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Study the influence of parameters of elastic coupling on the movement nature of support roller and rocker arm crank-beam mechanism

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ABSTRACT. In this work, the design of the support roller and crank-beam mechanism is studied. It allows peak damping values of reaction forces in a kinematic pair of rocker arms with a stable one and allows smooth transient modes of oscillation of the rocker arms in its extreme positions leading to a significant increase in reliability and service life of the mechanism. The mechanism is important to determine the law of the punitive movement of the rocker arm associated with the working body

KEYWORDS*:* mechanism, connecting rod, rocker, hinge, resource, elastic, analysis, crank, kinematics, roller, support, shock absorber, technology, experiment

I.INTRODUCTION

In our Republic of Uzbekistan, attention is being paid to the development of mechanical engineering, while on the basis of deep theoretical and experimental research on the development of resource-saving, highly efficient mechanisms for technological machines of the new generations, in particular for mechanical engineering. In the development strategy of the Republic of Uzbekistan in 2017–2021, the tasks are noted, in particular, “strengthening macroeconomic stability, maintaining high rates of economic development, increasing the competitiveness of the national economy, reducing resources and energy costs in the economy, and widespread introduction of energy-saving technologies into production”. The fulfillment of these tasks, in particular the development of design schemes, as well as methods for calculating lever mechanisms with elastic elements and with flexible links, will allow us to obtain the necessary laws of motion for driving wool.

All mechanisms composed only of solids are divided into two large groups: mechanisms with lower pairs, sometimes called rod or lever mechanisms, and mechanisms with higher pairs. Of the mechanisms with lower pairs, the four-link hinge mechanism is most prevalent. In a known mechanism, four links: a crank-making a complete rotation around a fixed axis, a rocker making a punitive movement, a connecting rod making a plane-parallel complex movement, a strut (housing) - a fixed link. The mechanism allows obtaining different laws of the movement 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-beam mechanism, shock phenomena occur in kinematic pairs at extreme positions of the links, which lead 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 rocker arm. For the elimination of shock phenomena in kinematic pairs in the extreme positions of the connecting rod and rocker arm, as well as the implementation of the necessary corrections of the laws of their motion, a new flat mechanism has been developed.

II. SIGNIFICANCE OF THE SYSTEM

The recommended crank-beam mechanism contains a crank, a connecting rod, a rocker arm and a stand (body) interconnected by kinematic pairs (hinges), while the kinematic pair (hinge) of the rocker arm is connected to the stand by a rubber cushion, as well as rubber shock absorbers installed in the body located along both sides of the rocker arm with the ability to interact with the rocker in its extreme positions. This design allows the depreciation of the peak values of the reaction forces in the kinematic pair (hinge) of the rocker arm with a stance and allows smooth transient oscillations of the rocker arm in its extreme positions, leading to a significant increase in the reliability and service life of the mechanism.

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The crank-beam mechanism consists of a rack 1 (body), crank 2, connecting rod 3 and rocker arm 4 connected by hinges A, B, C, D. The hinge D of the rocker arm 4 is mounted on the rack 1 by means of a rubber pad 5, and 4 with the ability to interact in the extreme positions of the rocker arm installed rubber (elastic) shock absorbers 6 and 7 connected to the stand 1 (see. Fig. 1.1).

