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Kinematic and Dynamic Study of the Rotary Crusher Transmission Gear in the Potato Digger

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ABSTRACT

Theoretical study of the geometric and kinematic as well as dynamic parameters of the gear drive wheel rotary soil crusher of the potato digger with planetary mechanism has been implemented and the optimal parameters have been selected.

In the result of field and laboratory experiments it has been revealed that the distance between the grouser bars of the drive gear wheel should be approximately 0.17 m, thus, the angle formed by the adjacent grouser bars makes 180, while their number should be 20.

Introduction

Different soil and clod crushers are used in order to increase the level of sifting potato and soil mass when harvesting with potato diggers. The main source of drive gear for the latter is the shutter shaft of the tractor capacity. The rotary crusher of the potato digger developed by us (Tarverdyan, Yesoyan and others, 2018) takes the movement from the gear wheel anchored in the soil. In the result effective horsepower of the tractor is saved and the productivity of the tractor and aggregate increases. The rotary crusher of the potato digger 1 (picture 1) is fixed to the front part of the digger through lever pivot-hinged system 2. In the transport state of the aggregate the crusher comes off from the ground with the help of the suspension system of the tractor together with the potato digger and in the working state it is settled on the soil bed border. The regulatory device regulates the sticking degree of working organs and soil grousers into the soil. The regulator device is a hydraulic cylinder 3 controlled by tractor distribution system which is connected with the potato digger through suspenders on the one side, and on the other side – with the horizontal bar connecting the levers (Picture 1). In order to give torque movement to the working rotors 5 from the gear wheel 4 of the crusher anchoring in the soil, four-bar planetary gear with parasitic tooth chisel was chosen in the result of studies and analysis.

The mechanism (Picture 2) consists of towing and steering arm 4, which gets the torque movement from the carrying wheel anchoring in the soil, parasitic wheels 2, central carrying wheel 3, the torque movement of which is given through half shafts to the rotors of the crusher and from the fixed internal annulus 1 which at the same time serves as mechanism casing. The numbers of the toothed gear are Z_1, Z_2 and Z_3 respectively.

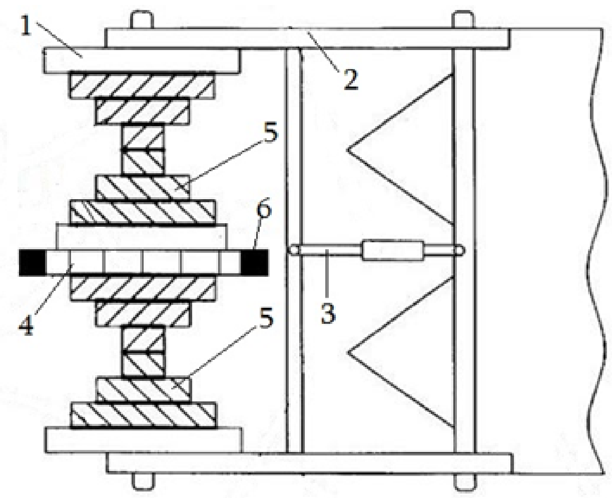

Figure 1. The scheme of the Rotary Crusher of the Potato Digger with Gear Drive Wheel *(composed by the authors).*

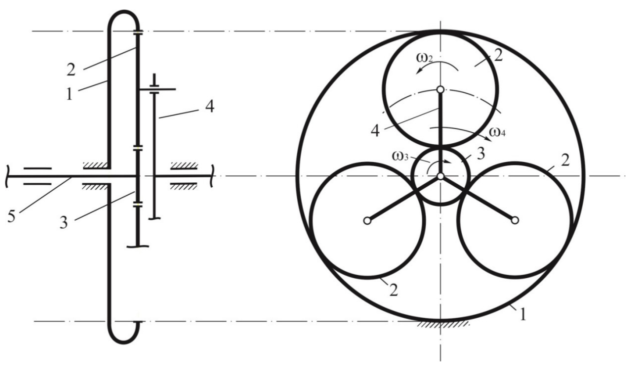

Figure 2. The Scheme of the Four-Bar Planetary Mechanism *(composed by the authors).*

