THE ACADEMY OF MANAGEMENT AND ADMINISTRATION IN OPOLE

METHODOLOGY OF ENGINEERING MANAGEMENT OF AGROTRONICS OF GRAIN PRODUCTION BY AGRICULTURAL ENTERPRISES

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Ivan Rogovskii, Liudmyla Titova, Mikola Ohiienko, Olga Snezhko,
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PREFACE

The basis of the monograph on the problem of ensuring sustainable human development, in particular environmental management that involves increasing the efficient use of existing resources and modern technologies of grain production. From the solution of this problem depends the food security of the country. Gross harvest of the grain portion of the crop must be sufficient to provide food grains at the level of international standards (up to 1 ton per population of Ukraine), as well as seed, feed grain and raw materials for other sectors of the economy.

The monograph aims to increase the gross grain harvest part of the crop of modern production technologies.

The main scientific results obtained by the authors in preliminary studies include the scientific and technical basis for structuring the existing 157 models combine the park, identifying multiple diferenciales structures technology agricultural companies of the grain production on the area of 58 hectares in the 21120 ha depending on krupnotovarnogo production with the possibility of finding internal mechanisms of development and pursuing their own interests for their development. In previous studies it was found that modern agrominera provision does not contain a clear recommendation of of its technical characteristics. The authors propose a method of determining the optimal cultivated area under crops for associativity, additively, grouped by homogeneous indicators in the totality of their deterministic characteristics. In scientific asset research team, also is the allocation of the group of ten dominating factors: agrolandscapes characteristics of the agricultural companies of the grain production, soil and climatic conditions, resource potential, technological support, technical support, system nasonville, adaptability of crop rotation, organization of work, staffing, social conditions. The coefficient of concordance we establish the existence of some rational and interrelated parameter values of each of the ten factors that can influence directly the farming enterprise. Previous scientific studies indicate the relevance of the solution of scientific problems of optimization of Park of combine harvesters for the permissible grain losses for agricultural enterprises with different levels of marketability based on the modeling of rates of harvest operations and engineering management of grain production in Ukraine.

Full matrix studies on the importance of various factors for increasing grain production in agricultural enterprises, the intensification of engineering management of domestic and foreign scientists was carried out.However, it was indirectly established that technological (1.52 rank), technical (2.04 rank) and organizational (3.14) factors are the main factors with a total concordance coefficient of 0.841. The authors confirmed the existence of a number of agronomic, technological, technical and social problems: ensuring maximum agro-landscape adaptability of land use, technologicalization of production and adaptability of the fleet structure to specific conditions of grain harvesting, ensuring minimum grain losses in all harvesting operations, application of crop rotations varieties adapted to mechanized harvesting, optimization of the system "field - combine - transport - grain flow" in a single production process with a given rate of harvesting within 2-4 thousand tons of grain per day, strict adherence to technological discipline in all operations for grain production crops, harmony of technical support of agricultural trials with observance of the set pace of their carrying out, introduction of system control of quality of works and their performance in the set volumes.

The monograph is based on the results of research on the topic 110/9-pr-2020 "Substantiation of methods to increase grain production in agricultural enterprises by intensification of engineering management" (№ state registration 0120U102086).

CHAPTER 1. ENGINEERING MANAGEMENT OF TECHNOLOGICAL EFFECTIVENESS OF FORMATION OF PLANTING FURROW BY WORKING BODY OF PASSIVE TYPE OF ORCHARD PLANTING MACHINE

Introduction

The technological effectiveness of process of orchard planting seedlings provides for the following stages [1]. Seedlings for planting are placed on the site on both sides of the machine [2]. There are 2-3 operators on special seats [3], depending on the speed of the tractor [4], which perform the following functions. One operator directly lowers the seedlings into the furrow [5], the other two alternately feed the first seedlings directly into the hands [6], taking (removing) the seedlings from the site of the machine. In the course of movement of the unit after installation of a sapling in a furrow by means of two dumps (wrappers) there is a backfill of a furrow [7]. The rear wheels compact the soil in the rhizome area [8]. The planting area can be pre-marked with lines with small furrows for even placement of seedlings [9].

The mechanization of production processes in horticulture is an innovative reserve for increasing labour productivity and profitability of fruit production [10]. The specialized machines created in recent years allow to increase the level of mechanization of technological operations on cultivation of fruit crops [11]. A review of literature sources [12] shows that the existing working bodies of orchard planting machine of active and passive type do not fully meet the quality criteria, namely: to be easy to manufacture, reliable, undemanding to the soil and energy efficient [13]. Thus, wedge-shaped passive organs, pushing the soil during the technological operation in both directions, form a landing gap [14]. This method requires significant energy costs [15], because, firstly, its application requires preliminary deep ploughing [16], which leads to significant additional costs, and secondly, compacting, the soil has a significant resistance to the motor unit [17]. Active milling-type bodies are no less energy-intensive, moreover difficult to manufacture [18], and therefore are characterized by a high initial cost and also require significant operating costs [19].

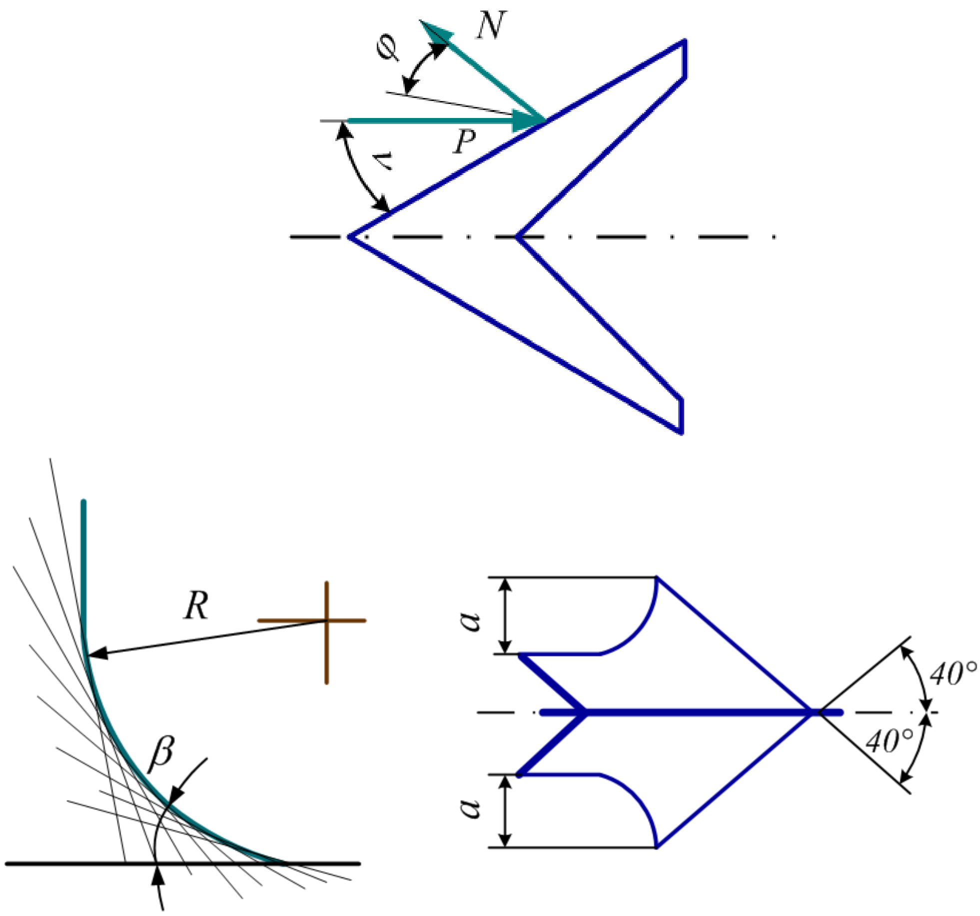
The aim of research is to reduce energy consumption and increase the reliability of the orchard planting machine in different conditions of its operation by developing a passive working body of advanced design, designed to form a planting furrow.

Materials and methods

According to the working hypothesis, all these requirements can be met by a passive type body, which, unlike a wedge-shaped opener, does not spread the soil, and pruning the layer, raises it to form a planting furrow, and after placing the seedling in it moving or rotating.

According to agricultural requirements, the depth of the planting furrow should be 0.20–0.35 m, and the width should ensure the free location of the root system of the seedling. To determine the optimal value of the last of these parameters, the process of entering and placing the root system of biennial apple seedlings grown on vegetative rootstocks in the interval between the opener extensions, the distance between which varied from 0.10 to 0.20 m step 0.02 m. The number of root hooks at their edges was recorded.

In accordance with the hypothesis of the advantages of the working body in terms of energy savings, a model and a prototype of the working surface of the opener were developed and manufactured. It consists of two cylindrical surfaces located at an angle of 40°. This arrangement provides the required bevel angle of the blade γ (figure 1.1a); the development was based on the mandatory condition of its self-cleaning from the fibrous stems of plants. The condition for sliding the plant stem along the blade of drill coulter point was $\gamma<90°-\varphi$.



a b

Figure 1.1. Scheme of interaction of plant stem with blade of drill coulter point.

It is known that the angle of friction of the plant stem on the blade of the tillage body $\varphi \cong 45°$, therefore: $\gamma < 90° – 45° = 45°$. It is known that the adhesion of soil to the working surface of the blade prevents weeds from sliding on it, so the bevel angle γ was reduced to 40°. The working surface of drill coulter point was built in accordance with the set goal – to minimize energy consumption during the operation, choosing the most optimal angle of crushing the soil.

Studies have shown that the most optimal variant of the working surface is a cylinder with a certain radius of curvature of the surface *R* and the initial angle of crumb β (figure 1.1b).

To create favourable conditions for plant survival and development of their root system, as well as to prune the soil layer to ensure the effect of its opening and the formation of the landing gap, the blades of the cylinders are elongated and they act as wings for the size of the opener by a (figure 1.1b).

The main indicator of the tillage tool is its traction resistance. It is known that this indicator is influenced by many factors, among them: the geometric parameters of the working surfaces of the tool, the physical and mechanical properties of the soil, the speed of interaction with it, the depth and width of cultivation, etc. [3]. Given such a number of factors, their change over time and unpredictability, especially in relation to physical and mechanical properties, it is theoretically impossible to describe the process of interaction of all forces and resistance factors [2, 4]. With this in mind, we will try to determine the expected traction resistance of the opener R_x, without going into the details of this complex process.

Based on the similarity of interaction with the soil of the developed of drill coulter point opener and ploughshare, as well as taking into account the similarity of their working surfaces, we apply the well-known rational formula of Academician V. P. Goryachkin:

$$R_x = f \times P + k \times a \times b \times n + \varepsilon \times a \times b \times n \times V^2, \tag{1.1}$$

where $(f \times P)$, $(k \times a \times b \times n)$ and $(\varepsilon \times a \times b \times n \times V^2)$ – traction supports associated, respectively, with the drawing of the working body in the open furrow, the deformation of the soil and the provision of raised and deformed soil kinetic energy.

Results and discussion

In figure 1.2 illustrates the dependence of root hooks on their edges on the distance between them. Increasing the distance from 0.16 m does not significantly affect the number of such cases. In view of this, as well as taking into account the direct dependence of the growth of the resistance of the opener on increasing its width, as the optimal parameter of the distance between the extensions of the opener, and hence the width of the landing gap of 0.16 m.

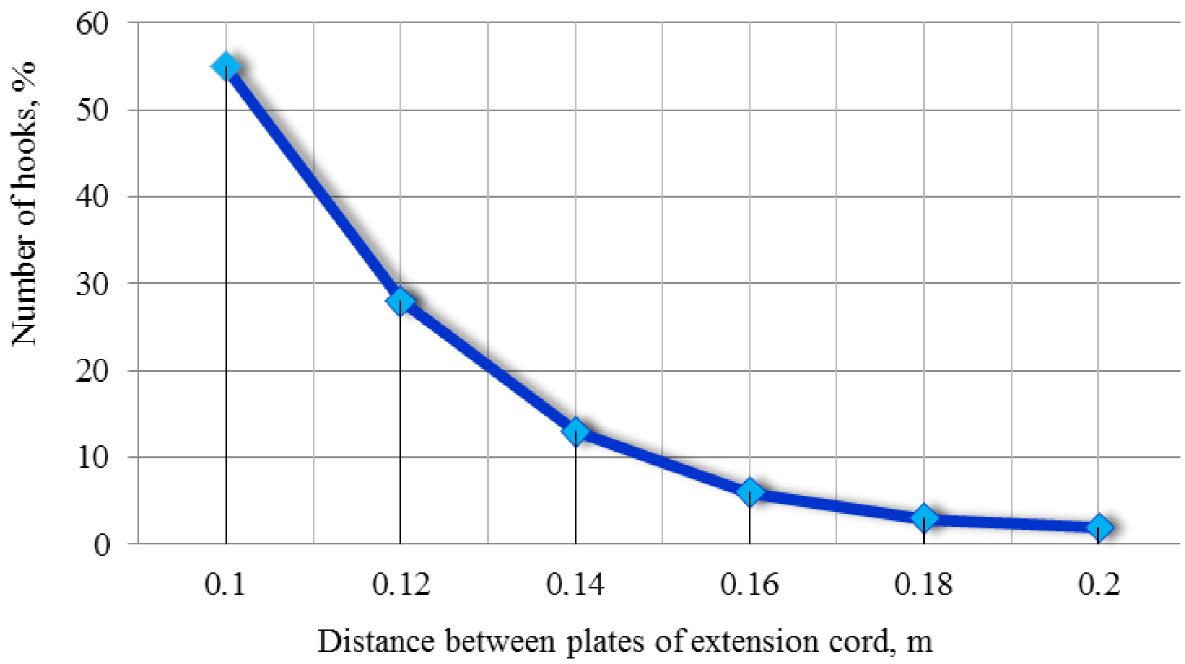


Figure 1.2. The number of hooking's of roots of two-year-old apple seedlings when trying to place their root system in area between the extensions of opener model.

It is necessary to consider some features of working bodies of orchard planting machine. Thus, in contrast to the plough body, its opener constantly works in the mode of laying the first furrow, in which the traction resistance increases by 2–3 times. In addition, the depth of its cutting is about 0.30 m, which in most cases exceeds the depth of tillage in previous years, and this leads to an increase in the expected resistance by 1.5–2.0 times. Taking into account these factors, it is necessary to increase the second component of the above formula by an additional factor μ. In our case, it was found that depending on the planting depth (0.15–0.35 m) and the condition and type of soil, this coefficient can take values from 1.0 to 6.0 (figure 1.3). As a result, the horizontal component of soil resistance R_x when planting to a depth of 0.30 m will be:

$$R_x = f \times P + \mu \times k \times a \times b \times n, \qquad (1.2)$$

where f – coefficient of friction of the body of drill coulter point with soil, f=0.5; P – gravity of the working body, P=8.0 kN, k – coefficient that characterizes the ability of the soil layer to resist deformation, k=35.5 kN/m^2; a, b – respectively the depth and width of the layer of soil to be cultivated, a=0.30, b=0.18 m; n – number of working bodies, n=1.