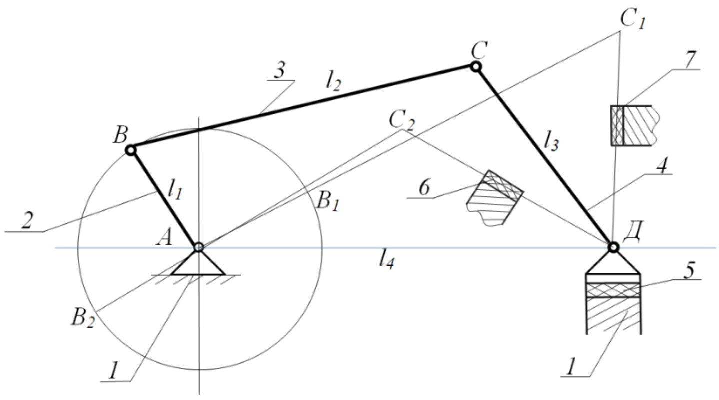

Fig. 1.1. Scheme crank-beam mechanism

Crank-beam mechanism works as follows. Crank 2 receives rotational motion from the drive motor (not shown in the figure).Accordingly, the movement of the crank 2 is transmitted to the connecting rod 3, then the rocker arm 4. The movement of the mechanism occurs in the plane. At the same time, in the extreme positions of the crank 2, the connecting rod 3 and the rocker arm 4, shock phenomena occur in the kinematic pairs A, B, C, D between the links 2, 3, 4 and 1. When struck, the rubber cushion 5 deforms and absorbs the peak values of the reaction forces of the hinge D of the rocker arm 4. In the extreme positions C1 and C2 of the rocker arm 4, maximum acceleration values occur, which leads to additional deflection forces. At the same time, due to the interaction of the rocker arms 4 in these positions with rubber shock absorbers 6 and 7, some absorption of forces takes place and leads to a smooth transition of the movement mode of the mechanism.

In this mechanism, it is important to determine the law of swinging movement of the rocker arm associated with the working body, taking into account the influence of the stiffness of the elastic connection and the moment of inertia of the rocker arm.

III. METHODOLOGY

The movement of the machine unit (Fig. 1.2), taking into account the moment of the asynchronous electric motor in the form of dynamic characteristics and working bodies, will be described by the following system of equations.

$$\frac{\omega_0 - \dot{\varphi}_1}{\omega_0} = \frac{S_k M_\partial}{2M_k} + \frac{M_\partial}{2\omega_c M_k}$$

$$J_1\ddot{\varphi}_1 = M_\partial - u_{21}M_1$$

$$J_2\ddot{\varphi}_2 = M_1 - u_{32}M_2$$

$$J_3\ddot{\varphi}_3 = M_2 - c_1\varphi_1 - в_1\dot{\varphi}_1 - M_{cn}$$

whereω_0 – the angular velocity of the ideal without load, 1/c; S_k- critical or maximum slip; M_∂ – induction motor driving moment, Nm; M_k – critical maximum motor torque in static mode (or tilting), Hm; ω_c–angular frequency of network, 1/s; J_1, J_2, J_3 – moments of inertia, respectively, of the masses of the machine unit, kgm^2; $\ddot{\varphi}_1, \ddot{\varphi}_2, \ddot{\varphi}_3$ - angular acceleration according to the masses of the machine unit, 1/s^2; M_1иM_2 – moments of resistance respectively, Hm; u_{12}andu_{23}-gear ratios according to the masses of the machine unit.Numerical solution of the problem was carried

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out with the following values of parameters: N=0,35 kVt, 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, s=(1,2-1,8)·10^2Nm/rad, J_3=(2,4-3,2) ·10^{-3}kgm^2 u_{12}=1,1; u_{23}=1,5.

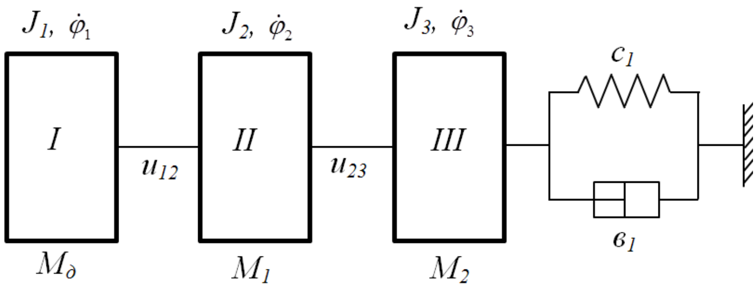
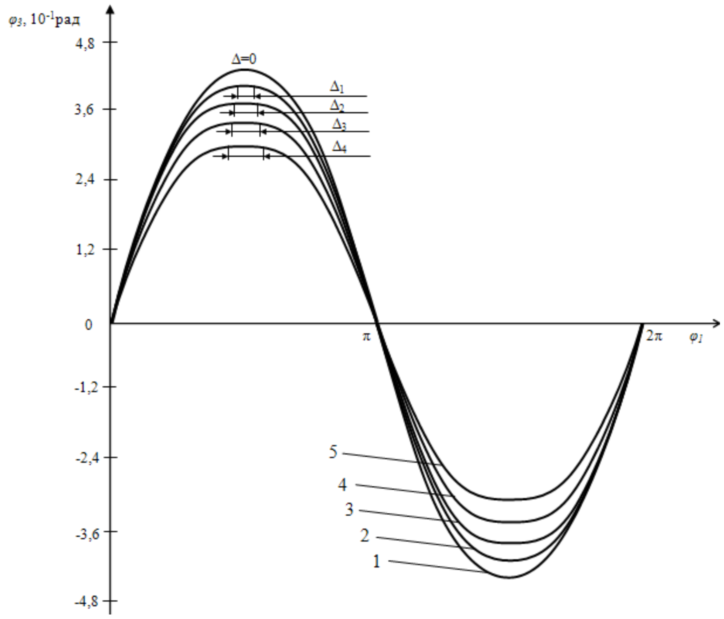


