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CALCULATION OF OPERATING MODES OF CARDS TRANSFERS OF THE EXPERIMENTAL DIGGER FOR HARVESTING TOPINAMBUR

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Annotation

The work of the cardan drive of a prototype digger for harvesting Jerusalem artichoke is characterized by a very difficult mode, large angles between the hinge shafts, which change during operation, dustiness and aggressiveness of the environment.

Due to the heavy duty operation of the cardan joints and improper operation, the most common violations are:

1. Failure to comply with the conditions for assembling the telescopic device of the cardan shaft, as a result of which very large angles appear between the forks of the intermediate shaft.
2. Violation of the regime of lubrication of cardan joints.

Even with proper operation of universal joints, their service life does not exceed 300 hours, although the manufacturer guarantees 600 hours, provided that the angle between the hinge shafts does not exceed 15 °.

The most typical causes of universal joint failures:

- the formation of dents on the surface of the spikes of the crosses along the generatrices of the cylinder or at an angle;
- wear of the surface of the spikes;
- deformation of crosspieces and forks of hinges;

Figure 1 shows a typical waveform. The figure shows the frequency of oscillation of the moment strictly coincides with the frequency of rotation of the shaft. The value of the ratio of the maximum value of the moment (Mmax) to the minimum value (Mmin) varies in a fairly wide range - from 3 to 10 times, depending on the soil entering the Elevator. Such torque fluctuations have a significant impact on reducing the service life of the cardan joints of a prototype digger for harvesting Jerusalem artichoke.

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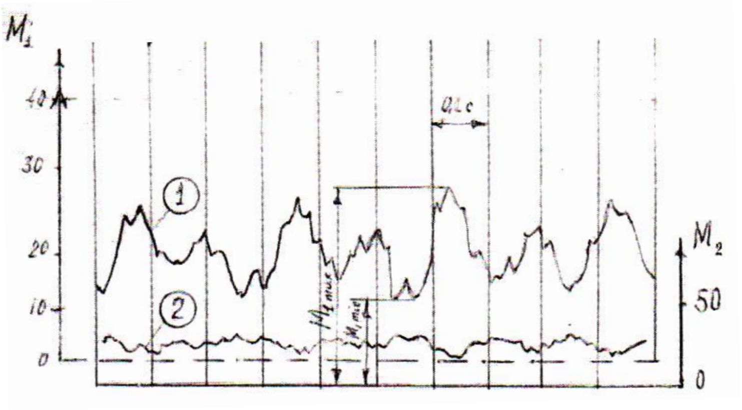

Fig.1 Characteristic oscillogram frequency of oscillations of the moment.

To identify the reasons for such a sharp change in the magnitude of the torque transferred by the cardan shaft from the PTO of the tractor to the prototype digger for harvesting Jerusalem artichoke, the theoretical studies of the kinematics and dynamics of the driveline were carried out.
With a constant angular velocity of the driving link, the angular velocity of the driven link is not constant, it is expressed by the equation:

$$\omega_2 = \omega_1 \frac{\cos\alpha}{1+\cos^2\gamma \cdot \sin^2\gamma} \quad (1)$$

where: 1 and 2 are the angular velocities of the driving and driven shaft, respectively;
α- angle of rotation of the drive shaft;
γ- is the angle between the drive and driven shafts of the transmission.
It is known that the spread of the maximum and minimum angular velocity of the driven shaft, referred to the average angular velocity of this link, is called the non-uniformity coefficient:

$$\delta = \frac{\omega_2 - \omega_1}{\omega cp}$$

For the cardan transmission of a prototype digger for harvesting Jerusalem artichoke, the angular velocities of the driving and driven shafts are constant under the following conditions:
1. The angles between the shafts connected by hinges must be equal, i.e. γ1=γ2
2.All transmission shafts lie in the same plane of the driven fork of the leading joint and the leading fork of the driven joint is zero.

