Rotary working body of the tillage machine

Andrey Ivanov¹, Maxim Puzyrev¹

¹Northern Trans-Ural State Agricultural University, 7, Respubliki str., 625003, Tyumen, Russia
E-mail: ivanovas@gausz.ru

Abstract. In the process of designing tillage and other agricultural machines, the main problem is to ensure the quality and optimal energy consumption of technological operations. For existing rotary and milling tillage machines, the main indicators of the quality of work and the energy intensity of the process depend on the kinematic, structural and technological parameters, the ratio of the angular rotational speeds of the disks, translational speed, diameter of the working body, number of cutting elements, depth of cut, ridge height, etc. The ridging of the bottom of the furrow is one of the important agrotechnical indicators of the operation of rotary tillage machines. The height of the ridges should be no more than 20% of the tillage depth. This requirement imposes certain restrictions on the choice of parameters and operating modes of rotary tillage machines. The purpose of the study is to establish the laws of the influence of kinematic and structural parameters of a double-disc rotary working body on the quality and energy intensity of soil cultivation. The object of the study is the process of cutting soil by an active-passive rotary working body. The influence of setting and design parameters on the energy intensity of the working process of a double-disc rotary working body is determined. Power characteristics of the working body, leading to the destruction of the ridge of the bottom of the furrow are defined. Range of adaptive high-speed regulation of the working process of a double-disk rotary working body is established. The energy parameters of the working process of the combined double-disc working body are assessed. It was found that the specific energy consumption of rotary tillage by a double-disc working body is reduced to 20-30% compared to milling. Evenness of the furrow bottom up to 80% is ensured.

1. Introduction

Ensuring optimal energy intensity and the quality of technological operations is one of the main problems in the design of agricultural tillage machines. Improving the design of such machines, as a rule, is aimed at increasing the reliability and failure-free operation, the resource of the machines themselves and their working bodies, at saving resources during the tillage and increasing productivity.

The design, kinematic and technological parameters of the operation of rotary agricultural machines significantly affect the energy intensity and quality of the tillage process. These parameters include the diameter of the machine’s working body, the angular speed and frequency of rotation of the disk organs, the number of cutting elements, the translational speed of the working body, the height of the formed ridge and the depth of cut [1-9]. One of the important agrotechnical parameters of the rotary tillage machine is the ridging of the furrow bottom. At the same time, the height of the formed ridges should be less than 20% of the depth of the cultivated soil. In this regard, there are a number of restrictions that determine the choice of parameters and operating mode of rotary machines.
The purpose of the study is to establish the laws of the influence of kinematic and structural parameters of a double-disc rotary working body on the quality and energy intensity of soil cultivation.

The object of study is the process of cutting soil by an active-passive rotary working body.

Research objectives:
1. To determine the impact of setting and design parameters on the energy intensity of the working process of a double-disc rotary working body.
2. To determine the power characteristics of the working body, leading to the destruction of the ridge of the furrow bottom.
3. To assess the energy parameters of the working process of the combined double-disc working body.

2. Materials and Methods

General research methodology:
1. Determination of the profiles and sizes of the ridges of the furrow bottom;
2. Definition of modes that ensure a flat bottom of the furrow;
3. Determination of the synchronization potential of the active and passive disk.
4. Determination of the kinematic and structural parameters of the double-disc working body, providing optimal energy intensity of the process.

The studies were conducted in the soil channel of the Chelyabinsk State Agroengineering University (Russia). The hardness of the soil was determined at 3 points with 3-fold repetition, using a hardness tester to a depth of 0.00...0.10 and 0.10...0.20 m, according to Russian State Standard GOST 29269-91 [10]. Number of measurements - 20 in different parts of the box with soil.

The determination of soil density was carried out as the ratio of the mass m of absolutely dry soil to the volume of the test sample, which was taken without disturbing its natural composition.

The determination of soil moisture was determined in accordance with Russian State Standard GOST 28268-89 [11] during the period of laboratory studies on the horizons at 5 points of the channel and box with 3-fold repetition for each horizon.

Since the main limitation in studies of the working process of tillage implements is the seasonality of the experiments, in order to ensure approximation to real conditions in the autumn-winter period, soil samples were excavated by preliminary thawing to the freezing depth (to the transition of the soil from frozen to plastic state) without violating its natural composition.

The soil was placed in wooden boxes with overall dimensions of 200x200x1500 mm to a depth of 20-22 cm. Boxes with soil samples were installed in the soil channel at the level of the day surface.