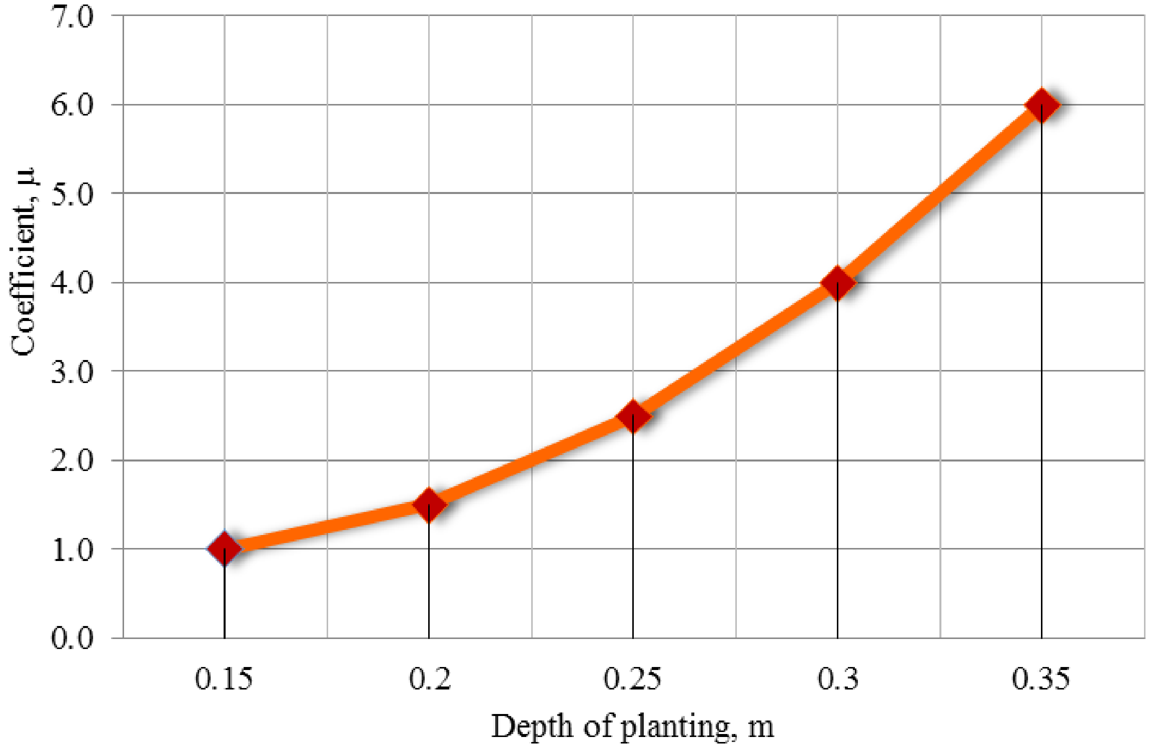


Figure 1.3. Dependence of additional average soil resistance coefficient μ on depth of planting.

Based on the results, the orchard planting machine can be aggregated with a tractor class 1.4, in contrast to the known orchard planting machine Spaperri MPS-1. Its aggregation is provided with class 3 tractors. The figure 4 illustrates the dependence of the resistance of the working body of the orchard planting machine on the depth of planting and soil type.

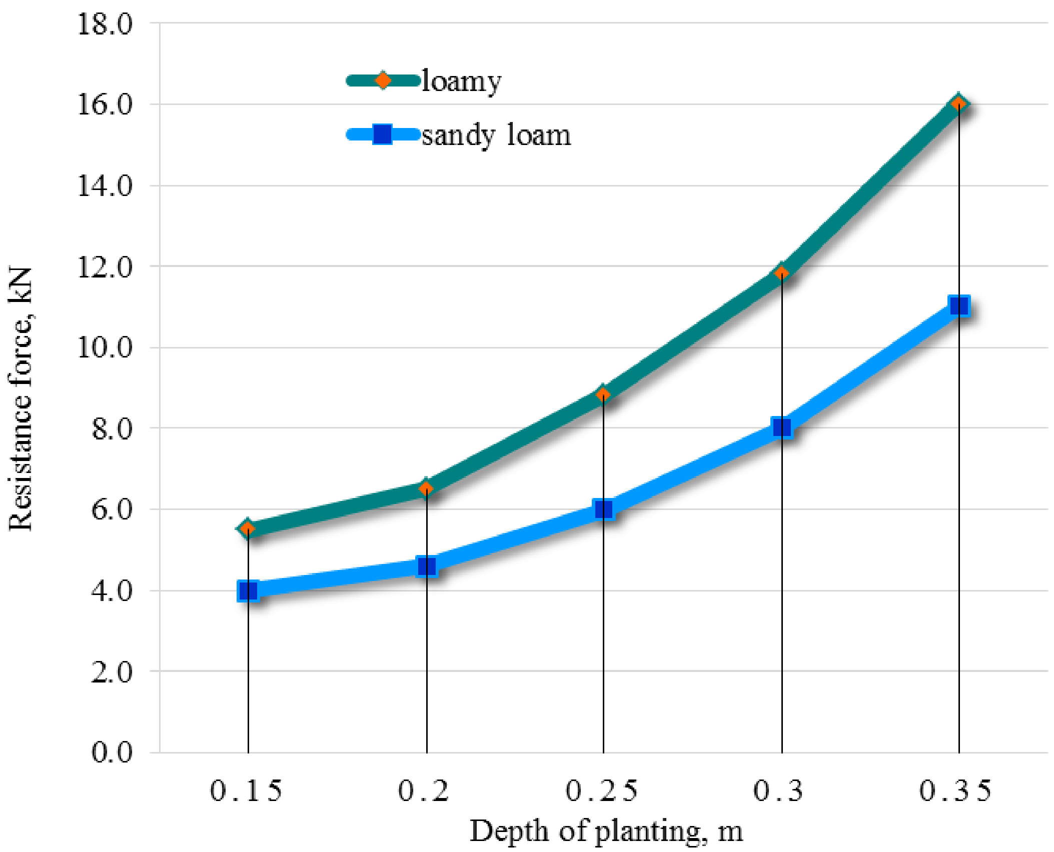


Figure 1.4. Dependence of the resistance force of the working body of the orchard planting machine on the depth of planting and soil type.

As a result, we get:

$$R_x = 0.5 \times 8.0 \text{ kN} + 4 \times 35.5 \text{ kN/m}^2 \times 0.3\text{m} \times 0.18\text{m} = 11.7 \text{ kN}. \tag{1.3}$$

Taking into account the calculations, a prototype of such a body was developed and manufactured (figure 1.5).

Figure 1.5. Prototype of the working body of the orchard planting machine during experimental research and testing.

Preliminary tests have shown its stable operation at all depths in the range of 0.15–0.35 m, and everywhere it is successfully aggregated with the tractor Belarus 82.1 (class 1.4), even on fallow lands after pre-surface treatment with disc tools.

The conducted researches confirmed the correctness of the working hypothesis accepted at the beginning of researches about possibility of essential reduction of the energy consumption necessary for performance of the technological operation connected with formation of a landing furrow by means of simple in manufacturing [20] and reliable in operation working body of passive type [21], spreads the soil [22], and pruning the layer [23], raises it to the formation of a planting furrow [24], and after placing the seedling in it lays it in the previous place without additional significant movement or rotation [25].

Conclusions to Chapter 1

Hooks of roots of two-year-old apple-tree saplings are absent at placement of their root system in an interval between extensions of model of an opener at width of a landing crack in the size of 0.16 m.

The optimum option for the working surface of blade of drill coulter point is a cylinder, which protrudes the wings beyond the dimensions of the opener, which provides a minimum horizontal component of soil resistance R_x during planting and is 11.7 kN when performing the operation to a depth of 0.30 m.

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CHAPTER 2. ENGINEERING MANAGEMENT OF GRAIN SEEDER-CULTIVATOR FOR SOIL-SPREADING SOWING

Introduction

One of the main indicators of sowing quality is the uniform distribution of plants by feeding area (*Zhai C., et al., 2019*). Until now, the most common method of sowing cereals is ordinary, in which plants occupy only about 30% of the field area (*Mircea C., et al., 2020*). Agrotechnical science has established that the necessary factors for the growth and development of cereals – light, water and nutrients can be rationally used only with a uniform distribution of seeds over the field area (*Vlăduţ D.I., et al., 2018*). The plants closest to the optimal feeding area are obtained by applying the soil-spreading method of sowing (*Jha A. & Kewat M., 2013*), which is performed by a seeder-cultivator. In addition to increasing yield capacity, this method allows you to combine pre-sowing tillage with sowing, which reduces the time of sowing, operating costs and causes less loss of soil moisture (*Rogovskii I., et al., 2020*).

The advantages of seeders-cultivators can be especially effective when it is used in farms, in most of which energy resources are represented by one or two tractors of traction class not more than 14 kN (*Ciuperca R., et al., 2020*). Pre-sowing tillage and sowing are performed by one tractor (*Voicea I., et al., 2020*). This causes a significant gap between the implementation of pre-sowing cultivation and sowing, which negatively affects the yield (*Belc N., et al., 2020; Rogovskii I.L., et al., 2020*).

Analyzing the results of research on mechanization of subsoil-spreading method of sowing, it should be noted: all studies confirm the high efficiency of subsoil-spreading method of sowing (*Farooq M, et al., 2011*), which is performed by seeders-cultivators (*Turan J., et al., 2015*); reasonable advantage of passive distributors in comparison with mechanical and pneumatic active (*Jin H., et al., 2014*); change of the direction of movement of seeds by passive distributors can be carried out in two ways – sliding on a curved surface or reflection (*Verma A. & Guru P., 2015*). The method of sliding is more studied, but it has a number of disadvantages (*Kocira S., et al., 2020*). It requires vertical feeding of seeds, which is not always possible (*Weckler P., 2019*); distributors operating on the principle of reflection are studied superficially (*Saitov V.E., 2014*); most studies have not taken into account the randomness of the physical and mechanical properties of seeds (*Abbaspour-Gilandeh Y., et al., 2018*).

Given the above, we can assume that a promising tendency is the development of openers for subsoil-spreading sowing with distributors operating on the principle of reflection (oblique impact) (*Daesescu I., et al., 2019*). To implement this direction, it is necessary to study the process in detail, taking into account the statistical characteristics of seed properties (*Rogovskii I.L., et al., 2020*).

The purpose of the study is to increase the efficiency of the process of subsoil spreading by improving the scheme and determining the rational parameters of the opener of the seeder-cultivator.

Materials and methods

The method of determining the statistical characteristics of the coefficient of air resistance K was, as follows: using the installation (figure 2.1), the time of fall of series of seeds t_i from a given height H was determined; according to the graphical dependences $K=f(t)$ for different values H, provided $x=H$, for each t_i, the corresponding values K_i were determined, which were processed statistically.

Statistical characteristics of the recovery factor K_b were determined by the flight range of the seed L_i after falling from a fixed height on an inclined reflector. According to graphic dependencies $K_b=f(L)$, a corresponding value L_i was determined for each value K_{bi}. The obtained values K_{bi} were processed statistically. The speed of seeds after climbing from the curved part of the seed line was determined by the flight range of the seed L_k after passing through the seed line of a certain radius.

The method of multivariate experiment is used to substantiate the optimal values of angles $\alpha_1(x_1)$ and $\gamma(x_2)$ and the height of the subblade area $h(x_3)$.

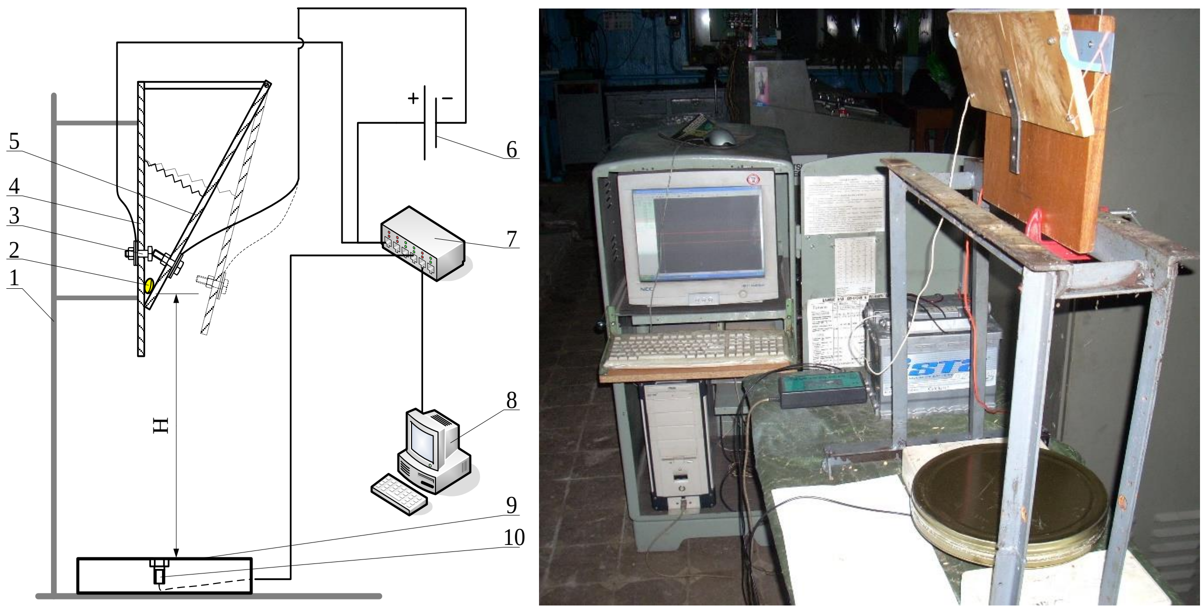


Fig. 2.1 The scheme of installation for definition of time of falling of seeds

1 – support; 2 – seeds; 3 – contact; 4, 5 – fixed and movable valve surface; 6 – current source; 7 – USB oscilloscope; 8 – PC; 9 – site; 10 – vibration sensor.

Table 2.1

The results of the implementation of the planning matrix of the experiment (Box-Behnken plan)

Codes			Codes and natural values			y (ν, [%])
x_1	x_2	x_3	x_1 (α_1, [deg.])	x_2 (γ,[deg.])	x_3 (h, [m])	
+1	+1	0	88	65	0.030	60
+1	-1	0	88	45	0.030	54
-1	+1	0	68	65	0.030	40
-1	-1	0	68	45	0.030	48
+1	0	+1	88	55	0.040	38
+1	0	-1	88	55	0.020	62
-1	0	+1	68	55	0.040	36
-1	0	-1	68	55	0.020	60
0	+1	+1	78	65	0.040	26
0	+1	-1	78	65	0.020	53

0	-1	+1	78	45	0.040	41
0	-1	-1	78	45	0.020	60
0	0	0	78	55	0.030	29

The results of the implementation of the planning matrix of the experiment are presented in table 1. According to the indicators of lateral scattering of seeds, due to the oblique impact, the angles τ between the central plane and the plane of the seed trajectory, flight range l and lateral deviation from the central plane c were taken (Fig.2). After the seeds fall from a fixed height H without initial velocity and reflection by the reflector *2*, the values l and c are measured. According to the obtained data, the statistical characteristics of the studied parameters are determined. Studies to determine the uniformity of seed distribution across the width of the seeding strip were performed on the installation, the scheme of which is shown in figure 2.2.