During the forward movement of the potato digger the wheel 4 (Picture 1), due to the resistance of the soil grousers 6 penetrating into the soil, when turning around also turns the carrier 4 (Picture 2) together. The carrier turning with gearing number i (i>1) turns the annular ring 3 (Picture 2) in the direction of its rotation. In the result the annular ring with the help of half shafts 5 (picture 2) rotates the crusher with a bigger rotation numbers which turns into active rotary working organ (Tarverdyan, Yesoyan and others, 2018).

Materials and methods

The effective operation of the crusher is conditioned mainly by the right selection of planetary mechanism and calculation of optimal geometric and kinematic parameters. The important parameter of the mechanism is the transmission number.

The correlation of the transmission from the 4th carrying ring to the 3rd tooth gear is defined through the following expression (Artobolevskiy, 1988, Zinovyev, 1975):

$$i_{43}^{(1)} = 1 + \frac{z_1}{z_3} \quad \text{or}$$

$$\omega_3 = \omega_4 \left(1 + \frac{z_1}{z_3}\right). \qquad (1)$$

The rotation directions of the carrying wheel and crusher rotors coincide. And the selection of the abovementioned option of the driving mechanism with parasitic tooth gear is conditioned by this as the crushed soil clods should be thrown away in the direction opposite to the aggregate movement.

For the effective grinding of soil clods and throwing them away at about 9-10 m/s speed (the experiments showed that at this and higher speeds the soil clods grinding is significantly intensified in the result of striking) the transmission number of the planetary mechanism should be more than 8 (Tarverdyan, 2014, Sineokov, Panov, 1977).

In the result of the primary calculation it has been found out that it is expedient to choose 8.2 (this number is limited by the mechanism size and mass). The transmission number from the third ring to the first in case of immobility of the carrier will be:

$$i_{31}^{(4)} = 1 - 8.2 = -7.2 \quad \text{or}$$

$$i_{31}^{(4)} = -\frac{z_1}{z_3} = -7.2 \qquad z_1 = 7.2 z_3. \qquad (2)$$

From coaxiality condition

$$z_2 = \frac{z_1 - z_3}{2} = 3.1 \cdot z_3 \, , \qquad (3)$$

we will obtain from (2) and (3):

$$\frac{z_1}{z_2} = \frac{7.2 z_3}{3 z_3} = 2.4 \, . \qquad (4)$$

The tooth number Z_1 of the 1st immobile annular ring should be chosen in a way so that the phenomena of undercutting and interference of the teeth are excluded (Artobolevskiy, 1988, Anurev, 1973). The mentioned requirements are met if $Z_1 \geq 60$ (Artobolevskiy, 1988). Let's consider that $Z_1 \geq 62$, in that case $z_3 = \frac{z_1}{7.2} = 8.6$, but Z_3 cannot be less than 13 (Artobolevskiy, 1988, Anurev, 1973), hence considering Z_1=15, we will define Z_1 and Z_2 retrospectively taking into account the coaxiality condition. We will obtain Z_1=105, Z_2=45.

Together with meeting the necessary conditions of the required transmission number and module, the conditions of mechanism assembly, satellite vicinity and transmission coaxiality should be met as well which has the following expression for the suggested mechanism:

$$z_1=2z_2+z_3,$$

which is also met $(105 = 2 \cdot 45 + 15)$.

The optimal number of satellites is defined through the following expression:

$$k = \frac{\pi}{\arcsin \frac{z_2+2}{z_3+z_2}} = \frac{\pi}{\arcsin 0.75} = 3.71 .$$

As K<4 (Artobolevskiy, 1988, Zinovyev, 1975) we consider k=3satellite.