The experiments were carried out on the assembly, which includes the working body, the power unit, the regulating and control part (Figure 1). As the power unit, electric motors of the AIR-80 series of alternating current with a power of (0.75; 1.5 kW) with a rotation frequency (920, 1500, 3000 rpm) were used. The working body consisted of two disks with cutting elements (Figure 2) [12]. One is rigidly bolted to the shaft, the second rotates freely in the bearing. The regulating and, at the same time, driving role is played by an electric motor with automatic control of the shaft rotation speed from 0 to 900 rpm, the E-MINI LP7 frequency converter (Figure 3).

During the study, the MD141614 digital measuring instrument of alternating current with the ability to transfer data to a personal computer was used as a control and measuring part.
Figure 1. Structural and principal diagram of the assembly. (CE) - control element; (RE) - regulating element; (PE) - working body, performing element; (SE) - sensing element; (OV) - output value interface; (IV) - electric motor, an element of an input value. 1, 2 - soil channel and the studied soil mass; 3 - double-disc working body; 4 - flat belt transmission; 5 - mechanism for controlling the depth and stability of the course; 6 - carrier; 7 - assembly frame; 8 - electric motor; 9 - worm gear; 10 - automatic control system; 11 – instruments and meters.
The digital meter included a voltmeter, ammeter, wattmeter (indicating type). The active power consumed from the network was measured. The rotational speed of the working bodies was measured with a DT-2234A phototach. The tachometer was calibrated with a T410-R clock type tachometer with an accuracy class of 1.0. As cutting elements, straight knife blades of various configurations were used. The values of the angles were taken based on the minimum allowable angles $\gamma_1=20^\circ; \gamma_2=22^\circ; \alpha_1=25^\circ; \alpha_2=45^\circ; \alpha_3=30^\circ$. The material of knife blades is Steel 45. Flat disks of the SZ-3.6 grain seeder are selected as the base of the working body. To prevent clogging of the gap between the disks by the soil, we use a notched flat disk.
3. Results and Discussion

In the works of Matyashin Yu.I., Grinchuk I.M., Egorova G.M., special cases of the general equations of motion of the working bodies of rotary tillage machines are considered [13]. To describe the trajectory of the active disk, we take the following parameters: drive from the tractor power take-off shaft; stright drum, the axis of which is located frontally and horizontally:

$$
\begin{align*}
  x &= R(\alpha / \lambda \mp \sin \alpha); \\
  y &= R(1 - \cos \alpha),
\end{align*}
$$

where

- $\alpha = \omega \cdot t$ – the angle of rotation of the working body from the initial position;
- $\omega$ – the angular speed of the drum;
- $t$ – the time of rotation of the working body at an angle $\alpha$;
- $R$ – the radius of the drum.

For passive disk:

$$
\begin{align*}
  x &= R(\alpha - \sin \alpha); \\
  y &= R(1 - \cos \alpha),
\end{align*}
$$

Let us consider the general effect of elementary resistance forces on the decrease in the frequency of rotation of a passive disk. According to the given equations, we built the trajectory of the end points of the knife blades with the following parameters: kinematic parameter $\lambda=3$; the radii at the ends of the knife blades $d_1=d_2=0.2$ m.

Figure 4 shows that three turns of the second active disk are made for one revolution of the passive disk. Such a difference is observed due to the given value of the kinematic parameter $\lambda=3$. In practice, this difference between the position of the knife blades differs from the theoretical one, since the rotation of the passive disk is caused by the reaction of the soil to the knife blade and part of the disk to the tillage depth. Moreover, the soil reaction to the passive disk depends on controlled, but not regulated and random factors, which are difficult to establish in the automatic process control mode and in some situations, it is not possible at all.

![Figure 4. The trajectory of the movement of the end points of the knife blades of the double-disc working body: 1 - passive disk; 2 - active disk.](image-url)
When milling with the drum along the direction of the machine for the conditions $\gamma=0$, $\beta=0$, the height of the ridge can be calculated by the formulas:

$$\frac{\pi}{m} = \lambda \sin \alpha_2 \mp \alpha_2$$  \hspace{1cm} (3)

$$h_2 / R = (1 - \cos \alpha_2)$$  \hspace{1cm} (4)

In the case when for the knife blade of the passive disk of the double-disc working body, the end point $B$ is located in the interval between the knife blades of the active disk $S_1$ under the condition $h_2 \to 0$ (see Figure 5), the following dependence will take place:

$$B \in [x_1; x_1 + S_1]$$  \hspace{1cm} (5)

The frequency of finding point $B$ in the interval $S_1$ depends on the supply of $S_2$. Since the kinematic parameter $\lambda_2 < 1$, this means that the trajectory of the end points of the knife blade of the passive disk is different from the trajectory of the active disk (Figure 5).