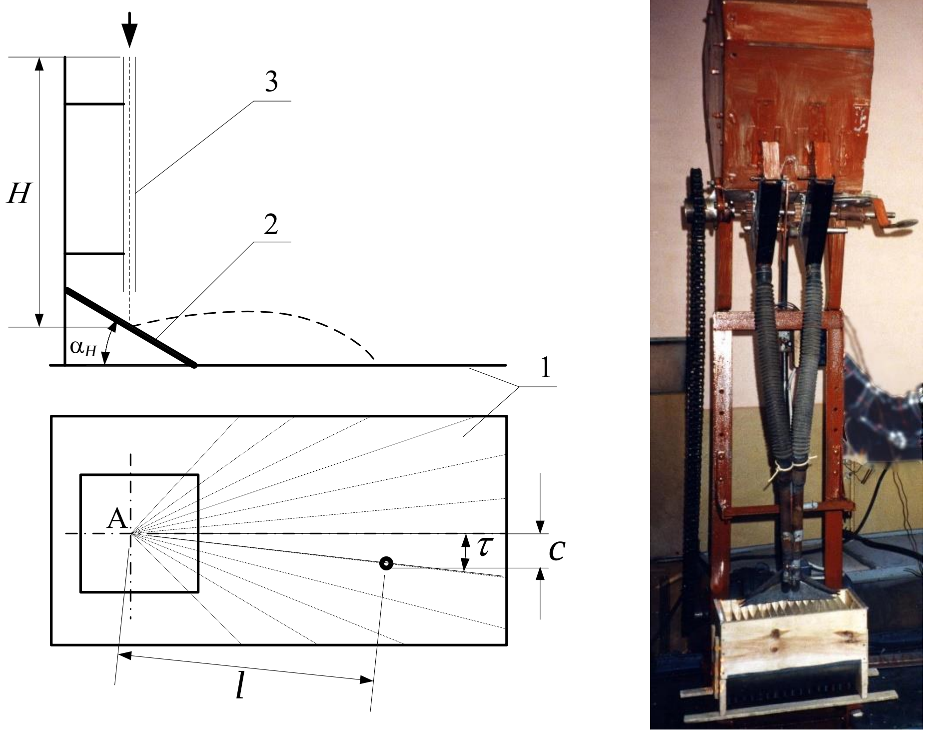
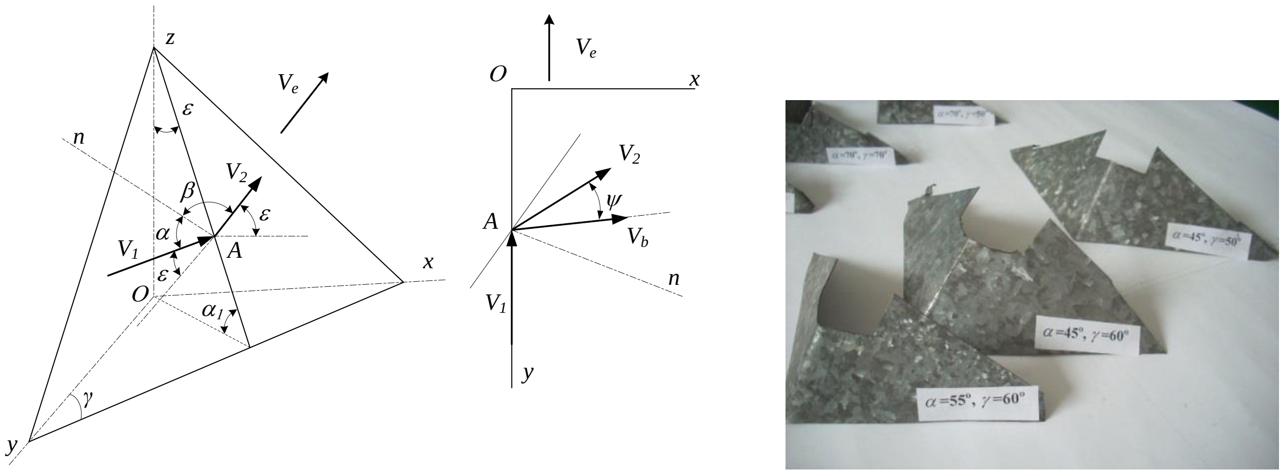


Fig. 2.2 Scheme of installation and the general look for definition of uniformity of distribution of seeds on width of strip of seeding

1 – platform; 2 – reflector-distributor; 3 – guide.

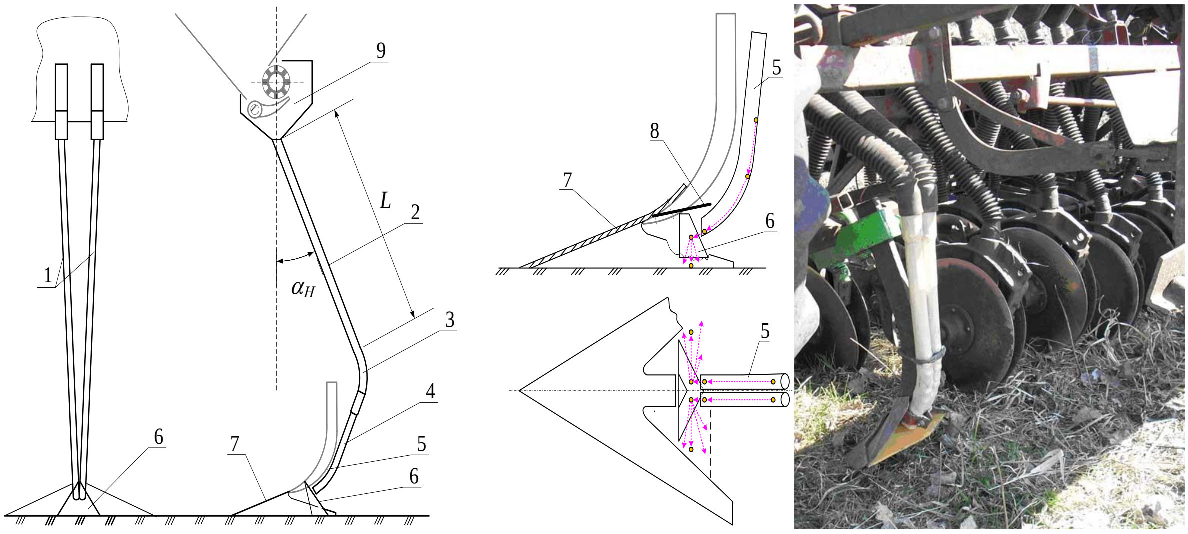
The grain of each cell was weighed. The average mass of seeds in the cells was taken as a number of random variables. After statistical processing, the coefficient of variation was obtained, which was taken as an estimate of uniformity (*Rogovskii I.L., et al., 2019*). The Box-Behnken plan matrix for three factors was used in the planning of the experiments (table 2.1). The main factors were the angles α_1 and γ (Fig. 2.3) and the height of the subblade area *h*. The main parts of the cultivator blade opener are (Fig. 2.4a): two seed ducts *1*, which have rectilinear inclined cylindrical sections *2* and *4*, and torus-like upper *3* and lower *5* sections; the reflector-distributor *6* and the cultivator blade *7* with the shield *8*. The reflector-distributor is a prism, the two working borders of which (right and left) are installed at certain angles to the horizon and the direction of movement. During the movement of the drill-cultivator by the sowing machine *9*, the seed drill is fed into the seed lines, from which it enters the reflector-distributor 6 at a certain angle (Fig. 2.4b). After reflectoring, the seed flies in the subblade area for some distance.



a b c

Fig. 2.3 Scheme for analysis of process of reflection and flight of seeds with feed in longitudinal vertical plane

Due to the different physical and mechanical properties of some seeds (coefficients of recovery and air resistance), the flight distances are different, which determines the distribution of grain along the bottom of the furrow. An important feature of the proposed opener is that in the right and left subblade area the seeds are fed by different seed ducts, which eliminates the divider of the seed flow.



a b c

Fig. 2.4 Cultivator blade of drill coulter

a – scheme; b – reflector-distributor in the subblade area; c – general appearance

Results

The main evaluation characteristic of seed movement through the seed duct and in the process of reflection is the coefficient of change of speed K_{zv}, which is defined as the ratio of speed after passing the operating device V_2 (or its section) to the input or potential (maximum possible) speed V_1. The process of movement of seeds by the seed duct, which has the form of an inclined cylinder, is considered as the movement of a material point on an inclined plane. The seed is affected by gravity $G=m\times g$, friction $F=f\times g\times \sin\alpha_H$ and air resistance $R=K\times m\times V$, where m – seed mass, kg; g – acceleration of free fall, m/s^2; f – friction coefficient; α_H – angle of inclination of the seed line; K – coefficient of air resistance. After solving the differential equation, the velocity V_2, without taking into account the air resistance ($K=0$), will be determined by the dependence $V_2=(2\times g\times L_H\times n)^{1/2}$, where L_H – length of seed duct; $n=\cos\alpha - f\times \sin\alpha_H$. It is advisable to take the potential speed V_1, speed of falling from a height $L_H \times cos\alpha_H$, which is determined by the dependence $V_1=(2\times g\times L_H\times cos\alpha_H)^{1/2}$.

With the following initial parameters, the coefficient of change of speed will be determined:

$$K_{zV_1} = V_2 \cdot V_1^{-1} = \sqrt{(\cos\alpha_H - f \cdot \sin\alpha_H)\cdot(\cos\alpha_H)^{-1}} = \sqrt{1 - f \cdot \mathrm{tg}\alpha_H} \,. \tag{2.1}$$

Taking into account the air resistance ($K \neq 0$), the solution of the differential equation will look like:

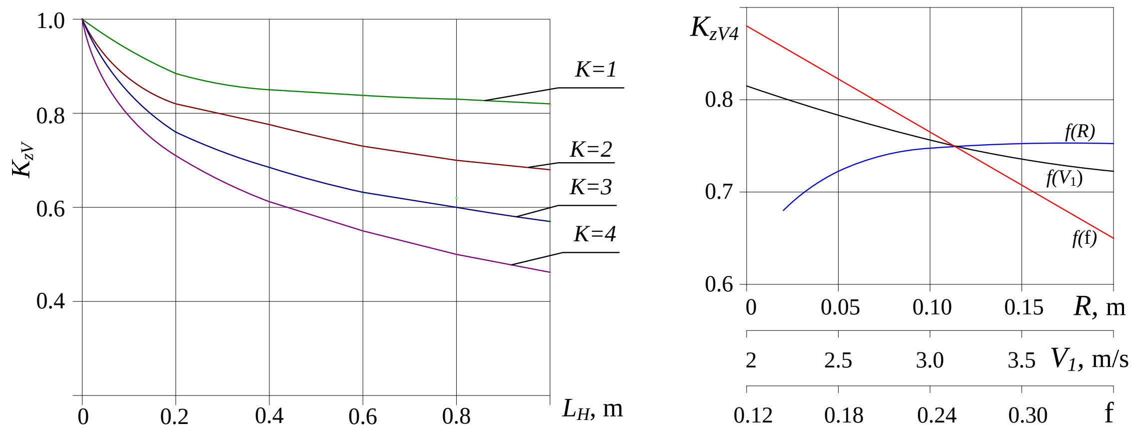
$$V_2(t) = n \cdot g \cdot K^{-1} \cdot \left(1 - e^{-K \cdot t}\right), \;\; x(t) = n \cdot g \cdot K^{-1} \cdot t - n \cdot g \cdot K^{-2} \cdot \left(1 - e^{-K \cdot t}\right) \tag{2.2}$$

In this case, the coefficient of change of speed is determined by the equation:

$$K_{zV_1} = V_2 \cdot V_1^{-1} = n \cdot g \cdot K^{-1} \cdot \left(1 - e^{-K \cdot t}\right) \cdot \left(2 \cdot g \cdot L_H \cdot \cos\alpha_H\right)^{-1/2} . \tag{2.3}$$

The dependences of the coefficient of change of velocity K_{zV1} during the movement of seeds in an inclined cylindrical seed line, taking into account the air resistance from the path L_H and K (at $\alpha_H=20°$; $f=0.25$) are shown in Fig. 2.5. From the figure we see that at $L_H=1.0$ m, $K=1.0$ sec^{-1} (close to real conditions) $K_{zV1}=0.81$, ie the speed decreases by 19%. Under such conditions $K=0$, $K_{zV1}=0.93$. That is, air resistance causes a decrease in speed by 14%. The process of movement of seeds on the upper torus-like section of the seed line is reduced to a single oblique impact. The coefficient of change of speed K_{zV2} is defined as the ratio $K_{zV2}=(K_b \times cos\alpha_2/cos\beta)$, where K_b – the coefficient of recovery, α_2 and β – the angles between the direction of speed before and after the impact and the normal, respectively. The process of seed movement along the lower section of the torus-shaped seed line is considered as the movement of a material particle along a cylindrical surface with a horizontal axis. In this case, the following forces act on the seed: the component of gravity $F_1=m\times g\times cos\alpha_4$; friction force due to gravity $F_2=f\times m\times g\times sin\alpha_4$; friction force due to centrifugal force $F_3=f\times m\times V^2/R$; air resistance force $F_4=K\times m\times V$, (where R – the radius of curvature). The process of motion along the arc of a cylinder with radius R is considered. The change in velocity is determined by the loss of kinetic energy E. In this case, the speed V_2 after passing the seed arc Δl will be determined by the equation:

$$V_2 = \sqrt{V_1^2 + 0.0348 \cdot \Delta\alpha\left(g \cdot \cos\alpha_4 \cdot R - f \cdot g \cdot \sin\alpha_4 \cdot R - f \cdot V_1^2 - K \cdot V_1 \cdot R\right)} . \tag{2.4}$$



a b

Fig. 2.5 Dependence of the coefficient of change of speed K_{zV}

a – on length of the seed line L_H and the coefficient of air resistance K; b – on the initial speed V_1, the radius of the arc R and the coefficient of friction f

According to calculations, the output speed V_2 of the previous arc Δl is taken as the input V_1 for the next. The influence of the main factors of the process on the coefficient of change of speed K_{zV4} can be seen from Fig. 2.6. It is worth noting that at a speed $V_1>1.5$ m/s, the radius of curvature R has little effect on the rate of change of speed K_{zV4}. This is explained by the fact that the main loss of kinetic energy (up to 90%) through the friction, is due to centrifugal force, the magnitude of which (energy) does not depend on the radius. The analysis of the process established that within the velocities of 2.0 m/s and 23.2 m/s the speed losses caused by the acting forces are distributed as follows: by the force of gravity of the seed – 7 - 13%; centrifugal – 71 - 82%; air resistance force 12 - 17%. From the lower torus-like section of the seed line, the seed enters the prismatic reflector-distributor. The process of reflection is reduced to oblique impact (Fig. 2.6). For this case, the coefficient of change of speed K_{zV} and speed after impact V_2 will be defined as:

$$K_{zV} = V_2/V_1 = \cos\alpha\sqrt{K_b^2 + \mathrm{tg}^2\alpha}\ ,\ V_2 = V_1 \cdot \cos\alpha \cdot \sqrt{K_b^2 + \mathrm{tg}^2\alpha} \tag{2.5}$$

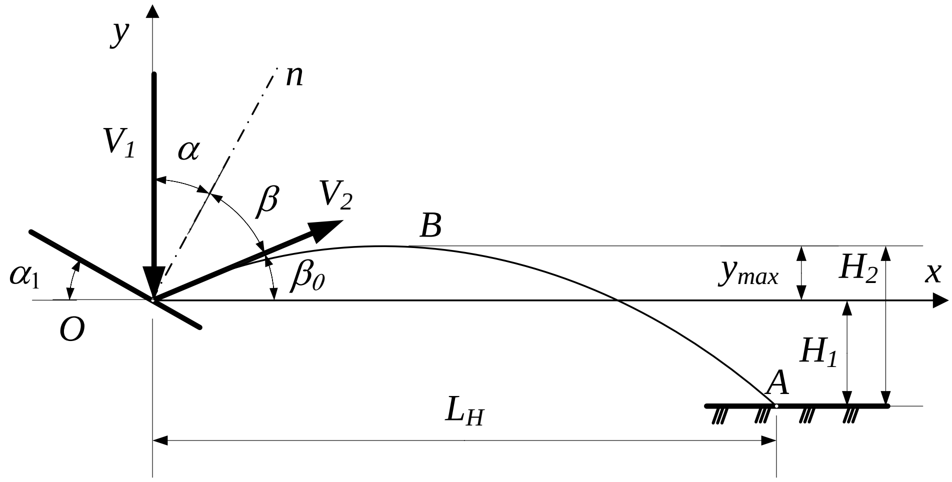


Fig. 2.6 The scheme of movement of seeds after reflection without taking into account air resistance

Using dependences (2.5) it is seen that the change of angle α is within 20 - 70° causes a change K_{zV} from 0.48 to 0.95 (at K_b=0.35). The effect K_b on K_{zV} decreases with increasing α. The speed after reflection V_2 and the angle β_0 determine the parameters of the flight trajectory of the seeds in the subblade area, which are the flight range L_H and the height of the trajectory above the bottom of the furrow H_2. These parameters are determined by the equations:

$$L_H = g^{-1} \cdot V_2^2 \cdot \sin\beta_0 \cdot \cos\beta_0 + V_2 \cdot \cos\beta_0 \cdot \sqrt{V_2^2 \cdot \sin^2\beta_0 + 2 \cdot g \cdot H_1} \cdot g^{-1}, \; H_2 = V_2^2 \cdot \sin^2\beta_0 \cdot (2 \cdot g)^{-1} + H_1 \quad (2.6)$$

where H_1 – the height of the reflection point above the bottom of the furrow.