For the kinematic and dynamic analysis of the mechanism it is necessary to define the radii of the dividing circumferences of the tooth gear (let's consider the module of the tooth gear m=10), we will obtain:

$$r_1 = \frac{m z_1}{2} = \frac{10 \cdot 105}{2} = 525 \text{ mm}$$

$$r_2 = \frac{m z_2}{2} = \frac{10 \cdot 45}{2} = 225 \text{ mm}$$

$$r_3 = \frac{m z_3}{2} = \frac{10 \cdot 15}{2} = 75 \text{ mm.}$$

Checking the coaxiality condition is done: $r_1 = 2r_2 + r_3 = 2 \cdot 225 + 75 = 525$ mm.

The coefficient of the mechanism is defined through the following expression:

$$\eta = 1 - \left| \left(1 - \frac{1}{i_{43}^{(1)}} \right) \right| \psi ,$$

where $\psi = 0.05$ is the coefficient of the losses, placing $i_{43}^{(1)} = 8.2$ we will obtain $\eta = 0.96$.

In order to make a dynamic analysis of the mechanism, it is necessary to identify the speed of the characteristic points of the planetary mechanism (Bat, Djanilidze and others, 1975, Loytsanskiy, Loureh, 2006). The radii of the dividing circumferences of the tooth gear r_1, r_2, r_3 and rotational velocity of the carrier ω_4 are known in the problem under discussion, it is necessary to identify the rotational velocity ω_3 of annular ring 3, the momentary velocity of the satellite and the speed of points A, B, and C (picture 3).

For the suggested mechanism the 3- annular ring and 4-carrier are rotating around the immobile axis, and the satellites make smooth movement. The speed of the O_1 center of the satellite rotation as carrier point will be:

$$V_{O_1} = (r_3 + r_2) \cdot \omega_4 . \tag{5}$$

Taking into account the O_2 momentary center of the satellite velocity:

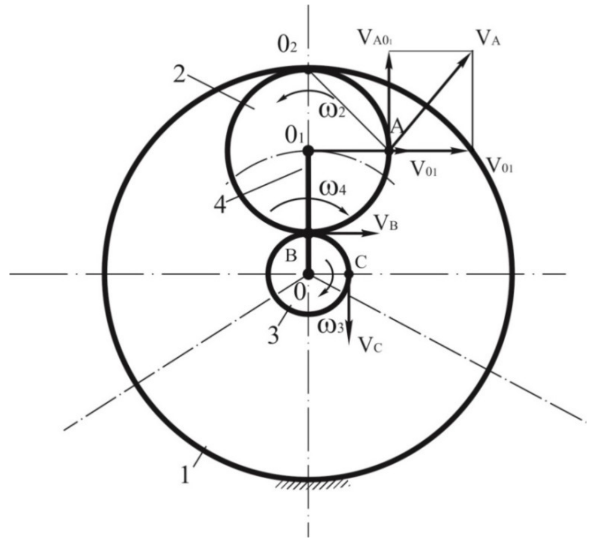

Figure 3. The Scheme of the Identification of the Speed of Characteristic Points of the Planetary Mechanism *(composed by the authors).*

$$V_{O_1} = (O_1 O_2) \cdot \omega_2 ,$$

or

$$\omega_2 = \frac{V_{O_1}}{O_1 O_2} = \frac{\omega_4 (r_2 + r_3)}{r_2} . \tag{6}$$

The velocity of Point B of 3- annular ring will be:

$$V_B = \omega_2 \cdot 2r_2 ,$$

and

$$\omega_3 = \frac{V_B}{r_3} = \frac{\omega_2 \cdot 2r_2}{r_3} . \tag{7}$$

The velocity of point A belonging to the sattelite will be:

$$V_A = \omega_2 \cdot O_2 A = \omega_2 \cdot r_2 \sqrt{2} .$$

The velocity of point C belonging to 3- annular ring will be:

$$V_C = \omega_3 \cdot r_3 = \frac{\omega_2 \cdot 2 r_2 r_3}{r_3} = \omega_2 \cdot 2r_2 . \tag{8}$$

After the identification of the velocities it is necessary to calculate the momentum of inertia of the mechanism to the 3rd ring from where the momentum and rotary movement are transferred to the working rotors.

