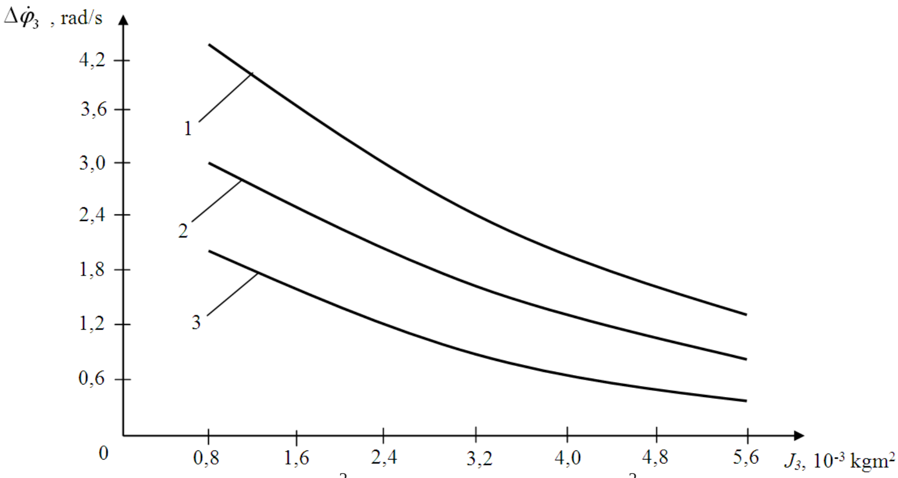

1-with c_1=1,2·10^2 Nm/rad; 2-with c_1=1,8·10^2 Nm/rad;
3-with c_1=2.4·10^2 Nm/rad

Fig 6. Graphical dependencies of the sweep of the angular velocity of the beam from its moment of inertia

V. CONCLUSION

A new scheme of a hinged four-link with an elastic coupling was developed. On the basis of solving the problems of the dynamics of a machine unit with the recommended mechanism, the laws of motion of the beam and the graphic dependences of the parameters are obtained.

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