Listed in Fig. 2.6. The scheme provides for the vertical supply of seeds to the reflector-distributor, which makes it impossible to place the seed line outside the cultivator blade. And such placement significantly increases the reliability of the cultivator blade opener. Therefore, as a promising, another scheme was investigated (Fig. 2.3). The working surface of the reflector x, y, z is inclined to the horizontal plane at an angle α_1, and the line of its intersection with the horizontal plane is an angle γ with an axis $0y$ that is parallel to direction of movement. The seeds are fed in a longitudinal vertical plane at an angle to the horizontal. Investigated the option when $\varepsilon = 90° - \alpha_1$. This condition guarantees the normal position of the reflection plane relative to the reflection plane. Flight range L_H and trajectory altitude H_2 are determined under the conditions: $\beta_0=\varepsilon$, $\alpha=90°-\gamma$. The width of the capture L_b is related to the range L_H by the ratio $L_b=L_H\times\cos\psi$, where ψ – the angle between the speed V_2 and the transverse direction). In this case, the angle β will be defined as $\beta=arctg(tg\gamma/K)$. Graphic dependence of the main characteristics of the process of reflection and movement of seeds in the subblade area on the angle γ (provided V_1=1.5 m/s; $\varepsilon=12°$; H_1=0.02 m) shown in Fig. 2.7.

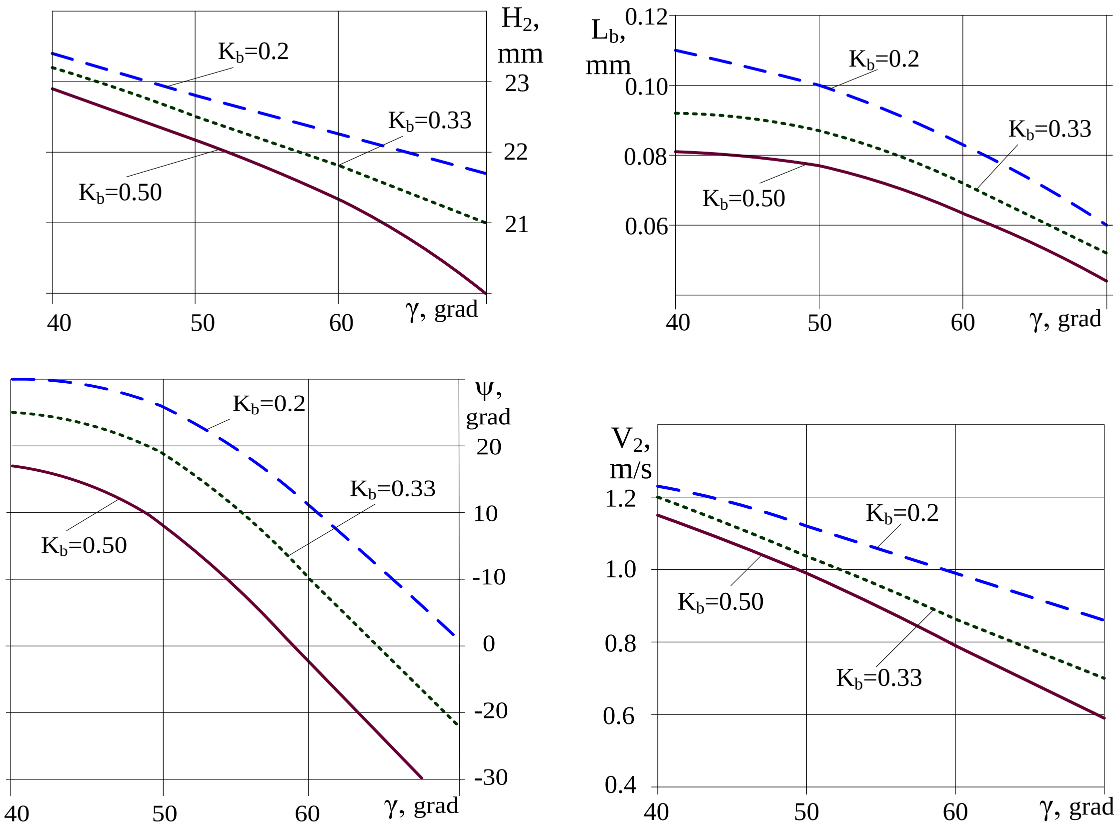


Fig. 2.7 Graphic dependence of the main characteristics of the process of reflection and movement of seeds in the subblade area on the angle γ

As the angle γ increases, all characteristics decrease, when increasing γ from 40 to 70° the average value of the recovery factor (K_b=0.33), the speed V_2 decreases from 1.04 to 0.761 m/s; angle ψ – from 35° to 0; seed flight distance in the transverse direction L_b – from 0.091 to 0.052 m; height of the flight trajectory of seeds H_2 – from 23.3 to 20.1 mm. According to the analysis of the results of previous experimental studies, the most uniform distribution of seeds occurs under the condition $L_b=0.5\times b$, where b – the working width of the blade wing. We take the working width of the blade wing $b=0.15$ m, so the desired flight range is 0.07 - 0.08 m. From the Fig.7 we see that values L_b correspond to the angle γ in the range of 55 - 60°. At values γ from Fig.7 we have: $\psi=14°$ - 0; H_2 – from 0.0233 to 0.0216 m. Therefore, according to the results of theoretical analysis at V_1=1.5 m/s; $\varepsilon=12°$; H_1=0.02 m the optimal value of the angle γ is in the range of 55 - 60°. The movement of the soil under the action of the cultivator blade opener was studied in order to determine the length L_2 and height z_3 of the subblade area (Fig. 2.8). Considering the wing of the cultivator blade as a wedge and taking into account the scientific positions on the laws of interaction of the triangular wedge with the soil, the dependences between the speed of the operating device (wedge) V_b and the relative speed of seam of soil V_r are used. To simplify, we consider the process by which it is conventionally assumed that the cultivator blade is stationary and the soil moves with speed V_r. Then the basic L_2 and z_3 are determined by equations (2.6), respectively, with $V_2=V_r$, $\beta_0=\varphi$, $H_2=z_3$, $H_1=z_1$. So:

$$L_2 = g^{-1} \cdot V_r^2 \cdot \sin\varphi \cdot \cos\varphi + V_r \cdot \cos\varphi \cdot \sqrt{V_r^2 \cdot \sin^2\varphi + 2 \cdot g \cdot z_1} \cdot g^{-1}, \quad z_3 = V_2^2 \cdot \sin^2\varphi \cdot (2 \cdot g)^{-1} + z_1 \qquad (2.7)$$

where $z_1=l_c \times \sin\beta$.

According to the results of calculations for the cultivator blade with a width of 0.33 m (β=28°; γ=32.5°; l_1=0.054 m; l_2=0.031 m; l_c=0.042 m; φ=16°) for V_e within 2 - 2.5 m/s (7.2 - 9 km/h) L_2 from 0.17 to 0.26 m, and z_3 from 0.031 to 0.036 m. The value z_3 is slightly larger than the theoretical height of the trajectory H_2, which is 0.022 - 0.023 m (H_2 Fig. 2.7). Statistical characteristics of the coefficient of air resistance K are given in table 2.2. Clear patterns of influence of height of falling on K are not revealed. As the value K for each crop varies considerably, the speed of the seed before hitting the distributor also changes, which contributes to the quality of seed distribution along the bottom of the furrow. Statistical characteristics of the recovery factor K_b were determined for wheat seeds, barley and peas. The experiments were performed at height of fall of 0.5 m and angles of inclination of the reflective plane to the horizon 20° and 30°. The obtained values of statistical characteristics are given in table 2.3.

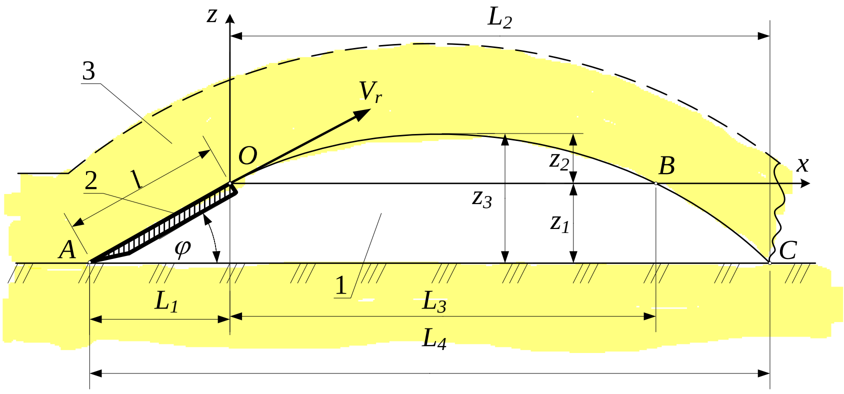


Fig. 2.8 The main parameters of the subblade area

1 – subblade area; 2 – wing of blade; 3 – seam of soil

Table 2.2

Statistical characteristics of air resistance coefficient K

Culture	Value of characteristics			
	K_{min}	K_{max}	m_K	v, %
Wheat	0.307	3.59	1.09	45.3
Barley	0.460	2.30	1.24	27.5
Peas	0.425	2.64	1.03	55.6

Table 2.3

Statistical characteristics of the recovery rate K_b

Characteristic	Value of characteristics at an angle of inclination α_b					
	20°			30°		
	wheat	barley	peas	wheat	barley	peas
The minimum value	0.02	0.02	0.02	0.10	0.10	0.10
The maximum value	0.55	0.55	0.51	0.64	0.66	0.64
The arithmetic mean	0.33	0.31	0.26	0.41	0.38	0.34
Coefficient of variation, %	36.4	42.1	34.9	32.0	38.0	34.8

Analyzing the data in table 2.3 it should be noted: the recovery factor is a random variable and varies for different cereals within significant limits (0.1 - 0.66). The average value K_b varies from 0.26 to 0.41, and the coefficient of variation from 32 to 42%. Fluctuations in the value cause variation in the flight range of the seeds, which contributes to better distribution of seeds in the transverse direction. The coefficients of change of speed during the movement of seeds along the curved lower part of the seed line are determined in order to establish the degree of reliability of the regularities of the influence of the main factors of the process, obtained by theoretical calculations. The research results are given in table 2.4: V_K – input speed taking into account air resistance (K=1.0), m/s; V_{2T} – theoretical input speed, m/s; V_{2E} – experimental input speed, m/s.

Table 2.4

Comparison of theoretical K_{zVT} and experimental K_{zVE} values of velocity coefficients

H, m	V_K, m/s	V_{2T}, m/s	V_{2E}, m/s	K_{zVT}	K_{zVE}	ΔK_{zV},%
0.5	2.82	2.22	2.19	0.787	0.777	-1.2
0.75	3.37	2.57	2.68	0.764	0.795	3.9
1.0	3.72	2.80	3.11	0.753	0.836	9.9

From the table 2.4 we see that the difference between the theoretical and experimental values of the coefficient of change of speed does not exceed 10%. Therefore, we can assume that the reliability of theoretical calculations is quite high. Experiments to determine the parameters of lateral scattering of seeds were carried out at heights of the fall H – 0.5 and 1.0 m, and angles α – 20 and 30° (Fig. 2.2). According to the results of processing the corresponding measurements (Fig. 2.9) the following characteristics were obtained for winter wheat: the average value of the angle $\tau(m_\tau)$ 18.1 - 22.1°; coefficient of variation – 60 - 64%; the average value of the flight range $l(m_l)$ – 0.13 - 0.16 m; coefficient of variation – 34 - 37%. As the angle of installation of the plane to the horizon α and the height of the fall H increases, there is a tendency to decrease the value of m_τ and m_l. Characteristic values for some crops vary from 7 to 11%. The deviation of the plane of the flight trajectory of the seed from the central contributes to a more uniform distribution of seeds at the bottom of the furrow.

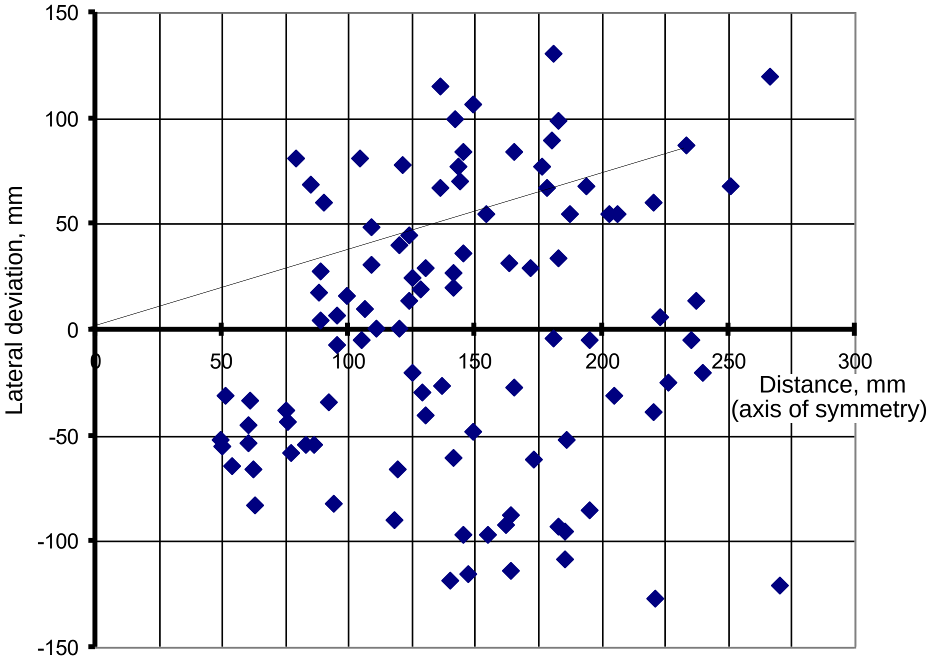


Fig. 2.9 The nature of the placement of seeds after reflection

The main parameters of the reflector-distributor with a flat reflection surface are: the height of the reflection point above the bottom of the groove – H_1; the angle between the direction of the speed V_1 at which the seed is fed to the reflector and the horizontal plane – ε; the angle between the reflective and horizontal planes – α_1; the angle between the line of intersection of the reflective and horizontal planes and the direction of movement – γ; height of subblade area – h. According to the results of previous research, we accept H_1=0.02 m. According to our reasonable scheme of the reflection process $\varepsilon=90°-\alpha_1$. To substantiate the optimal values α_1, γ and h the method of planning a multifactorial experiment is applied. After processing the experimental data, we obtained the regression equation of the coefficient of variation of the distribution of seeds along the width of the wing of the cultivator blade, which will look like this:

$$\nu = 29 + 3.875\cdot\alpha_1 - 2.875\cdot\gamma - 11.75\cdot h + 12.65\cdot\alpha_1^2 + 8.65\cdot\gamma^2 + 7.40\cdot h^2 + 3.25\cdot\alpha_1\cdot\gamma - 2\cdot\gamma\cdot h\,. \quad (2.8)$$

At fixed values h, the regression equations will look like this:

- at $h = 0.02$ m: $\nu = 1193 - 21.134\cdot\alpha_1 - 12.14\cdot\gamma + 0.0325\cdot\alpha_1\cdot\gamma + 0.127\cdot\alpha_1^2 + 0.0865\cdot\gamma^2$, (2.9)

- at $h = 0.03$ m: $\nu = 1185 - 21.134\cdot\alpha_1 - 12.34\cdot\gamma + 0.0325\cdot\alpha_1\cdot\gamma + 0.127\cdot\alpha_1^2 + 0.0865\cdot\gamma^2$, (2.10)