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Since in a real transmission these conditions cannot be met, this means that the angular velocities of the driving and driven shafts are not equal. Then the ratio of angular velocities

$$\frac{\omega_2}{\omega_1}=\frac{\cos\gamma_1\cdot\cos\gamma_2\left(1+tg^2\gamma_3\right)\left(1+tg^2\gamma_3\cdot\cos^2\gamma_2\right)}{\sin^2\alpha\cdot\cos^2\gamma_2\left(1+tg^2\gamma_3\right)^2+\left[\cos\alpha\cdot\cos\gamma_1\left(1+tg^2\gamma_3\cdot\cos^2\gamma_2\right)-\sin\alpha\cdot\sin^2\gamma_2 tg\gamma_3\right]^2}$$

In this expression denoting:

$$A=\cos\gamma_1\cdot\cos\gamma_2\left(1+tg^2\gamma_3\cos^2\gamma_2\right)$$

$$B=\cos^2\gamma_2\left(1+tg^2\gamma_3\right)^2$$

$$C=\cos\gamma_2\left(1+tg^2\gamma_3\cdot\cos^2\gamma_2\right)^2$$

$$Д=\sin^2\gamma_2\cdot tg^2\gamma_3$$

Get

$$\omega_2=\omega_1\frac{A}{\sin^2 a\cdot B+\left[C\cos\alpha\cdot Д\sin\alpha\right]^2} \qquad (2)$$

где γ_1,γ_2-углы между валами ведущего и ведомого шарнира;

γ_3-угол между вилками промежуточного вала;

$\alpha=\omega t$ -угол поворота ведущего вала.

Дифференцируя выражение (2) по времени, получим зависимость для определения углового ускорения на ведомом валу (J_2). Далее, на основании равенства инерционных моментов возникающих на ведущих и ведомых валах передачи имеем по Чудакову [1, 2, 3]:

$$J_2 j_2=J_1 j_1 \qquad (3)$$

where J1J2-corresponding to the moment of inertia of the masses of the driving and driven shafts.

Hence the value of the inertial moment on the driven shaft:

$$Mj=J_2 j_2=J_2\omega_1^2\frac{\sin 2a\left(b+c^2+d^2\right)-2cd\cos^2\alpha}{\left\{b\sin^2\alpha+\left[c\cos\alpha-d\sin\alpha\right]^2\right\}^2} \qquad (4)$$

Depending on different values of γ1, γ2, γ3, it is possible to construct a family of curves that determine the nature of changes in inertial moments.

Figure 2 shows one of the curves of this family: the change in moment is sinusoidal in nature with an oscillation period T=2ω.

