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MATHEMATICAL MODEL OF HYDRAULIC MANIPULATORS OF IMPULSE-FORMING MACHINES

Summary

In work the analysis of construction schemes of hydraulic manipulators is carried out, elements of their classification by power and kinematic indicators are defined. The design scheme of the heavy duty hydraulic manipulator is proposed. The mathematical model of proposed hydraulic manipulator, which can be used as basic data for calculation technical and operational characteristics of earthmoving machines, is obtained. The mathematical model hydraulic control system is described in detail.

Keywords: hydraulic manipulator, impulse machine, earthmoving machines, impulse systems, mathematical model, calculation method.

Hydraulic manipulators are now widely used in various industries and scientific research. Transport-technological mobile machines equipped with hydraulic crane-manipulators, and industrial robots based on hydraulic manipulators are currently one of the most popular and widely used technical devices used in basic areas of the economy to perform basic and auxiliary technological operations, including lifting transport, handling and storage operations [1].

The manipulator is a controlled device or machine for performing motions functions similar to the functions of the human hand when moving objects in space, equipped with a working body. The manipulator consists of links connected by mobile kinematic pairs (rotational and translational) [2].

Designs of hydraulic manipulators and working processes occurring in them for various conditions were investigated and analyzed in the series of works [3-6].

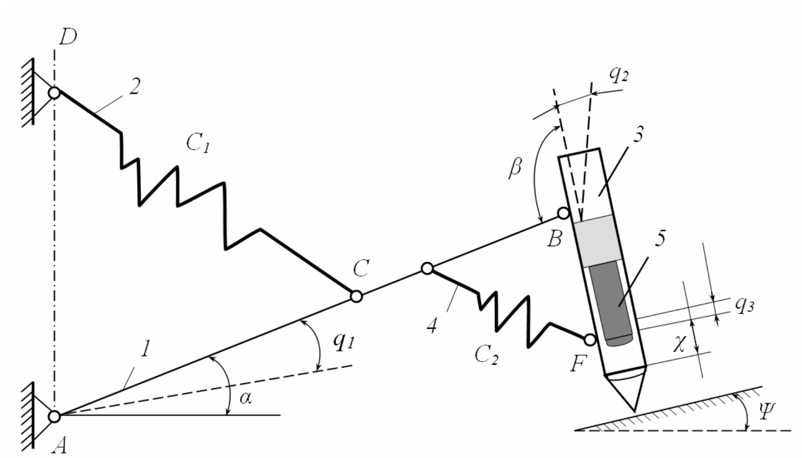
As base points in this case the classification of hydraulic manipulators considering conditions of their application is supposed. It is possible to refer to them: application, degree of loading and degree of mobility

Depending on subjective opinions of designers and researchers the same manipulator can be referred to various groups. For an exclusion of it, i.e. for more accurate differentiation of groups we have introduced the coefficient of loading of K_L representing force relation on executive organ of S to manipulator G weight:

$$K_L = S/G. \qquad (1)$$

Let's establish borders of coefficient change for each of groups:

The hydraulic manipulator scheme that is most appropriate for the operating conditions of the hydraulic impulse machines is proposed (Fig. 2). The scheme possesses necessary degree of mobility and at the same time has small number of joint assemblies and thereof considerable rigidity of the manipulator in general.



1 - a spit; 2 and 4 - hydraulic cylinders; 3 - frame; 5 - pane; q_1,q_2 and q_3 - generalized coordinates

Fig. 2. The design scheme of the manipulator:

For research of mechanical systems with a large number of degrees of freedom Lagrange's equation of the II class is widely used in the form of:

$$\left.\begin{array}{l} \dfrac{\mathrm{d}}{\mathrm{dt}}\left(\dfrac{\partial T}{\partial \dot{q}}\right)-\dfrac{\partial T}{\partial q_1}=Q_1 \\ \text{-----------------------} \\ \dfrac{\mathrm{d}}{\mathrm{dt}}\left(\dfrac{\partial T}{\partial \dot{q}_i}\right)-\dfrac{\partial T}{\partial q_i}=Q_1 \end{array}\right\} \quad (2)$$

where: T is kinetic energy of system; $q_1, q_2 \ldots\ldots q_i$ are generalized coordinates; $\dot{q}_1, \dot{q}_2 \ldots \dot{q}_i$ are generalized speeds; $Q_1, Q_2 \ldots. Q_i$ are generalized force; i is number of degrees of freedom.

When determining the sizes which are entered into system (1) it is accepted that the turn of a spit in the horizontal plane is auxiliary operation of an operating cycle, and while investigating it is lowered. The generalized coordinates of the system are $q_1 = \alpha; q_2 = \beta; q_3 = \mathrm{X}$

Let’s define kinetic energy of the system

$$T = T_1 + T_2 + T_3. \qquad (3)$$

Noting that the spit rotates about the axis passing through point A, the frame of the impact device rotates together with the spit, and the pane makes the difficult movement, we will receive:

$$T_1 = \frac{G_1}{2g}\rho_1^2 \cdot \dot{q}_1^2;$$

$$T_2 = \frac{G_2}{2g}\left(l_2^2 \cdot \dot{q}_1^2 + \rho_2^2 \cdot \dot{q}_2^2\right);$$

$$T_3 = \frac{G_3}{2g}\left\{\left[\dot{q}_3 - q_1 l_2 \cos\left(q_2 - \frac{\pi}{2}\right)\right]^2 + \left[\dot{q}_2(b - q_3) + \dot{q}_1 l_2 \sin\left(q_2 - \frac{\pi}{2}\right)\right]^2\right\}$$