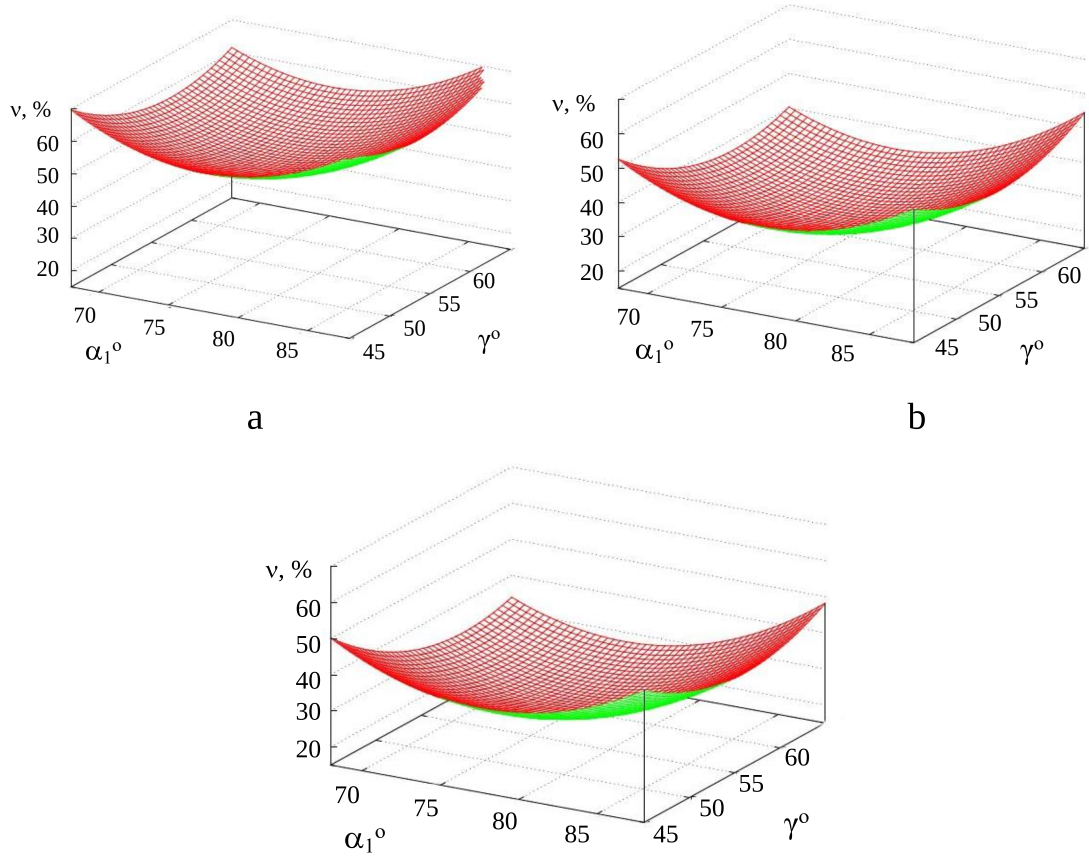
- at $h = 0.04$ m: $\nu = 1191 - 21.134\cdot\alpha_1 - 12.54\cdot\gamma + 0.0325\cdot\alpha_1\cdot\gamma + 0.127\cdot\alpha_1^2 + 0.0865\cdot\gamma^2$. (2.11)

The corresponding response surfaces are shown in Fig. 2.10. From the figure we see that depending on the angles α_1 and γ there are minimum values of the coefficient of variation ν of seed distribution, which correspond to the optimal values of α_1 and γ. As the value h increases, ν decreases (the indicator improves) to a certain value h, and then does not change. For example, at α_1=78° and γ=55° for h=0.02 m, ν=48.2% and for h=0.04 m, ν=24.7%. Given the fact that when increasing h by more than 0.03 m value ν decreases slightly, it is advisable to consider the condition h=0.03 m. The optimal value of the angle γ at h=0.03 m is determined by equation (2.10) under the condition $d\nu/d\gamma=0$. Then at α_1=78° value γ_{opt}=56.6°. As well as according to theoretical researches (Fig. 2.1) at α_1=78°, optimum values of an angle γ are within 55 - 60° that coincides with experimental data. Thus, the following parameters of the prismatic reflector-distributor are substantiated by experimental research: $\varepsilon=90°-\alpha_1=12°$; α_1=78°; γ=56.6°; h=0.03 m; H_1=0.02 m.

Field research was conducted at the NULES of Ukraine "Agronomic Research Station" on winter wheat. A serial cultivator blade 33 cm wide with a special riser and a reflector-distributor with reasonable parameters was installed instead of the disk opener of the John Deere N542C seeder. The following agrotechnical indicators of work for experimental and serial openers, respectively, were obtained: average depth of wrapping – 3.8 and 4.1 cm; coefficient of variation – 15.3 and 19.7%; number of spikelets per 1 m^2 – 612 and 496; grain weight of one spikelet – 1.54 and 1.48 g. The quality index of seed distribution in the transverse direction (coefficient of variation) for the experimental opener – 28.6%. Estimated increase in yield – 21.6%, when calculating the specific energy consumption, the equation for determining the power on the tractor hook N_w will look like this:

$$N_w = N_{eH} \cdot \eta_N \cdot \eta_{mT} \cdot \left[1 - A_\delta + \left(B_\delta - f \cdot g \cdot \{3.6 \cdot \eta_N \cdot \eta_{mT} \cdot W_e\}\right) \cdot V_P\right], \qquad (2.16)$$

where N_{eH} – rated effective engine power of the tractor, kW; η_N – coefficient of use of nominal effective power of the engine; η_{mT} – efficiency of tractor transmission; A_δ, B_δ – factors that determine the dependence of the skid coefficient δ on the operating speed V_p (subject to constant engine load) dependence $\delta=A_\delta-B_\delta\times V_p$ (based on the results of the analysis of traction characteristics for tractors John Deere 3071 A_δ=20.6 and 18.5; B_δ=0.99 and 0.90 respectively); f – coefficient of resistance to rolling of the tractor; W_e – energy saturation of the tractor, kW/t.



a b

c

Fig. 2.10 Graphs of the dependence of the coefficient of variation ν on the angles γ and α_1

a – h=0.02 m; b – h=0.03 m; c – h=0.04 m

To determine the specific resistance of the working machine K_V (kN/m) used the dependence $K_V=K_0+K_0\times K_s\times(V_p-V_0)$, where K_0 – specific resistance at $V_p=V_0$=5 km/h, kN/m; K_s – the rate of increase of traction resistance with increasing speed by 1 km/h (in fractions of a unit); V_p – working speed of the unit, km/h. Before determining the energy-saving working speed of the sowing unit, combined graphs of dependences of the width of capture B_p, specific resistance K_V and productivity on the working speed V_p were constructed under the condition of using 90% of the effective power of the tractor engine, when plotting the factors of formula (16) are taken as follows: N_{eH}=58 kW; η_N=0.9; η_{mT}=0.9; f=0.18; W_e=18 kW/t; A_δ=0.206; B_δ=0.99; K_s=0.045. It was found that for John Deere 3071 more energy-saving operating speed of 8.4 km/h with a width of B_p=5.4 m.

Conclusions to Chapter 2

It is established that neglect of air resistance in the seed line leads to an error of more than 15%. In the process of movement of seeds on the lower curvilinear section at the input speed of 1.5 - 3 m/s speed losses are distributed as follows: due to the force of friction from the gravity of the seeds – 7 - 13 %; friction force from the centrifugal force – 71 - 82%; air resistance –12 - 16%.

The dependences of the length L_2 and height z_3 of the subblade area on the parameters of the cultivator blade and the speed of the seeder-cultivator V_e were performed analytically according to the calculations performed for a typical cultivator blade (b=0.33 m) at V_e from 2.0 to 2.5 m/s (7.2 - 9.0 km/h), L_2 from 0.17 to 0.26 m; z_3 from 0.031 to 0.036 m.

Field tests of the experimental cultivator blade opener show that the coefficient of variation of the depth of seed earning decreases compared to the serial disc opener, by 4% (from 15.3 to 19.3%); the uniformity of seed distribution in the transverse direction is 28.6%; the number of spikelets per 1 m^2 increases by 23% (from 496 to 612 pcs.) with a grain weight of one spikelet of 1.48 - 1.54 g, which allows us to predict an increase in yield by 22%.

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CHAPTER 3. RESEARCH ON FUEL CONSUMPTION FOR DIFFERENT VALUES OF CAPACITY FACTOR OF ENGINE OF COMBINE HARVESTER

Introduction

Growth of agricultural production during the last years stipulated expansion of demand on an agricultural technique from the side of agrarians [1], that gave the shove of both market of technique development, domestic mechanical engineering and import of products from the leading world producers [2].

It is difficult to understand even to the specialists the matters of acquisition of economically efficient for the household, high-quality technique, among the variety of combine harvesters [3]. Thus collection of grain-crops is the decisive, final stage that gives a big impact on a production cost of grain in an agricultural enterprise [4]. Estimating the prospect of acquisition of combine harvester, the Ukrainian consumer pays the special attention on its fuel economy [5].

For this purpose, is used the index of specific fuel consumption (consumption of fuel in kg or litres per 1 ton of collected grain) [6]. Monitoring of work in the real exploitation of combine harvesters shows that this index varies from 2,5 l/t to 7,0 l/t [7]. It is obvious that on the fuel economy of combine harvester affects the brand of engine of combine harvester and specific agrobiological, organizational, service conditions of its use by an agricultural enterprise [8].

Exact answer on the fuel consumption of combine harvester at collection of grain-crops it is possible to get only by testing in the specific conditions of the specific enterprise [9].

However, it is almost impossible to provide mentioned approach due to its costliness and duration [10].

Therefore, the development of model that would enable conducting such estimation virtually by a computer graphic design is relevant and perspective.

Purpose of research

The research is based on theoretical researches and on monitoring of work of combine harvesters in the conditions of real exploitation. The information got from monitoring is a basis for making decisions on the modeling of fuel consumption of a combine harvester.

Materials and methods

The fuel consumption of the combine harvester is usually affected by the construction features of the combine harvester: productivity, engine capacity and fuel economy of the engine, the size of the silo and the grain unloading speed [11], the transportation speed of the combine harvester and other features [12]. On the other hand, the fuel consumption is significantly affected by the harvesting conditions, that are characterized by the following indicators: type and yield of the crop, straw condition, selected mode of crop harvesting of non-cereal part (swath or shredding with spreading), straw and grain humidity, clogging, fallowness, rut length, slope of the field, humidity and hardness of the soil, time of transportation to the grain unloading place and to the place of night parking [13].

The periods of operating time, related with a work of engine of combine harvester and its fuel consumption, include: time spent on the main work (threshing of the grain) [14], time spent on turning at the end of the fill [15]; time spent on moving the combine harvester to the place of unloading of grain and back; time spent on unloading grain into the vehicle; idle time moving (to the night parking place and back, from field to field also) [16]. We will consider fuel consumption in each of these periods of time [17]. At the implementation of basic work, the engine power of combine harvester is outlaid on the movement of combine harvester on the field [18], threshing of corn mass, growing and throwing (collection or growing shallow) of straw [19]. Specific fuel consumption of combine harvester at basic work it is possible to describe by a formula:

$$g_{K1} = q_1 \cdot Ne_n \cdot \xi_1 \cdot \left(10^3 \cdot W_{h1} \cdot \gamma_n\right)^{-1}, \text{ l/t} \tag{3.1}$$

where: q_1 – actual specific fuel consumption by engine at use of power factor, gram/kilowatt-hour; ξ_1 – coefficient of the use of effective power for main operations of combine harvester; Ne_n – operating engine power, kilowatt; W_{h1} – productivity of the combine for the basic working time, calculated on carrying capacity of combine harvester, t/h; γ_n – specific weight of diesel fuel, kg/l.

Productivity during performing the main operations (harvesting) is calculated using dependency:

$$W_{h1} = 3.6 \cdot q_n \cdot k_y \cdot (1 + \delta_c)^{-1}, \text{ t/h} \tag{3.2}$$

where: q_n – nominal (passport) carrying capacity of combine harvester, kg/sec; k_y – coefficient which takes into account the terms of collection (humidity, clogging, straw condition); δ_c – straw condition (ratio of non-grain part to grain mass unit).

In the calculations as the nominal carrying capacity of a particular brand of combine harvester was taken the numerical value written in passport [20], and in the case of absence of such information, was calculated by the method [21]. Calculation of the coefficient of harvesting conditions k_y was performed according to the following method [22]. Numerical value of specific fuel consumption by combine harvester during turnings at the end of bend is determined by following dependence:

$$g_{K2} = \tau_2 \cdot q_2 \cdot Ne_n \cdot \xi_2 \cdot \left(10^3 \cdot W_{h1} \cdot \gamma_n\right)^{-1}, \text{ l/t} \tag{3.3}$$

where: τ_2 – specific working time spent for turning at the end of the bend, hour; q_2 – actual specific fuel consumption by the engine at use of power factor ξ_2, gram/kilowatt-hour; ξ_2 – coefficient of the use of effective power during turnings at the end of bend.

Specific expenses of working hours on turnings can be determined by a formula:

$$\tau_2 = T_2 \cdot W_{h1} \cdot \left(0.36 \cdot L_g \cdot B_p \cdot U\right)^{-1}, \text{ hours} \tag{3.4}$$

where: T_2 – average duration of turning of the combine harvester at the end of the bend, hour; L_g – average length of the rut, m; B_p – working width of capture, m; U – average yield, t/ha.

Specific fuel consumption of the combine harvester during moving to the place of unloading of grain and back:

$$g_{K3} = \tau_3 \cdot q_3 \cdot Ne_n \cdot \xi_3 \cdot \left(10^3 \cdot W_{h1} \cdot \gamma_n\right)^{-1}, \text{ l/ha} \quad (3.5)$$

where: τ_3 – specific working time spent for moving to the unloading point of the bunker and back, hour; q_3 – actual specific fuel consumption of the engine at use of power factor ξ_3, gram/kilowatt-hour; ξ_3 – coefficient of use of effective power during moving to the unloading point of the bunker and back.

Calculation of the specific time spent on movements is determined by the following dependence:

$$\tau_3 = T_3 \cdot W_{h1} \cdot (3600 \cdot G_b \cdot \rho_3)^{-1}, \quad (3.6)$$

where: ρ_3 – specific weight of grain, t/m^3; G_b – combine harvester bin volume, m^3; T_3 – average time for combine movement to and from the place of unloading, sec.

Specific fuel consumption during unloading of the grain harvester bunker into transportation vehicle is determined by the following formula:

$$g_{K4} = \tau_4 \cdot q_4 \cdot Ne_n \cdot \xi_4 \cdot \left(10^3 \cdot W_{h1} \cdot \gamma_n\right)^{-1}. \quad (3.7)$$

Specific working time spent on unloading of the combine harvester bunker can be determined by the following formula:

$$\tau_3 = W_{h1} \cdot (3.6 \cdot V_r \cdot \rho_3)^{-1}, \quad (3.8)$$

where: V_r – speed of unloading of grain from the bunker, kg/sec.

During harvesting combine harvester moves from field to field and from the parking lot to the field [23]. In this case, the engine is not running at its full power and is actually idling [24]. The specific fuel consumption at such operations is determined by the following formula:

$$g_{K5} = \tau_5 \cdot q_5 \cdot Ne_n \cdot \xi_5 \cdot \left(10^3 \cdot W_{h1} \cdot \gamma_n\right)^{-1}, \quad (3.9)$$

where: τ_5 – specific working time spent on relocation; q_5 – actual specific fuel consumption of the engine at use of power factor ξ_5, gram/kilowatt-hour; ξ_5 – coefficient of use of effective power during moving from field to field and from parking place to field.

Specific working time spent on relocation are approximately determined using the following dependency:

$$\tau_5 = 2 \cdot L_s \cdot (T_{1d} \cdot V_p)^{-1} + L_f \cdot W_{h1} \cdot (S_s \cdot V_p \cdot U)^{-1}, \qquad (3.10)$$

where: L_s – average distance from the parking lot (machine yard) to the place of work (field), km; T_{1d} – average basic working time of a combine harvester per day, hour; V_p – speed of the combine harvester during relocation, km/h; L_p – average distance from field to field, km; S_s – average area of the harvesting field, ha.