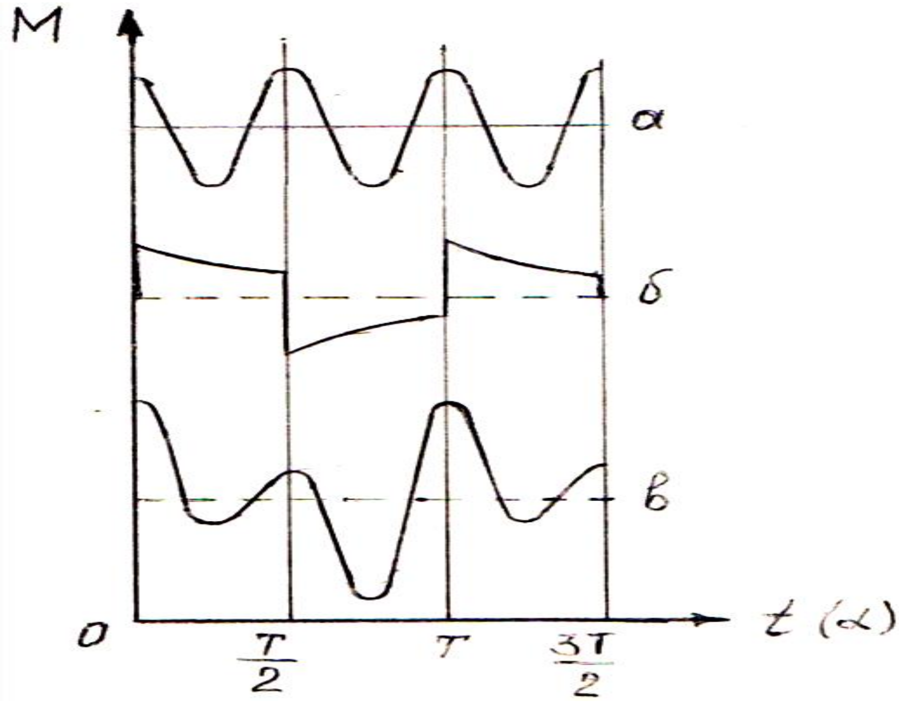

Fig.2. Change of moments

In a cardan transmission, there are blows in the cross in one period, the system receives two (opposite) blows, equal in magnitude, but opposite in direction, as can be seen in Fig. 2.

Summing up graphs (a) and (b), we obtain a curve characterizing the change in the additional torque that occurs in the driveline. The curve (Fig. 2) obtained theoretically is very similar in character to the oscillograms obtained during field measurements (Fig. 1). Since dynamic loads vary over a wide range, they will certainly have a negative impact on the operation of the ability of both the transmission itself and the components of the entire machine.

One of the most effective ways to reduce dynamic loads is to introduce elastic elements into the kinematic scheme of the driveline. For a qualitative assessment of the influence of elastic elements on the operation of the driveline, the differential equation [4, 5,6]

$$J_0 \ddot{\varphi} + x\dot{\varphi} + k\varphi = M + f(t) \quad (5)$$

$$\varphi = \varphi_0 + \varphi_1$$

Here

φ - total angle of twist of the intermediate shaft;

φ0-angle of twist of the intermediate shaft from the variable and constant parts of the moment;

x- coefficient taking into account the ability to dampen vibrations;

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k is the elasticity of the system;
M; f (t) is a constant variable of the load part.
The solution of the differential equation (5) has the form:
a) taking into account elasticity and attenuation.

$$M_{gon} = k\varphi = \frac{kf_0(k - J_0\omega_2^2)\sin\omega t}{\left(k - J_0\omega_1^2\right)^2 + x^2\omega_1^2} - \frac{}{\left(k - J_0\omega^2\right)^2 + x\omega_1^2}\cdot\cos\omega t$$

b) with absolutely rigid shafts

$M_{gon} = f_0 \sin\omega t$

Let us study the limiting cases of system elasticity:

1 for $k \to \infty$ $\left(k \rangle\rangle J_0 \cdot \omega_2^1\right)$ and $x \to 0$ we have $M_{gon}^1 = Mgon$

2 for $\kappa \to J_0\omega_1^2$ $\left(k - J_0 \cdot \omega_2^1\right) = J_{_0}\omega_2^1$ and $x \to 0$ we have M_{gon}^1 =2 $Mgon$

3 for $\kappa \to 0$ $\left(k \langle\langle J_0 \cdot \omega_1^2\right) = J_{_0}\omega_2^1$ and $x \to 0$ we have $M_{gon}^1 \rangle\rangle Mgon$

4. when $\kappa \to J_0\omega_1^2$ $M_{gon}^1 \rangle\rangle Mgon$ we have (resonance)

5 when $x\omega_1 \rangle\rangle \left(K - J_0\omega_1^2\right)$ we have $\left(M_{gon}^1 \rangle\rangle M_{gon}\right)$

Based on the analysis of the limiting cases, it can be concluded that the introduction of an elastic element, which will reduce the rigidity of the shaft (k), will lead to a sharp decrease in the additional moment.
Based on the research, it can be recommended as additional elements introduced into the transmission torsion or rubber dampers, which ensure stable and long-term operation of the cardan drive of a prototype digger for harvesting Jerusalem artichoke.

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