(4)

where G_1,G_2 and G_3 are according to the gravity of a spit, frame and pane; g is acceleration of gravity; ; ρ_1 and ρ_2 are radiuses of inertia of a spit and frame; l_2 is length of a spit, b is an initial deviation of the center of pane masses from frame suspension point. Acceleration phase at reverse motion:

- for pane

$$Q_3^I = P_A - \frac{R_m}{\varepsilon}\left[1 - \frac{q_3}{l_p}(\varepsilon - 1)\right] - G_3 \cos\psi;$$

(5)

- for frame

$$Q_2^I = -F + P_A + G_2 \cos\psi - \frac{R_m}{\varepsilon}\left[1 - \frac{l_p - q_3}{l_p}(\varepsilon - 1)\right] - G_3 q_2 \frac{l_3 \sin\beta}{\sqrt{l_3^2 + l_4^2 - 2 l_3 l_4 \cos\beta}}$$

(6)

where P_A is driving force in the camera of reverse motion; R_m is maximum reaction force; ε is extent of gas compression in the pneumatic camera; l_p is the size of pane operating course; F is force of pressing soil; l_3 and l_4 are distance to suspension points of a hydraulic cylinder of the frame rotation. Deceleration phase at reverse motion:

- for pane

$$Q_3^{II} = -\frac{R_m}{\varepsilon}\left[1 - \frac{q_3}{l_p}(\varepsilon - 1)\right] - G_3\cos\psi; \tag{7}$$

- for frame

$$Q_2^{II} = -G_2 cos\psi + \frac{R_m}{\varepsilon}\left[1 - \frac{l_p - q_3}{l_p}(\varepsilon - 1)\right] - G_2 q_2 \frac{l_3\sin\beta}{\sqrt{l_3^2 + l_4^2 - 2l_3 l_4\cos\beta}} \tag{8}$$

Phase of the operating course:

- for pane

$$Q_3^{III} = -R_m\left[1 - \frac{q_3}{l_p}\cdot\frac{(\varepsilon - 1)}{\varepsilon}\right] - G_3\cos\psi; \tag{9}$$

- for frame

$$Q_2^{III} = -G_2 cos\psi + R_m\left[1 - \frac{l_p - q_3}{l_p}\frac{(\varepsilon - 1)}{\varepsilon}\right] - c_2 q_2 \frac{l_3\sin\beta}{\sqrt{l_3^2 + l_4^2 - 2l_3 l_4\cos\beta}} \tag{10}$$

The generalized force for a spit doesn't depend on phases of the pane movement and is defined according to the received equation:

$$Q_1 = -c_1 l_2 q_1\cos\left(\alpha + \psi\right) - c_1 l_2\cos\alpha - c_2 q_1 \frac{\left(l_1 - l_2\right)\sin\beta}{\sqrt{l_3^2 + l_4^2 - 2l_3 l_4\cos\beta}} \tag{11}$$

where c_1 and c_2 are stiffness coefficients of two hydraulic cylinders (Fig. 1)

So the system of the differential equations of the second order describing the movement of the manipulator in each of phases of operating cycle is obtained:

- in acceleration phase of pane at reverse motion:

$$\begin{cases} \frac{\ddot{q}_3}{l_2}(cq_3 - a - b + d) + \frac{c}{l_2}\dot{q}_3^{\,2} - \frac{c_1 f + c_2 k}{l_2} l_2 q_3 = -k - \frac{l_p}{j_2}(c_1 f + c_2 k); \\ \frac{\ddot{q}_3}{l_2}(s - cq_3 + i - jq_3 + m_3 q_3^{\,2}) + \frac{\dot{q}_3 q_3 c}{l_2} + \frac{\dot{q}_3^{\,2}}{l_2}\left(\frac{y - z}{l_2} + iq_3\right) - q_3\left(\eta - c_2\frac{\lambda}{l_2}\right) = 0; \\ \frac{\ddot{q}_3}{l_2}\left(\frac{d}{l_2} + m_3\right) + \frac{\dot{q}_3^{\,2}}{l_2}\left(\frac{b}{2} - m_3 q_3 + d\right) + \eta q_3 = -\chi \end{cases}$$

(14)

- in phase of the operating course of pane:

$$\begin{cases} \frac{\ddot{q}_3}{l_2}(cq_3 - a + b + d) + \frac{c}{l_2}\dot{q}_3^{\,2} - \frac{c_1 f + c_2 k}{l_2} q_3 = -h - \frac{l_p}{l_2}(c_1 f + c_2 k); \\ \frac{\ddot{q}_3}{l_2}(i - b - (c + j)q_3) + \frac{\dot{q}_3^2}{l_2^2}(m_3 q_3^{\,2}(j + c - d)q_3 - 2(c + d) - y - b + 2) + \\ +c_2 \lambda q_3 = \eta \\ \frac{\ddot{q}_3}{l_2}\left(d + m_3 l\right) + \frac{\dot{q}_3^{\,2}}{l_2}\left(2 - b - \frac{\gamma}{2} q_3\right) + \eta q_3 = H \end{cases} \quad (15)$$

At the same time statement and the solution of two problems is possible. The first, or direct: to determine power and energy indicators of the shock mechanism by the set design parameters and kinematic indicators of the shock mechanism. The second, or inverse: to lay down design parameters of the manipulator and its kinematic parameters on the set energy and power indicators of the shock mechanism.

Mathematical modeling of the hydraulic manipulator for the working conditions of the hydraulic impulse machines has been carried out. A mathematical model containing differential equations of motion of the main

links was compiled. The design scheme of the hydraulic manipulator is described. The system under consideration is a mathematical model of a hydraulic shock mechanism. The proposed mathematical model can be used to optimize the designs developed hydraulic shock mechanisms

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