Fig. 1.2. The design scheme of the two-mass machine unit crank-beam mechanism

In fig. 1.3 shows the patterns of punitive movement of the rocker with the influence of the stiffness coefficient of elastic connections established in the extreme positions of the rocker arm in the mechanism housing.Analysis of the regularity of the movement of the rocker shows that the amplitude of oscillation in the absence of elastic bonds reaches 0.46 rad, and when subjected to an elastic bond with a stiffness coefficient of 1.2 · 102 Nm / rad, the amplitude decreases to 0.41 rad. At the same time, in fact, there appears a peck of the rocker arm within (0.1-0.12) π of the crank of the mechanism.



1-without elastic connection; 2-whenc_3=1,2·10^2Nm/rad; 3- wherec_3=1,8·10^2Nm/rad; 4- whenc_3=2,4·10^2Nm/rad; 5- whenc_3=3,0·10^2Nm/rad

Fig. 1.3. Patterns of movement of the rocker arm when changing the stiffness coefficient of the elastic connection

With an increase in the stiffness coefficient of the elastic connections, up to 3.0 · 102 Nm / rad, the height of the rocker arm in the extreme positions reaches (0.16-0.21) π of the angular displacement of the crank.

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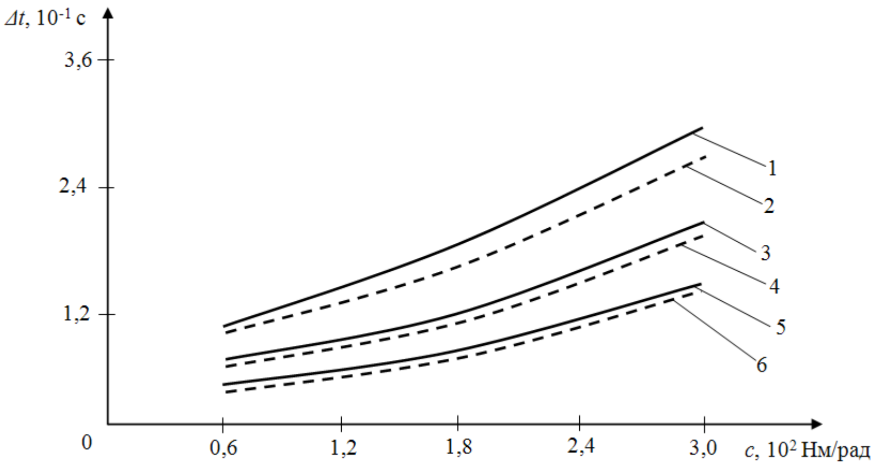

1, 3, 5- taking into account M_{cn}=1,8·10 Nm; 2, 4, 6- without regard toM_{cn};
1, 2-whenω_1=2,2·10^{2}rad/s; 3,4- whenω_1=1,8·10^{2}rad/s;
3- whenω_1=1,1·10^{2}rad/s

Fig. 1.4. Graphic dependences of the change in the dwell time (interaction) of the rocker within the position on the change in the stiffness coefficient of the elastic connection and the frequency of rotation of the crank