**Figure 5.** Range of ridge formation by the active working disk.

Based on the fact that the rotation frequency of the passive disk depends on the soil reaction, therefore, the distribution of its end points in the formed furrow bottom can be compared with the Markov processes of reproduction and death in continuous time. Moreover, if the end point of any knife blade does not fall into the supply interval $S_1$, the influence of the cutting force on the ridge is not significant or equal to zero [14]. Therefore, the optimal distribution of end points will be a pure death process, while the intensity of the flow will lead to the destruction of the furrow ridge. Let us analyze the influence of the design parameters of the developed working body on the distribution of end points with a knife blade ratio of 3:8 (Figure 6) [14].
Figure 6. The trajectories of the end points of the knife blades of the double-disc working body when $z_1=3$, $z_2=8$.

The distribution of the end points of the knife blades of a double-disc working body in the supply range of the active disk is uneven and varies from 1 to 3 hits. Since during the construction of the trajectories, the influence of the angular velocities of the working body was not taken into account, therefore, the degree of influence and uniform distribution of the knife blades of the passive working disk on the ridge of the furrow will be possible with the following ratios of knife blades: 4:8; 8:8; 3:6; 7:7 [14].

Table 1. Milling power when varying structural and kinematic parameters.

| No. | Knife angle of attack | Translational speed, $v_m$, m/s | Kinematic indicator, $\lambda$ | Milling power, kW | Soil moisture 15-18% | Density 1,545 kg/m$^3$ | Depth of tillage 10-12 cm |
|-----|-----------------------|---------------------------------|-------------------------------|-------------------|-------------------|-------------------|-------------------|
|     |                       |                                 |                               | $X_1$             | $X_2$             | $X_3$             | $X_{av}$         | $\sigma^2$       |
| 1   | 0                     | 1                               | 4.1                           | 0.45              | 0.5              | 0.5              | 0.5              | 0.000            |
| 2   | 15°                   | 1                               | 2.7                           | 0.49              | 0.34             | 0.5              | 0.44             | 0.299            |
| 3   | 15°                   | 1.5                             | 2.7                           | 0.3              | 0.54             | 0.7              | 0.51             | 0.202            |
| 4   | 20°                   | 1.5                             | 4.1                           | 0.15             | 0.2              | 0.35             | 0.23             | 0.156            |
| 5   | 20°                   | 2                               | 2.7                           | 0.3              | 0.35             | 0.25             | 0.38             | 0.346            |

In order to establish the influence of the coefficient of destruction of the furrow ridge on the change in the kinematic index of the active working disk, a number of experiments were carried out. The simulation results are presented in table 2.

Table 2. Planning matrix and simulation result.

| Experiment number | $X_1$, pcs | $X_2$ | Experiment Y | Basic $\lambda_{min}$ |
|-------------------|------------|-------|--------------|-----------------------|
| 1                 | 1          | 0.3   | 0.972        | 3.24                  |
| 2                 | 0.428      | 0.5   | 0.726        | 2.42                  |
| 3                 | 0.714      | 0.7   | 1.26         | 1.8                   |
| 4                 | 1          | 0.5   | 1.62         | 3.24                  |
| 5                 | 0.428      | 0.7   | 1.69         | 2.42                  |
| 6                 | 0.714      | 0.3   | 1.54         | 1.8                   |
| 7                 | 1          | 0.7   | 2.27         | 3.24                  |
| 8                 | 0.428      | 0.3   | 0.726        | 2.42                  |
| 9                 | 0.714      | 0.5   | 0.9          | 1.8                   |
The results of the studies showed that the decrease in the kinematic index in experiments No. 1, No. 2, No. 9 (see table 2) was less than the permissible value for the active working bodies of rotary machines. From the position of the minimum specific energy consumption of the soil cultivation process, the optimal effect of the passive disk on the ridging of the bottom of the furrow was noted in experiments No. 3, No. 4, No. 5, while the ratio of knife blades was 5:7; 3:3; 3:7. The optimal value of the coefficient of soil destruction lies in the interval $k_d=0.5...0.7$.

The result of determining the difference in energy costs during milling and rotary tillage by a double-disc working body is presented in Figure 7.

![Comparative characteristics of energy consumption](image)

**Figure 7.** Graph of comparing energy costs with a double-disc working body and an active disk: row 1 - milling; row 2 - rotary tillage by a double-disc working body.

4. **Conclusions**

1. Experimental studies have confirmed the adequacy of the mathematical model to the real process, as evidenced by the convergence of comparisons.
2. The modes and design parameters that ensure the reduction of rotary tillage by a double-disc working body to 20-30% in comparison with milling are defined.
3. The kinematic and structural parameters are determined at which the furrow bottom is evened up to 80%.
4. The reduction in the specific energy consumption of milling compared to serial machines by 11-17% is substantiated.

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