The expression of the kinetic energy of the suggested mechanism will be as follows (Bat, Djanilidze and others, 1975, Loytsanskiy, Loureh, 2006):

$$W = W_4 + 3W_2 + W_3 , \tag{9}$$

where W_4 is the kinetic energy of the carrier:

$$W_4 = \frac{I_4 \omega_4^2}{2} ,$$

W_2 is the kinetic energy of the satellite, as they make smooth parallel movement, hence:

$$W_2 = \frac{I_2\omega_2^2}{2} + \frac{m_2 V_{O_1}^2}{2}$$

W_3 is the kinetic energy of the annular rings:

$$W_3 = \frac{I_3\omega_3^2}{2}.$$

The kinetic energy brought to the ring is as follows:

$$W = \frac{I^*\omega_3^2}{2}, \quad (10)$$

where I^* is the momentum of inertia of the mechanism.

Inserting the values into (8) and equaling the right parts of (8) and (10) (the value of V_{O_1} is inserted from the (5)) we will obtain:

$$\frac{I_4\omega_4^2}{2} + 3\left[\frac{I_2\omega_2^2}{2} + \frac{m_2(r_2+r_3)^2\omega_4^2}{2}\right] + \frac{I_3\omega_3^2}{2} = \frac{I^*\cdot\omega_3^2}{2}. \quad (11)$$

After the modification we will get the following for the given momentum of inertia:

$$I^* = I_3 + 3I_2\left(\frac{\omega_2}{\omega_3}\right)^2 + 3m_2(r_2+r_3)^2\left(\frac{\omega_4}{\omega_3}\right)^2 + I_4\cdot\left(\frac{\omega_4}{\omega_3}\right)^2. \quad (12)$$

Results and discussions

M_3 is transferred to the mechanism through 2 half shafts of the 3- annular ring. Besides the centrifugal inertia moment of satellites P_{i2} as external moment factors affect the mechanism which is applied conditionally at O_2 point and the P_{i4} centrifugal inertia moment of carrier 4 which is applied at the center of gravity of the carrier along the longitudinal axis O_3. It is known that:

$$P_{i2} = m_2\omega_4^2\left(r_3 + r_2\right),$$
$$P_{i4} = m_4\omega_4^2\left(OO_3\right). \quad (13)$$

The analysis of the problem brings to the carrier, hence to the identification of the M_4 momentum balancing M_3 applied upon the driving grouser wheel.

Let's consider the balance condition of the 3- annular ring (Picture 4):

$$\sum M = 0$$

or

$$M_3 - P_{32}\cdot r_3\cos\alpha = 0, \quad (14)$$

Where α is the angle formed by the normal n-n of the profiles of satellite teeth and annular ring and the common tangent t-t of the initial circumferences of those tooth gear. From the (14):

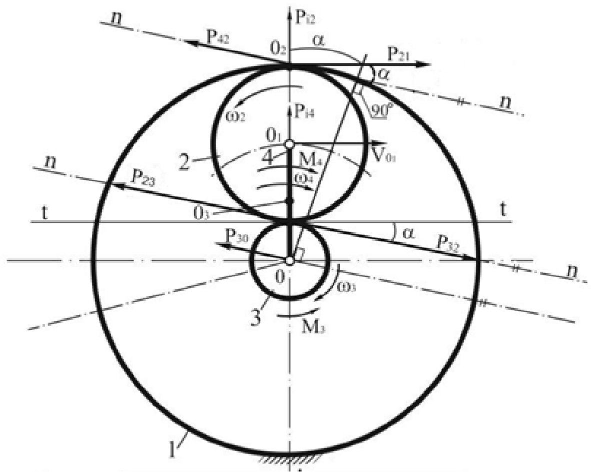

Figure 4. The Scheme of the Dynamic Analysis of the Planetary Mechanism *(composed by the authors).*

$$P_{32} = \frac{M_3}{r_3\cos\alpha}. \quad (15)$$

From the balance condition of the forces influencing 3-annular ring we define the axial force of that wheel:

$$P_{30} = -P_{32}. \quad (16)$$

Considering the last expression and from the balancing conditions of the forces influencing the satellite we define the correlation force of the satellites and the carrier, P_{24}.