Specific fuel consumption can be determined using the analytical model of Leiderman [25], which is a system of polynomial functions with constant coefficients, also taking into account the environmental impact:

$$g_f = g_n \cdot \left(A - B \cdot n \cdot (n_n)^{-1} + C \cdot n^2 \cdot (n_n)^{-2}\right), \text{ gram/kilowatt-hour} \qquad (3.11)$$

where: g_f – actual specific fuel consumption, gram/kilowatt-hour; g_n – documented specific fuel consumption at rated engine speed, gram/kilowatt-hour. Thus for simplification the coefficient of the use of power will be marked as $n \cdot (n_n)^{-1} = k_N$.

Results and discussion

Using the specified method of determination of parameters of Leiderman's functions was made the specified equation of specific fuel consumption, taking into account construction, terms of work and influence of environment $g_f = g_n \cdot \left(1.8757 - 1.7471 \cdot n \cdot (n_n)^{-1} + 0.8714 \cdot n^2 \cdot (n_n)^{-2}\right)$.

Taking into account the entered coefficient of the use of power k_N we will have:

$$g_f = g_n \cdot \left(1.8757 - 1.7471 \cdot k_N + 0.8714 \cdot k_N^2\right). \qquad (3.12)$$

As the engine of combine harvester in terms of operation, is different from the tractor (higher forcing, distribution of power take-off on two sides, more difficult working conditions (dustiness, constantly high temperature, etc.), nominal crankshaft speed), the following linear correlation is appropriate $g_f \cdot (g_n)^{-1}$ on the described modes of work (main work, relocation, turnings). Therefore for the calculation of actual specific fuel consumption of engines g_1, g_2, g_3, g_4, g_5 it is suggested to use equation

(12) in figure 1. Modern combine harvesters are equipped with engines that have sufficient power for various harvesting methods and technologies, for difficult conditions too. For different types of work, the engine has its value of the load factor. In particular for normal harvesting conditions during laying of straw in a roll $\xi_1 = 0.7...0.75$ (figure 3.1); during grinding of straw by a shredder $\xi_1 = 0.8...0.9$. Other values of engine load factor during cornering ξ_2, during relocation to the place of unloading of the grain tank ξ_3, during unloading of the grain tank ξ_4, during moving to the parking place and to the field ξ_5 differ slightly and are within range $0.2...0.35$. The final specific fuel consumption of combine harvester during harvesting is determined by the sum of:

$$\xi_\Sigma = \sum_{i=1}^{5} \xi_i = \xi_1 + \xi_2 + \xi_3 + \xi_4 + \xi_5 . \tag{3.13}$$

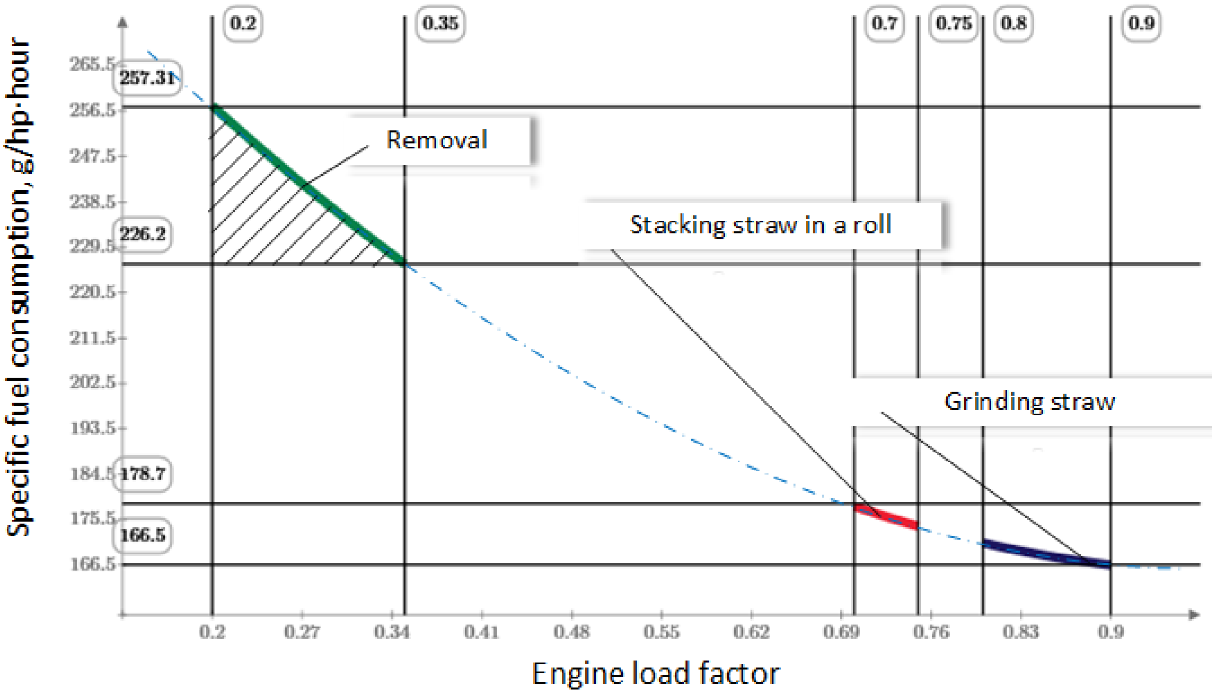

Fig. 3.1 Estimated value (12) of actual specific fuel consumption for different values of use of power coefficient

Obtained mathematical model allows us to establish the influence of individual factors, typical for different working conditions, on the specific fuel consumption of combine harvesters. The result of this work is the determination of the dependence of specific fuel consumption change on yield and straw condition of the crops. The object of the study is the combine harvester Slavutich KZC-9F. More than 300 units of Slavutich KZC-9F operate in the fields of Ukraine. As the conditions of use were taken average conditions for the forest steppe zone of Ukraine. According to this model was developed a program for determining the specific fuel consumption (figure 3.2). Delphi 7 was used for programming. These dependencies (figure 3.3) showed that specific fuel consumption of the combine harvester tends to decrease with increasing of crop yield. There is a sharp decrease in the segment of low yield, regardless of the straw condition.

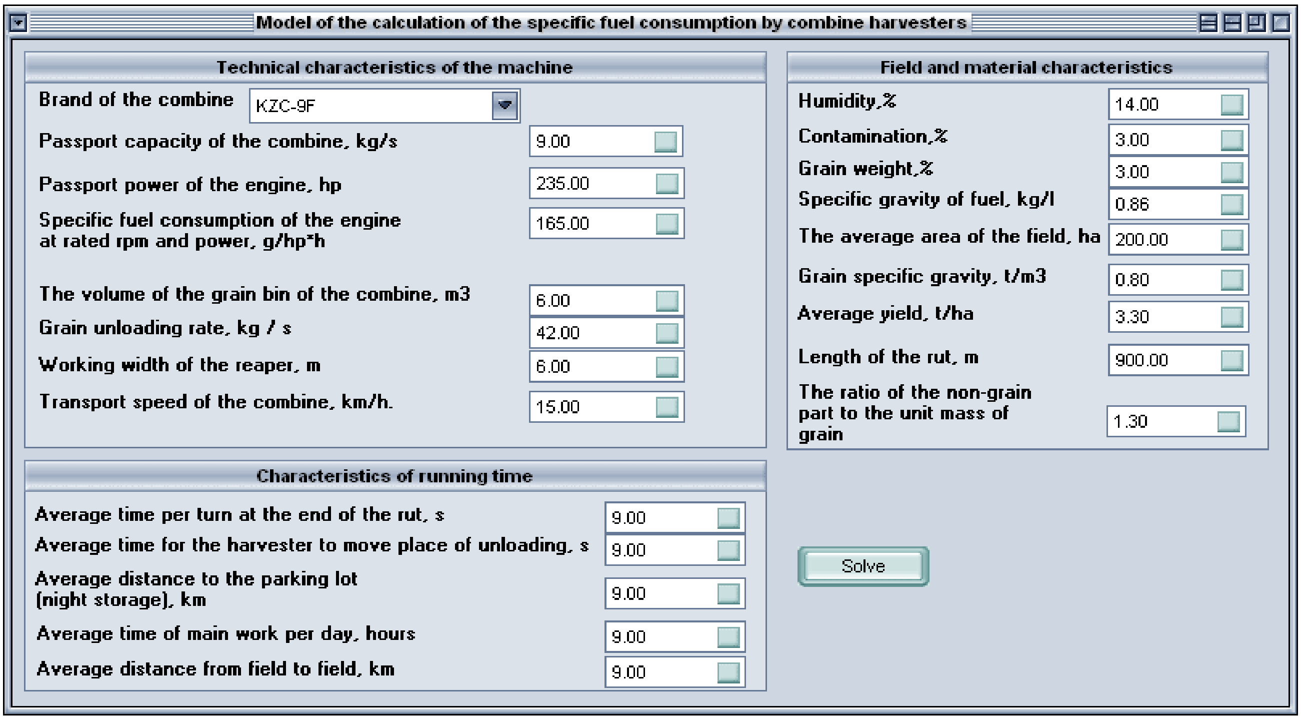

Fig. 3.2 Main page of the program for calculating the fuel consumption.

This happens due to the underloading of thresher by grain and of the combine harvester engine in general. When the normative yield, that provides documented carrying capacity, is reached, further reduction of the specific fuel consumption becomes minimal. This reduction is greater for a straw – 1:1.0. Therefore, when the ratio of grain to the non-grain part of the mass is 1:1.5 and the yield is increased to 2.5 t/ha, we observe a sharp decrease in the specific fuel consumption.

Fig. 3.3 Dependence of specific fuel consumption of combine harvester KZC-9F on yield and straw conditions.

For example, at yield 1.5 t/ha specific fuel consumption is 4.52 l/t, and at 2.5 t/ha – 3.58 l/t. In addition, with a further increase in yield, the specific consumption slightly decreases and at 4.5 t/ha it becomes 3.52 l/t. The straw condition index also significantly affects the formation of the specific fuel consumption of the combine harvester. With a yield of 4.5 t/ha and a ratio of grain to the non-grain part of 1:1.0, the specific fuel consumption is 2.83 l/t, and at the same yield and straw rate and a ratio of grain to the non-grain part of 1:2, the specific fuel consumption is 4.15 l/t (increased on 47%). Also straw rate affects the extreme point. So the sharp transition of reduction of specific fuel consumption for straw condition index occurs at a yield of 2.9 t/ha, and at a straw rate of 1:2 this change occurs at a yield of 1.9 t/ha.

Conclusions to Chapter 3

1. The authors found out that specific fuel consumption of combine harvester tends to decrease with increasing of crop yield. There is sharp decrease in segment with low yield, regardless of straw condition. This happens due to under loading of harvesting weight of thresher and of combine harvester engine in general. When the normative yield, which provides passport throughput of combine, is

achieved, further reduction of specific fuel consumption becomes minimal. This reduction is greater for straw – 1:1.0. Therefore, when the ratio of grain to non-grain part of mass is 1:1.5 and yield is increased to 2.5 t/ha, we observe sharp decrease in specific fuel consumption.

2. The straw index also significantly influences the formation of specific fuel consumption of combine harvester. With yield of 4.5 t/ha and ratio of grain to non-grain part of 1:1.0, the specific fuel consumption is 2.83 l/t, and at the same yield and straw rate and ratio of grain to non-grain part of 1:2 the specific fuel consumption is 4.15 l/t (increase of 47%). Also straw rate affects the extreme point. So the sharp transition of reduction of specific fuel consumption for straw rate occurs at yield of 2.9 t/ha, then at straw rate of 1:2 this change occurs at yield of 1.9 t/ha.

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CHAPTER 4. CHANGE OF TECHNICAL CONDITION AND PRODUCTIVITY OF GRAIN HARVESTERS DEPENDING ON TERM OF OPERATION

Introduction

Modern methodological approaches to the development of technological maps of harvesting do not take into account the service life of the combine [1] and its technical condition [2]. Failure to consider the previous service life of combines [3] and the probable values of technical condition leads to significant miscalculations in forecasting the timing of the harvest [4]. For machines of short seasonal use [5], the most significant of the indicators of reliability as serviceable systems are statistical or probable values of operating times between the next failures [6] and the duration of downtime [7].

Modern researchers [8, 9] have found that a comprehensive indicator of the efficiency of recovery systems using the coefficients of technical readiness [10]. This rogue is calculated as the ratio of the average operating time between failures to the average operating time between failures plus the average downtime to eliminate failures [11].

Purpose of research

The purpose of the research is to establish the empirical dependences of changes in the technical condition and productivity of combine harvesters Slavutich KZC-9F depending on the service life.

Materials and methods

Since the number of tested combines was constant (48 combines), the number of recorded failures, or the total number of hours worked to calculate the reliability indicators with a given accuracy [12], was determined for the recoverable objects [13]. Supervision of combines operation was performed according to the plan $NM(F\sum, T\sum)$

[14], according to which $N = 48$ combines were simultaneously studied. After each failure of the robot [15], the ability of the combine was restored $F\sum = 80$ failures, or when reaching the total operating time of all combines reached $T\sum = 257$ hours. Conditions for conducting rattan research of the season $F\sum = 80 < 109$; $T\sum = 257$ hours $\ll 1500$ hours. For plans NMF [16] and NMT [17] we calculate a sample estimate of operating time for failure of combine for harvest period of the first year of operation: $\widehat{\text{T}}\text{o} = S \cdot m^{-1} = 241 \cdot 13^{-1} = 18.53$ hours. Selective estimate of the average duration of recovery $\widehat{\text{T}}\text{в}=\text{m}^{-1}\sum_{\text{i}=1}^{m}\text{tв} = 58 \cdot 13^{-1} = 4.46$ hours.

The point estimate of the readiness factor [18] for the combine for the harvest period of the first year of operation is calculated by the formula: $\text{K}_{\Gamma}^{1} = \widehat{\text{T}}_0 \cdot \left(\widehat{\text{T}}_0 + \widehat{\text{T}}_\text{B}\right)^{-1} = 18.53 \cdot (18.53 + 4.46)^{-1} = 0.81$.

Point estimation of readiness coefficients [19] for combines for the harvest period of the first year of operation, respectively for combines №2 K_Γ= 0.58; №3 K_Γ=0.73; №4 K_Γ=0.51; №5 K_Γ=0.76; №6 K_Γ=0.59; №7 K_Γ=0.76; №8 K_Γ=0.57; №9 K_Γ=0.58; combine №10 was decommissioned due to engine failure after 60 hours [20].

The average value of the readiness factor for nine combines is determined from the formula: $\text{K}_{\Gamma}^{1} = \text{N}^{-1}\sum_{i=1}^{N} K_{\Gamma i} = 9^{-1}(0.80 + 0.58 + 0.73 + 0.51 + 0.76 + 0.59 + 0.76 + 0.57 + 0.58) = 0.65$.

Results and discussion

The regularity of the decrease in the average value of the readiness factor with increasing service life is shown in figure 4.1 and the polynomial of the fourth degree: $K_{\Gamma i} = -0.0043X^4 + 0.0784X^3 - -0.5072X^2 + 1.3051X - 0.3009$ $\quad \text{R}^2 = 0.9862$.