IV. EXPERIMENTAL RESULTS

It is important to determine the time of the rocker arm when it interacts with the elastic stops installed in the mechanism housing. In fig. 1.4 graphical dependences of the change in time of the rocker arm in extreme positions on the variation of the stiffness coefficient of elastic bonds are presented. Analysis of the graphs shows that with an increase in the stiffness coefficient of the elastic coupling from 0.9 · 102 Nm / rad to 3.4 102 Nm / rad and at ω1 = 1.8 102 rad / s, the time of the rocker arm in the extreme positions increases from 0 , 06 s to 0.095 s at Mn = 0, and at Mn = 1.8 10 Nm, this time is reduced to 0.045 s. Therefore, to increase the dwell time of the rocker in extreme positions, it is advisable to increase the stiffness of the elastic bonds. Specific values are chosen depending on the technological requirements of the machine. The increase in technological load also includes to some extent the change in the dwell time of the rocker arm (see Fig. 1.4, graphs 1, 3, 5). In addition, an increase in technological resistance leads to an increase in the range of oscillations of the angular velocity of the rocker arm (see Fig. 4.5). So, when you increase$M_{cn}$from 6,0 Nmto 70 Nmwhenc_2=1,2·10^{2}Nm/radswing angular speed rocker ($\Delta\dot{\varphi}_3 = \dot{\varphi}_{3\max} - \dot{\varphi}_{3\min}$)increases from 0,72 rad/sto 2,69 rad/s, whenc_3=2,4·10^{2}Nm/radswing of the angular velocity of the beam increases to 1,52 rad/s.

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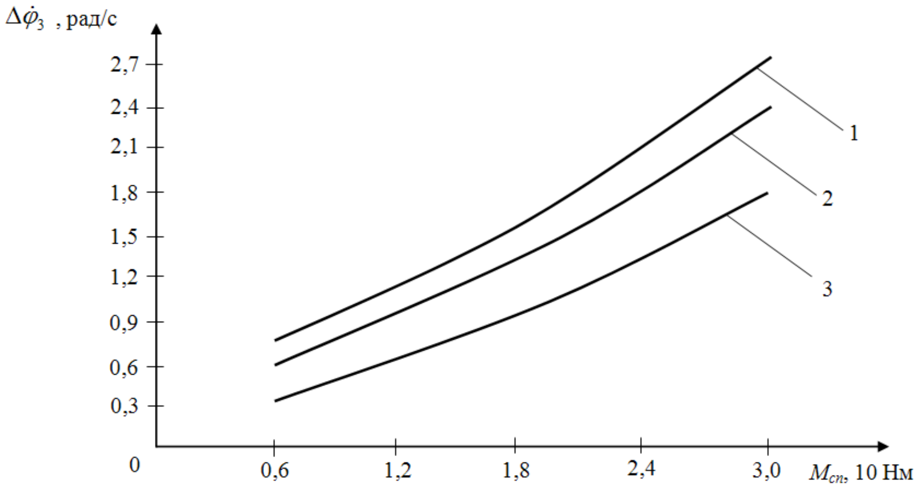
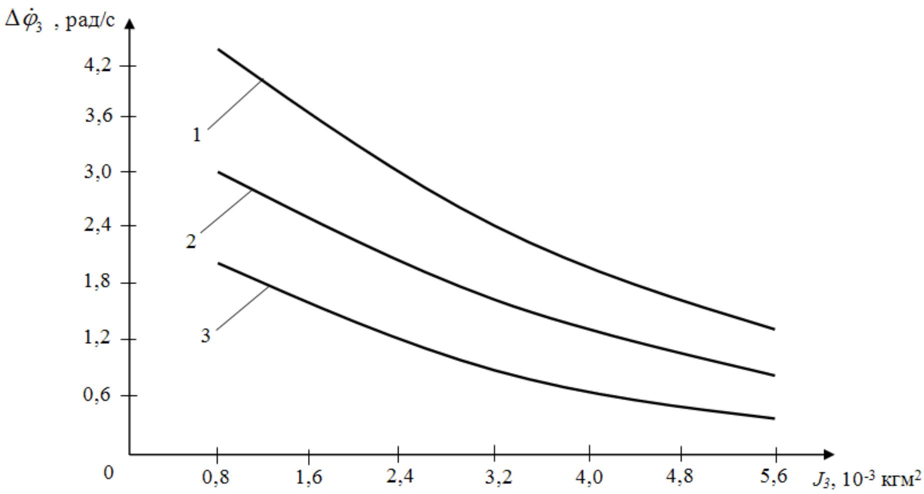