$$\overline{P}_{21} + \overline{P}_{23} + \overline{3P}_{i2} + \overline{P}_{24} = 0,$$

where $P_{21} = -P_{23}\cos\alpha$. Placing the values from the (13) and (15) we will obtain:

$$-P_{23}\cos\alpha - \frac{M_3}{r_3\cos\alpha} + 3m_2\omega_4^2\left(r_3 + r_2\right) + P_{24} = 0, \quad (17)$$

from this $P_{24} = -P_{42}$ is defined. The balancing condition of the carrier defines the balancing momentum M_4.

$$M_4 + P_{42}\cdot\left(r_3 + 2r_2\right)\cos\alpha = 0,$$

where α is the angle formed by P_{42} force direction and the perpendicular taken to that direction from the carrier axis:

$$M_4 = -P_{42}\cdot\left(r_3 + 2r_2\right)\cos\alpha,$$

or by inserting the value of P_{42} from the (17) we will obtain:

$$M_4 = \left[\frac{M_3}{r_3}\left(\frac{1}{\cos\alpha} + 1\right) - 3m_2\omega_4^2\left(r_3 + r_2\right)\right]\cdot\left(r_3 + 2r_2\right)\cos\alpha.$$

The momentum of M_4 from the carrying wheel is applied to the carrier 4 which through rotation moves the satellites.

For the dynamic analysis of the mechanism as well as the grinding it is necessary to identify the angular acceleration of the carrier. The carrier weight is P_4, satellite – P_2, annular ring – P_3 (Picture 5). The position of any ring of the mechanism is defined by the generalized coordinate φ, for which the Lagrangian formula is the following (Bat, Djanilidze and others, 1975).

$$\frac{d}{dt}\frac{\partial W}{\partial \dot{\varphi}} - \frac{\partial W}{\partial \varphi} = P_{\varphi} . \quad (19)$$

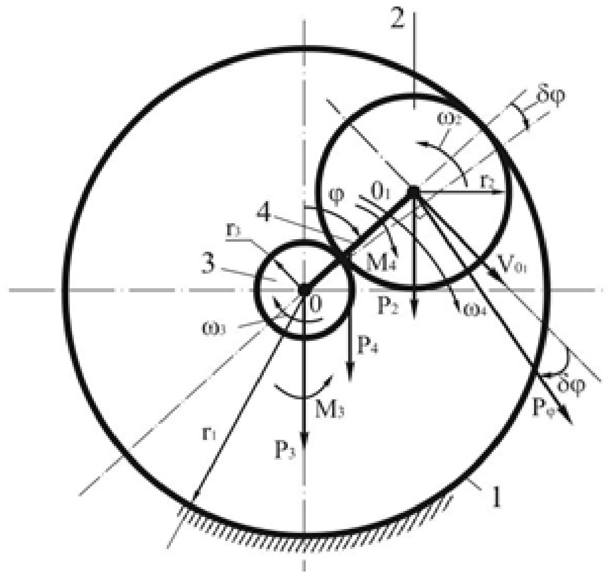

Figure 5. The Scheme of the Identification of the Angular Acceleration of the Carrier in the Planetary Mechanism *(composed by the authors).*

The active forces in the system are P_4, P_2, P_3, and momentum M_4.