During the five years of operation after 2013, the value of the readiness factor decreased from the value of $K_{\Gamma 2}$ = 0.86 to $K_{\Gamma 7}$ = 0.64, ie by 26%. The average seasonal value of the reduction of the readiness factor was $K_T = 0.044$, or 5.2% per year. According to the assessment [21], we assume that the statistical indicators of the coefficient of readiness of the pattern of change with increasing service life are close to the Weibull distribution [22]. For the second year of operation of the combine harvester determine. Stationary readiness factor for the second year of operation:

$$К_Г = \mu \cdot (\mu + w)^{-1} = 0.25 \cdot (0.25 + 0.040)^{-1} = 0.86.$$

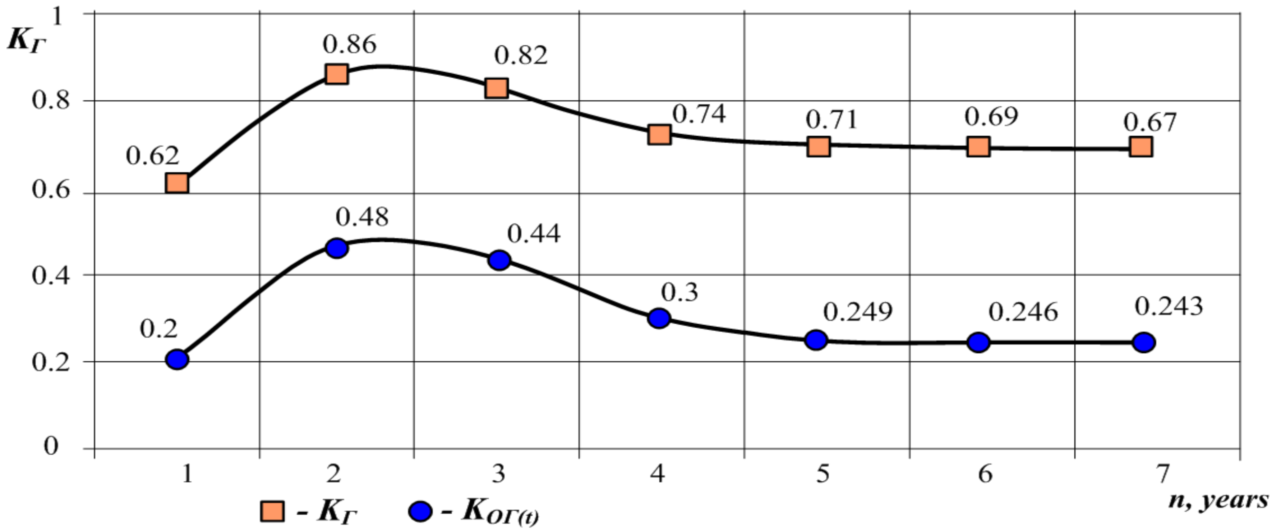


Fig. 4.1 Regularities of change of stationary coefficient of readiness and non-stationary coefficient of operational readiness.

Non-stationary coefficient of readiness to change jobs:

$$К_Г(f_3) = \mu \cdot (\mu + w)^{-1} + w \cdot (\mu + w)^{-1} \cdot e^{-(w+\mu)t_3} = 0.25 \cdot (0.25 + 0.040)^{-1} + 0.040 \cdot (0.25 + 0.040)^{-1} \cdot e^{-(0.25+0.040)} = 0.866 \approx 0.87.$$

Stationary coefficient of operational readiness for change:

$$К_Г(f_3) = \mu \cdot (\mu + w)^{-1} \cdot e^{-wt} = 0.86 \cdot e^{-0.041 \cdot 14} = 0.86e^{0.54} = 0.86 \cdot 1.77^{-1} = 0.86 \cdot 0.56 = 0.48.$$

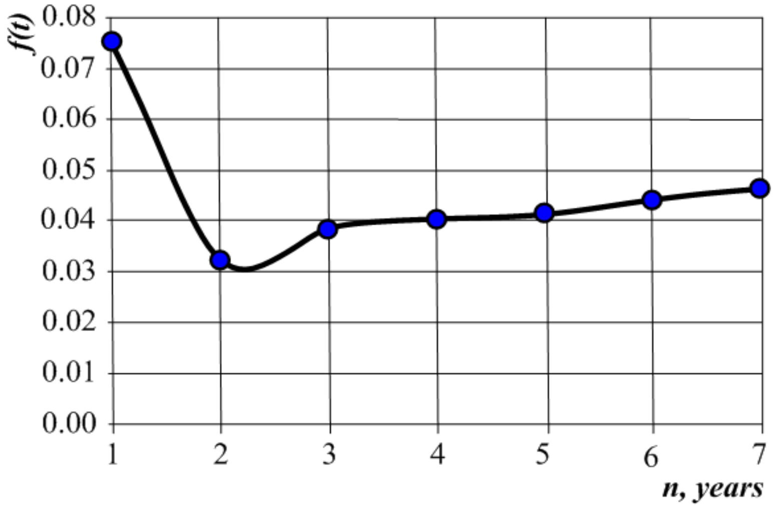
Non-stationary coefficient of operational readiness for a change of work:

$$К_Г(\tau, t) = \left[\mu \cdot (\mu + w)^{-1} + w \cdot (\mu + w)^{-1} \cdot e^{-(w+\mu)t_3}\right] \cdot e^{-wt} = 0.87 \cdot 0.56 = 0.49.$$

Stationary and non-stationary coefficient differ by 0.01, and stationary readiness factor ($К_Г$) and stationary operational readiness factor ($K_{О.Г}(t)$) differ $\Delta К_Г = K_Г - K_{О.Г}(t) = 0.86 - 0.48 = 0.39$.

We determine the main characteristics of the Weibull distribution for the coefficient of readiness with increasing service life. $\bar{Т}_н Г(1 + m^{-1}) w^{-1}$ – the average value of operating time between failures. $f(t_3) = m\omega(\omega \cdot t)^{m-1} e^{-(\omega t) m}$ – distribution density. $F(t_3) = 1 - e^{-wt}$ – failure probability distribution function. $D\bar{T}_{н} = w^{-2} \cdot [\Gamma(1 + 2 \cdot m^{-1}) - \Gamma^2(1 + m^{-1})]$ – dispersion.

Calculations of Weibull distribution characteristics for the second year of combine harvester operation (as an example): $\bar{T}_{н2} \Gamma(1 + 27^{-1}) \cdot 0.04^{-1}$ =22.23 hours. $f_2(t_3) = 2.70 \cdot 0.04(0.04 \cdot 14)^{2.70-1} e^{-(0.04 \cdot 14)^{2.70}} = 0.033.$ $F_2(t_3) = 1 - 2.71^{-(0.04 \cdot 14)^{2.7}} = 1 - 0.82 = 0.18.$ $D\bar{T}_{н2} = 0.04^{-2} \cdot [\Gamma(1 + 2 \cdot 2.7^{-1}) - \Gamma(1 + 2.7^{-1})] = 625 \cdot 0.1289 = 78.$ The patterns of change of the stationary readiness factor and the non-stationary operational readiness factor depending on the service life are shown in figure 1. Regularities of change of density of distribution K, and functions of distribution of probability of failure are shown in figure 4.2.



a) b)

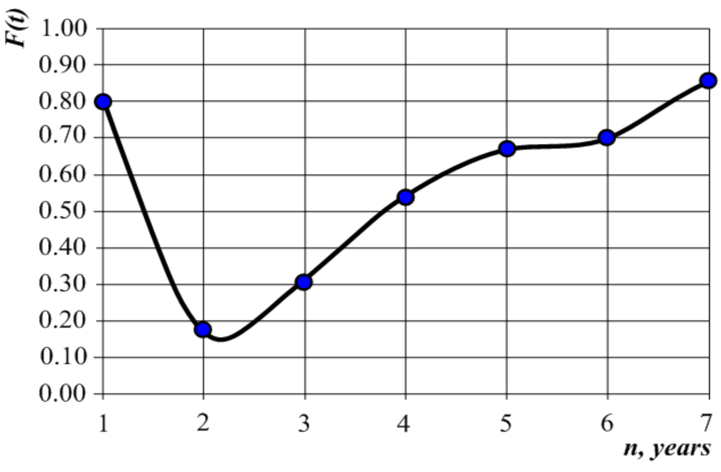


Fig. 4.2. The function of distribution: a – of operating values between failures with increasing time of operation; b – of probability of failure during the change with increasing time of operation.

The main characteristics of the Weibull distribution of combines with increasing effluents are summarized in table 4.1.

Table 4.1

Characteristics of Weibull distribution of operation of combines with increasing time of operation.

Years of operation of the combine	T, hours	σ_{kg}	$f(t)$	N
1	10.79	0.075	0.80	49
2	22.93	0.032	0.18	78
3	19.65	0.038	0.31	94
4	14.42	0.040	0.54	141
5	11.65	0.041	0.67	89
6	11.21	0.044	0.70	82
7	10.30	0.046	0.86	73

The figure 4.1 shows that the discrepancy between the experimental and calculated values of Weibull distribution is not significant (up to 3%). With the amount of information $N = 48$ and the absence of a statistical series, the standard deviation is determined from the formula: $\sigma_{kg}^1 == (N-1)^{-1/2} \cdot \left(t_i - t_{cp}\right)^2 = 0.141$. The value of the coefficient of readiness for the harvest of the first year of operation is: $К_{kg}^1 =$ 0.65±0.141. Determine the coefficient of variation: $V = \sigma_{kg}^1 \cdot (К_{Г}^1)^{-1} = 0.141 \cdot 0.61^{-1} = 0.34$. According to the known coefficient of variation y_p=0.34 from table 2 we find the parameters and coefficients of the Weibull distribution: в=3.22; B_B=0.896; C_B=0.306.

Find the offset parameter: $С = \bar{К}_{Г} - аК_{В} = 0.61 - 0.418 = 0.198$. Determine the upper confidence limit for the readiness factor: $К_{в} = (\bar{К}_{Г} - С) \cdot Z_1^{1/3.22} + С =$ 0.63. $К_{Н} = (\bar{К}_{Г} - С) \cdot Z_3^{1/3.22} + С = (0.61 - 0.198) \cdot 0.96 + 0.198 = 0.59$. Analysis of the rate of change of the indicator of selective assessment of the readiness factor allows us to conclude that the service life of the main units, systems, mechanisms of combines is exhausted by the service life of five to six years and a total output of 1200-1400 hours with an average output for the season of 200-280 hours. After the specified service life in years and production in m.-h. the operating time between failures decreases from 25 motor hours to 12-13 motor hours, the duration of downtime on failures is stabilized within 6 hours, and the rate of readiness is reduced to 0.67...0.64 (figure 4.3).

Table 4.2

Main characteristics of Weibull distribution of combines.

Indicators	Terms of operation						
	1	2	3	4	5	6	7
Experimental value 2	-	0.16	0.230	0.350	0.41	0.450	0.49
Estimated value 2	-	-	0.226	0.292	0.358	0.424	0.49
Infelicity, %	-	-	1.700	19.800	14.52	6.100	0

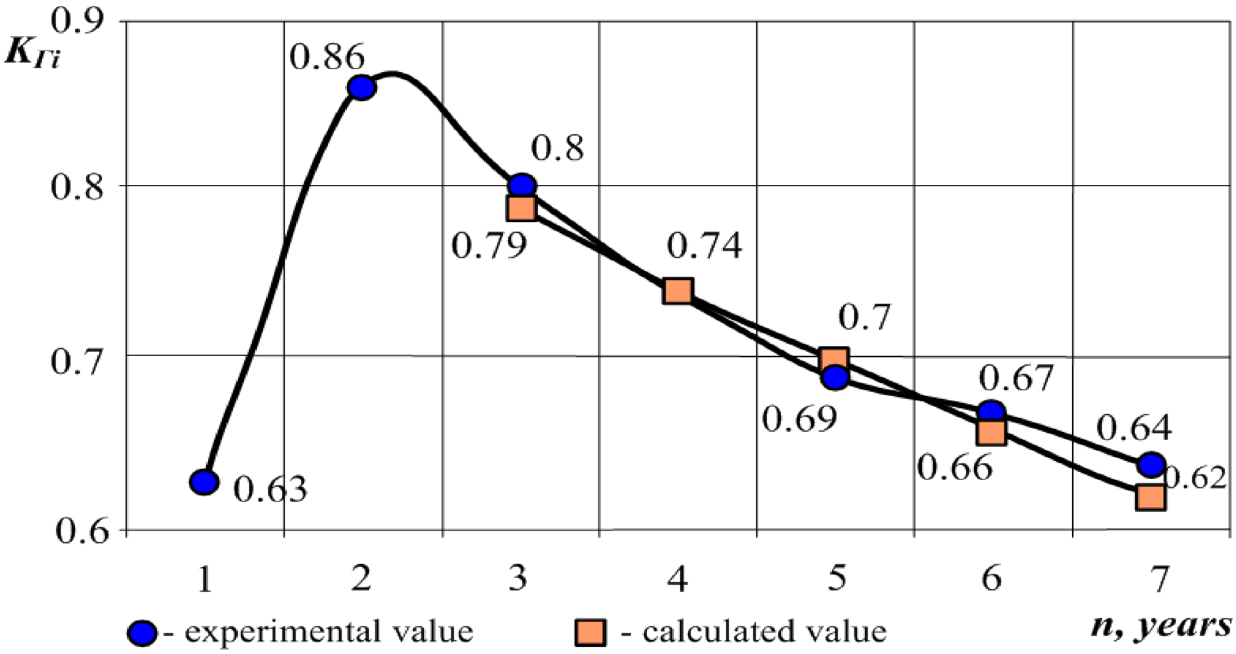


Fig. 4.3 Change of readiness coefficients of the combine depending on time of operation.

The probable value of the coefficient of readiness for the harvest after the second can be calculated empirically $К_{Гi} = К_{Г2} \cdot e^{-(0.015+0.005 \cdot x_i)3}$, where x_i – year of operation of combine. For example, for the third year of operation: $К_{Г3} = 0.86 \cdot e^{-(0.015+0.005 \cdot 3)3} = 0.79$. The average value of the reduction of the readiness factor for the operating season (within seven seasons of operation) can be determined by the formula: $\gamma_{Кс} = [К_{Гmax} - K_{1min}] \cdot [n - 2 \cdot (0.86 - 0.64) \cdot (7-2)^{-1}]$, season^{-1}. Within the normative costs for overhaul repairs of combines and maintenance of their operability during the harvest after the second season of operation, there is an annual decrease in the readiness factor by 0.036. The influence of the readiness factor on the performance of the combine is determined by its influence on the utilization time of the change. The coefficient of readiness determines the relative indicator of technical condition Z introduced by us, which is included directly in the equation according to which the coefficient of use of change time is determined. We accept that the value of Z increases linearly with increasing years of use of the combine. The change in the relative technical condition is shown in figure 4.4. Productivity (in tons) of combines with increase in service life due to decrease in coefficient of readiness decreases:

$$\gamma_Q = \left(Q_{\Gamma\max} - Q_{\Gamma\min}\right) \cdot (n-2)^{-1} = (1133 - 605) \cdot (7-2)^{-1} = 106, \text{ton/season}.$$

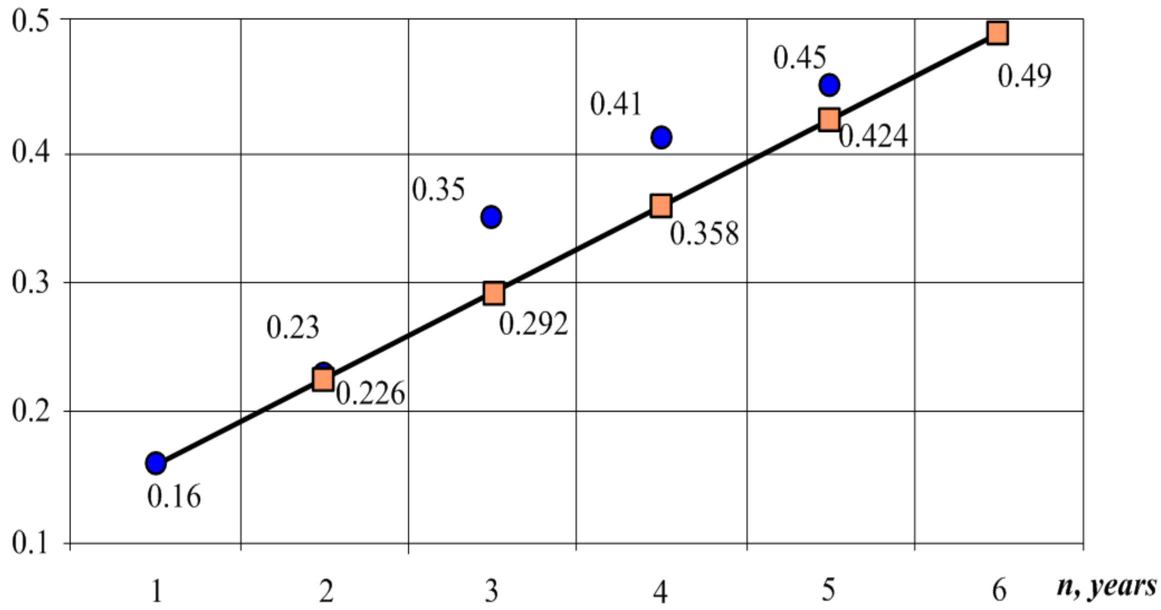


Fig 4.4 Change of a relative index of the technical condition 2 with increasing time of operation.