1-whenc_3=1,2·10^{2}Nm/rad; 2- whenc_3=1,8·10^{2}Nm/rad;
3- whenc_3=2,4·10^{2}Nm/rad

Fig. 1.5. Graphic dependences of the change in the amplitude of oscillations of the angular velocity of the rocker on the change in the moment of resistance

It should be noted that the range of oscillations of the angular velocity of the rocker depends largely on its moment of inertia (see. Fig. 4.6).The increase in the moment of inertia of the rocker from 0.8 · 10-3 kgm2 to from 5.6 10-3 kgm2 with c2 = 2.4 102 Nm / rad the range of oscillations of the angular velocity of the rocker arm decreases from 2.1 rad/s to 0, 31 rad/s for non-linear patterns.When reducing the stiffness coefficient of the elastic connection $\Delta\dot{\varphi}_3$ decreases from 4,22rad/sto 2,05 rad/s. Therefore, choosing the necessary values ofJ_3, c_2, M_{cn}, l_1, l_2, $l_3$you can get the required change valuesφ_3, Δt, $\Delta\dot{\varphi}_3$ and others for the corresponding technological machine.



1-whenc_3=1,2·10^{2}Nm/rad; 2- whenc_3=1,8·10^{2}Nm/rad;
3- whenc_3=2,4·10^{2}Nm/rad

Fig. 1.6. Graphic dependences of the change in the amplitude of oscillations of the angular velocity of the rocker from its moment of inertia

The task of the dynamics of the machine unit with a crank-beam mechanism with elastic connections in the extreme positions of the rocker is solved. The laws of the movement of the rocker arm with the change of the stiffness coefficient of the elastic connection are obtained.The dependences of the time variation of the stand and the range of oscillations of the angular velocity of the rocker on the change in the stiffness coefficient of the elastic coupling, the rotational speed and on the change in the moment of resistance and the moment of inertia of the rocker are obtained. With an increase in the stiffness coefficient of the elastic connections, up to 3.0 · 102 Nm / rad, the height of the rocker arm in the extreme positions reaches (0.16-0.21) π of the angular displacement of the crank.

Based on the study of the shear strain of the sleeve of the composite gears, the graphical dependences of change in the deformation of angular displacements of the gear hubs and the wheel on the shafts and the shear modulus are plotted. To reduce the impact interactions of the teeth when meshing due to large values and acceptable values of external moments areM_1=(0,025…0,028)·10^2 Nm, M_2=(0,03…0,036)·10^2Nm. To reduce the deformation values of the angular displacements of the rubber bushings, the values of the shear modulus are considered appropriate (0,33…0,42)·10^{-3}N/m^3.

V.CONCLUSION

A method for calculating the moment of friction force in a fifth-class kinematic pair with longitudinal grooves of the axis is proposed. Formulas for calculating the moment of friction in the kinematic pair are obtained. The laws of change of the friction moment due to the change of the width of the grooves and the length of the kinematic pair are constructed. When changing external load (1,0÷8,0)·10^2Nmoment from frictional forces at a=0,70 increases to 40,5 Nmaccording to a linear pattern, and with the coefficient a=0,40, M_{mp}increases only up to 15,4 Nm. Therefore, the recommended lubrication values are$a$≤(0,4÷0,5).

REFERENCES

[1] A. Dzhuraev and others. Theory of mechanisms of machines. Ed. G. Gulom 2004 p. 582.
[2] E.T. Grigoriev Calculation and design of rubber shock absorbers. M., MashGIZ, 1960, p.157.
[3] Makarov A.I. and others. Calculation and design of spinning machines production engineering, 1963 512 p.
[4]Djuraev, A.Zh.,Davidbaev, B.N., Zhalyayev, A.A., Mirzakhanov, Yu.U. transmission with tensioning roller. // Patent Uz. Res. UZ IAP 4228, 31.03.97. № 1
[5]Dzhuraev, A.Zh.,Davidbaev, B.N., Zhalyaev, A.A., Melamedov, R.Yu., Mirzakhanov Yu.U. Tension roller of flat drive // Patent №50 F M RUz. RA №6, 1996 Belt transmission. Auth. switch № 1767258, BM № 37, 1989