Let us give the angle φ elemental growth of δ_φ (clockwise). In order to define the generalized force P_φ, we need to calculate the work done by the active forces influencing the elemental movement δ_φ.

$$\delta_A = M_4 \cdot \delta\varphi + P_4 \cdot \left(\frac{r_3 + r_2}{2}\right) \cdot$$
$$\cdot \sin\varphi \cdot \delta_\varphi + P_2 \cdot \left(r_3 + r_2\right) \sin\varphi \cdot \delta_\varphi ,$$

or

$$\delta_A = \frac{1}{2}\left[2M_4 + \left(P_4 + 2P_2\right)\cdot\left(r_3 + r_2\right)\sin\varphi\right]\cdot \delta_\varphi . \quad (20)$$

Taking into consideration that according to the definition $\delta_A = P_\varphi \cdot \delta_\varphi$ we can express (20) in the following way:

$$P_\varphi = \frac{1}{2}\left[2M_4 + \left(P_4 + 2P_2\right)\cdot\left(r_3 + r_2\right)\sin\varphi\right] . \quad (21)$$

The kinetic energy of the entire mechanism to the given rotation axis of the 4th rim is defined by the following expression ($\varphi = \varphi_4$):

$$W = \frac{I^* \cdot \omega_4^2}{2} = \frac{I^* \cdot \left(\dot{\varphi}\right)^2}{2} . \quad (22)$$

According to the expression (9) the kinetic energy of the entire mechanism will be:

$$W = \frac{I_4 \omega_4^2}{2} + \frac{3 I_2 \omega_2^2}{2} + \frac{3 m_2 V_{O_1}^2}{2} + \frac{I_3 \omega_3^2}{2} , \quad (23)$$

where $I_4 = \frac{1}{3} \cdot \frac{P_4}{g}\left(r_3 + r_2\right)^2$, $I_2 = \frac{P_2 r_2^2}{2g}$, $m_2 = \frac{P_2}{g}$,

$V_{O_1} = \omega_2 r_2$ $\quad I_3 = \frac{P_3 \cdot r_3^2}{2g}$.

Taking into account that $\omega_4 = \omega = \dot{\varphi}$, according to (6) and (7) we will have the following expressions for ω_3 and ω_2:

$$\omega_3 = \omega_4\left(1 + \frac{r_1}{r_3}\right) = \frac{\left(r_1 + r_3\right)}{r_3} \cdot \dot{\varphi} ,$$

and

$$\omega_2 = \frac{\omega_4\left(r_2 + r_3\right)}{r_2} = \frac{\left(r_2 + r_3\right)}{r_2} \cdot \dot{\varphi} .$$

Inserting the values into the (23) we will obtain the following for the kinetic energy:

$$W = \frac{P_4\left(r_3 + r_2\right)^2}{6g} \cdot \left(\dot{\varphi}\right)^2 + \frac{3 P_2 r_2^2 \left(r_2 + r_3\right)^2}{4g \cdot r_2^2} \cdot \left(\dot{\varphi}\right)^2 +$$
$$+ \frac{3 P_2 r_2^2 \left(r_2 + r_3\right)^2}{2g \cdot r_2^2} \cdot \left(\dot{\varphi}\right)^2 + \frac{P_3 r_3^2 \left(r_1 + r_3\right)^2}{4g \cdot r_3^2} \cdot \left(\dot{\varphi}\right)^2 ,$$

or

$$W = \frac{1}{12g}\left[\left(2P_4 + 27P_2\right)\left(r_2 + r_3\right)^2 + 3P_3\left(r_1 + r_3\right)^2\right] \cdot \left(\dot{\varphi}\right)^2 . \quad (24)$$

The partial derivative of the kinetic energy according to the generalized velocity $\left(\dot{\varphi}\right)$ will be:

$$\frac{\partial W}{\partial \dot{\varphi}} = \frac{1}{6g}\left[\left(2P_4 + 27P_2\right)\left(r_2 + r_3\right)^2 + 3P_3\left(r_1 + r_3\right)^2\right] \cdot \dot{\varphi} . \quad (25)$$

Let us derive the obtained result according to the time, we will have:

$$\frac{d}{dt}\frac{\partial W}{\partial \dot{\varphi}} = \frac{1}{6g}\left[\begin{array}{l}\left(2P_4 + 27P_2\right)\cdot \\ \cdot\left(r_2 + r_3\right)^2 + 3P_3\left(r_1 + r_3\right)^2\end{array}\right]\cdot \ddot{\varphi}. \quad (26)$$

Inserting (21) and (26) expressions into the Lagrangian equation and considering that $\frac{\partial W}{\partial \varphi} = 0$ (according to (24) kinetic energy does not depend on the generalized coordinate φ) we will obtain the differential equation of the mechanism movement for the generalized coordinate φ :

$$\frac{1}{6g}\left[\left(2P_4 + 27P_2\right)\left(r_2 + r_3\right)^2 + 3P_3\left(r_1 + r_3\right)^2\right]\cdot \ddot{\varphi} =$$
$$= \frac{1}{2}\left[2M_4 + \left(P_4 + 2P_2\right)\left(r_3 + r_2\right)\sin\varphi\right],$$

where we define the required angular acceleration $\ddot{\varphi}$ of the carrier from:

$$\ddot{\varphi} = 3g\frac{\left[2M_4 + \left(P_4 + 2P_2\right)\left(r_3 + r_2\right)\sin\varphi\right]}{\left[\left(2P_4 + 27P_2\right)\left(r_2 + r_3\right)^2 + 3P_3\left(r_1 + r_3\right)^2\right]}. \quad (27)$$

Based on the expression (27) it can be concluded that for the equal rotation of the grinding rotors of the potato digger, it is necessary that:

$$M_4 = -\frac{\left(P_4 + 2P_2\right)\left(r_3 + r_2\right)\sin\varphi}{2}.$$

This condition stipulates the number and size of the soil grousers on the driving gear wheel rim.

It is evident that the bigger the number of the soil grouser bars on the rim is, the more equal the rotational movement is. But it should be noted that the increase in the number of grouser bars assumes decrease in the inter-bar distance which will definitely bring to the abrupt decline in soil mass resistance and skidding of the carrying gear wheel (Tarverdyan, 2014, Sineokov, Panov, 1977, Golushkevich, 1948). Considering this condition and based on the field and laboratory experiment results it has been established that the circumferential distance between adjacent grouser bars should be bigger than 0.17m (the radius of carrying wheel rim is 0.525 m, grouser length – 0.1 m). In this case the number of the grouser bars is 20. So, that radii of the adjacent grouser bars will form central angle of 180, 20 grouser bars will be installed on the rim, 3 grouser bars will be in touch with the soil simultaneously and the area creating clinging resistance during the movement will actually stay stable.

It should be noted that in field and laboratory experiment we used carrying wheel, the radius of which is 0.25 m, the grouser length – 0.1 m. In that case the number of grouser bars is 10. It is evident that the optimum number (n) of the bars is conditioned by the radius of the carrying wheel R_W. In practical calculations, the approximate number of the grouser bars can be defined in the following correlation:

$$n = 40R_W,$$

where R_W is the wheel radius in meters. The number (n) defined in this way should be rounded up to the closest even number.

Conclusion

The best option of drive gear mechanism of rotary soil crusher in the soil digger is 4-ring planetary mechanism with parasitic satellites, the transmission number of which from the carrying wheel to the working rotors should ensure the throwing V velocity of soil clods at 9-10 m/s (in the discussed case the chosen i transmission number is 8.2).

In the result of kinematic and dynamic analysis of the mechanism theoretical expressions have been derived which give us the opportunity to define the resistance momentum of M_3 of the working rotors and the necessary momentum of M_4 transferred from the clinging tooth gear to its balancing carrier.

The conditions of the equability of the rotation for the working rotors have been identified according to which the angular acceleration of the carrier will be zero or a value very close to it. Based on that condition the necessary weights of the tooth gear and the carrier have been determined.

The necessary number of the grouser bars on the wheel rim was established to exclude the equal rotation and skidding of the clinging drive gear wheel.

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