A decrease in the readiness factor by 0.01 causes a decrease in the combine's productivity by 21.2 t/season. Determine the specific decrease in productivity per unit of reduction of the readiness factor: $\Delta\gamma = \left(Q_{2\max} - \mathrm{Q}_{7_{\min}}\right) \cdot (\mathrm{K}_{1\prime 2\max} - \mathrm{K}_{1'7\min})^{-1} =$ 31.3 tons. According to the results of experimental research, the relative indicator of technical condition for the service life of seven years varies from the values of 0.16 (second year of operation) to the value of 0.49 (seventh year of operation) in figure 4. According to the obtained experimental data, the factors included in the equation have values $Z_2 = 0.16$, $K_2 = 0.066$, ie the equation will be as follows $Z = 0.16 + 0.066 \cdot (n_i - 2)$, where n_i – serial number of the year of operation. If the coefficient of readiness of the combine harvester decreases with increasing service life, the recovery factor increases in figure 4.5.

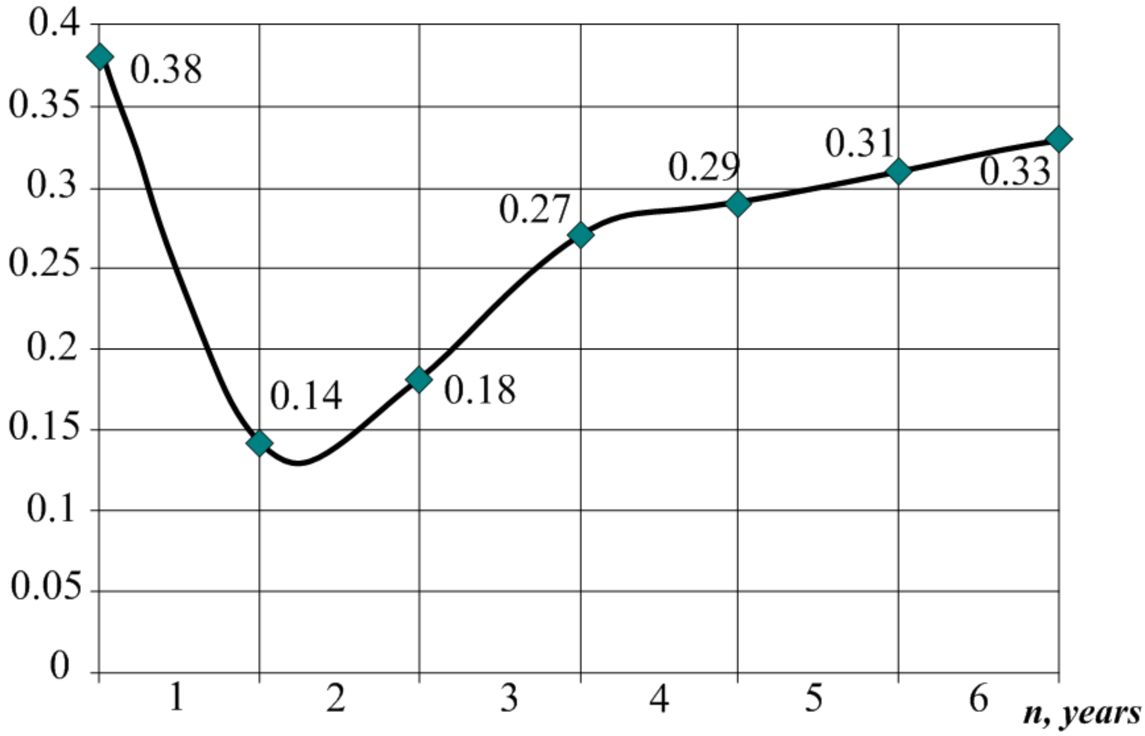


Fig. 4.5 Dependence of the recovery factor on the service life of the combine.

If the duration of downtime to eliminate failures after the third year of operation changes to a maximum of 22%, the recovery rate increases by 135%.

Conclusions to Chapter 4

1. During the five years of operation of the combine harvesters Slavutich KZC-9F, the value of the readiness factor decreased from 0.86 to 0.64, by 26%. The average seasonal value of the reduction of the readiness factor was 0.044, or 5.2% per year.

2. The productivity of the combine harvesters Slavutich KZC-9F increases with increasing service life due to a decrease in the readiness factor. A decrease in the readiness factor by 0.01 causes a decrease in the productivity of the combine by 21.2 tons/season.

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CHAPTER 5. MODELING THE WEIGHT OF CRITERIA FOR DETERMINING THE TECHNICAL LEVEL OF AGRICULTURAL MACHINES

Introduction

Prospects of technical solutions of agricultural machines are substantiated by their technical level [1]. Under the technical level of agricultural machinery is understood the degree of perfection of this technical solution in comparison with the basic model of the agricultural machinery [2]. The basic domestic [3] or foreign agricultural machine can serve as a basic sample [4]. Often the task of determining the technical level of agricultural machinery is reduced to a comparative assessment of domestic [5] and foreign agricultural machinery [6] when using them in the given natural and economic conditions of the country [7]. The solution of such a problem is considered in this article.

Solving this problem requires the following four steps.

Stage 1. Substantiation of evaluation criteria [8] and nomenclature of indicators of agricultural machinery [9].

Stage 2. Identification of alternatives to technical solutions of machine-tractor units [10] or technological complexes of agricultural machines [11].

Stage 3. The choice of method for calculating the technical level of agricultural machinery [12] and multicriteria evaluation [13].

Stage 4. Analysis of results [14] and development of proposals for decision making [15].

All four stages form the basis of the overall methodology of systems analysis [16]. To perform these steps [17], a variety of methods are proposed [18], including in regulations [19]. In this case [20], a generalized method for agricultural production is not and cannot be, because the evaluation criteria [21] and the nomenclature of indicators are different for different machine-tractor units [22]. However, common to all tasks is the fact that in the third stage it is necessary to determine and use the weight of the criteria [23].

The use of weights is provided by almost all existing methods of assessing the technical level of the agricultural machine [24], because the indicator of the technical level is a complex indicator [25], in the calculation of which weights are used [26]. The value of the coefficient of technical level of an agricultural machine depends on how objectively justified the weights [27]. Hence the conclusions for deciding on the use of this machine-tractor unit [28].

Purpose of research

The purpose of the research is devoted to the method of substantiation of the weight of the criteria in determining the technical level of agricultural machinery. The use of relative, in parts of change, values of evaluation indicators will allow to objectively and reliably obtain the value of the weight of the criteria.

Materials and methods

Consider the problem of determining the weight of the criteria in the study of the technical level of combine harvesters. On the basis of the analysis of normative documents [29], forecasts of development of equipment for assembly of grain crops the following evaluation criteria [30] and the indicators corresponding to them are chosen (table 5.1).

The weight of the criterion reflects the degree of its importance in the rank of the sequence [31], their values lie in the range 0...1 and are determined in the following ways: cost regression dependencies [32]; equivalent ratios [33]; expert assessments [34]; limit and nominal values [35].

The method of cost regression dependencies is used when conducting a comparative assessment of costs for the creation and operation of agricultural machinery, depending on the technical level of the machine itself. It can be used when assessing the technical level of domestically produced or developed agricultural machinery. It is impossible to estimate foreign-made agricultural machines by this method, because the costs of creating an agricultural machine are unknown.

Table 5.1

Criteria and indicators for assessing the technical level of combine harvesters.

Criterion	Indicator	The unit of measurement of the indicator
Reduction of direct costs	Operational expenses	(UAH/ha)
Reducing energy consumption	Direct fuel consumption	(l/ha)
Reducing labor costs	Specific labor costs	({people-hours}/ha)
Reducing material consumption	Specific material consumption	(kg/ha)
Improving reliability	Earnings on failure	(hours)

The method of equivalent ratios is used in cases where it is possible to justify the relationship between the number of agricultural machinery produced and the values of their technical level. For agricultural machinery, such a connection cannot be detected, even if it exists.

The method of expert assessments is used when other methods cannot be applied. Although this method is recommended by many authors and regulations, the resulting weights are subjective. Therefore, the method of limit and nominal values should be preferred, because it is an analytical method, and the limit values of evaluation indicators can be justified on the basis of analysis of changes in existing technical solutions of agricultural machinery over a period of time. The proposed formula for determining the weight V_i by the method of limit and nominal values has the form:

$$V_i = (p_{ih} - p_{in})^{-1} \cdot (\sum_{i=1}^{n} \{p_{ih} - \}^{-1})^{-1}, \quad (5.1)$$

where p_{in} – the limit (forecast) value of the indicator of the i-th criterion of the technical level of agricultural machinery; p_{ih} – nominal value of the indicator of the i-

th criterion of the technical level of agricultural machinery; n – the number of indicators of the technical level of agricultural machinery.

Results and discussion

The use of formula (5.1) gives results that contradict the essence of technical progress in agricultural engineering and agricultural operation. Let's show it on a simple example. Calculate the weights according to three criteria, the values of which and the results of calculations by formula (5.1) are given in table 2.

Table 5.2

Example of using dependence (5.1) to determine the weight of the criteria.

Evaluation indicator	The value of indicators		The weight of the criterion
	nominal	limit	
Operating costs, (UAH/ha)	20	10	0.270
Energy intensity, ({kW h}/ha)	8	4	0.676
Operating time on failure, (hours)	50	100	0.054

This example shows that it is desirable to double all indicators. It was expected that the weights for all criteria should be the same, but this did not happen. Therefore, formula (5.1) is not suitable for determining the weight, because the value of the weight is not affected by the relative differences between the limit and nominal value, and their absolute values.

We have proposed such a ratio to determine the weight of the criteria of the technical level of agricultural machinery, which allows to implement this feature. The greater the relative difference between the limit and nominal values for this criterion, the greater the weight. If this difference is equal to zero, the nominal value has reached the limit level, then the weight is also equal to zero. Therefore, to determine the weight of the criteria of the technical level of agricultural machinery, it is proposed to use not absolute values, but relative, i.e. the share of improvement of the criterion in the future. Then the formula for determining the weight will take the form:

$$V_i = (1 - q_i) \cdot (\sum_{i=1}^{n} \{1 - q_i\})^{-1}, \tag{5.2}$$

where q_i – the share of improvement of the i-th criterion of the technical level of agricultural machinery in the future:

$$q_i = p_{in} \cdot (p_{ih})^{-1}, \tag{5.3}$$

$$q_i = p_{ih} \cdot (p_{in})^{-1}, \tag{5.4}$$

Depending on the direction of improvement of the technical level of agricultural machinery, one of two formulas is used to calculate qi. If, in accordance with the requirements of technical progress, the indicator needs to be reduced, then formula (5.3) is used, and if the indicators need to be increased, then formula (5.4) is used. The nominal values are the average statistical values of the technical level of agricultural machinery, which were achieved during the assessment of the technical level for domestic machinery.

The limit values of the indicators are substantiated by the results of the forecast of the development of agricultural technologies and equipment, taking into account the current level achieved by foreign firms.

The advantage of the proposed method is that it is not necessary to know the numerical values (marginal and nominal), it is enough to know by what percentage or how many times you need to improve this indicator in order to then express the relative change in the share. This feature is quite important when using data on foreign machines, the achieved indicators of which are used in setting the limit values. For example, the operating costs of the Slavutich KZS-9M combine should be reduced by 30%. Then $q_i = (100 - 30)/100 = 0.70$. Let the energy consumption need to be reduced 1.4 times. Then $q_i = 100/140 = 0.714$.

Table 3 shows the values of changes in selected indicators for the period up to 2020, which are based on scientific and technical forecast of agricultural machinery for harvesting grain crops, and which should be used to assess the technical level of combine harvesters and the weight of criteria calculated by the formula (5.2).

Table 5.3

Limit values of technical level indicators and their weight for combine harvesters.

Indicator	The value of indicators		The value of change in indicator	q_i	V_i
	nominal	limit			
Operational expenses	100%	70%	reduce by 30%	0.700	0.1473
Direct fuel consumption	100%	75%	reduce by 25%	0.750	0.1228
Specific labor costs	1.00	0.714	reduce by 1.4 times	0.714	0.1405
Specific material consumption	100%	60%	reduce by 40%	0.600	0.1965
Earnings on failure	60 ha	240 ha	not variable	0.200	0.3939

It is established that in the process of operation of the combine harvester Slavutich KZC-9M with a reduction of 30% of operating costs, the weight in determining the technical level is 0.1473. Studies have shown that a 25% reduction in direct fuel consumption is 0.1228 weight in determining the technical level of the combine. Note that the obtained weights for the given conditions of improvement of the evaluation indicators reproduce the objective values. These values are normalized, i.e. $\sum V_i = 1$.

Conclusions to Chapter 5

The proposed model of substantiation of the weight of the criteria of agricultural machinery in determining the technical level of machinery allows to obtain results objectively and reliably, as it provides for the use of relative, in fractions of the values of evaluation indicators.

It is established that in the process of operation of the combine harvester Slavutich KZC-9M with a decrease of 30% of operating costs, the weight in determining the technical level is 0.1473. Studies have shown that a 25% reduction in direct fuel consumption is 0.1228 weight in determining the technical level of the combine.

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CHAPTER 6. RESEARCH OF MICRODEFORMATION AND STRESS IN DETAILS OF AGRICULTURAL MACHINES BY IMPLEMENTING HOLOGRAPHY

Introduction

During operation, agricultural machinery is exposed to various external and internal loads [1], which cause intense wear of working surfaces [2], material fatigue [3], and damage to contact surfaces [4]. The situation becomes especially dangerous when the power load is accompanied by the combined action of elevated temperatures [5], an asymmetric loading cycle [6] and a change in the lubrication regime [7]. In most cases [8], the destruction of parts begins from the surface layers and is determined by their resistance to wear [9], as well as the magnitude of the contact load in the working areas [10]. All existing measures to improve the reliability of agricultural machines are ineffective and imperfect if quality control of the manufacture and repair of products is not properly organized [11].

The presence of surface and hidden defects in parts, damages that manifest themselves in the form of internal metal delamination [12], intercrystalline and external cracks, curvature of crystal lattices, non-metallic inclusions, pores and fistulas, which leads to a loss of machine performance [13]. Untimely replacement of such products with defective surfaces will certainly lead to the appearance of a conditionally serviceable machine in operation [14]. The probability of failure of such machines is 50% and depends on the time when [15], under favourable conditions, an undetected defect or damage will destroy the part and disable the machine [16]. An acute problem is the rational use of domestic materials [17], the production of which is being established by agricultural engineering enterprises [18]. Improper use of materials without scientific justification and preliminary calculations for strength [19] and fatigue causes premature failures, loss of performance and low reliability of equipment [20].

Especially important is the task: to assess the residual life of the product, to determine the values of the parameters of its permissible or limiting states, to establish the value of internal and residual stresses [21], to find and register changes in the technical state of the product under the influence of the established loads [22]. This creates conditions for studying the stress state of parts [23], establishing the location and magnitude of stress concentrators [24], and thereby predicting the residual life of machines.

The study of the stress state of the working surfaces of parts is based on those developed by scientists from the Munich Center for Applied Optics (Germany), Ghent University (Belgium) with the direct participation of the authors. Analysis of the literature has shown that the study of the processes passing with the destruction of surfaces of the limiting state must be carried out in a complex combination of two types of optical interferometry [25]. This is due to the special possibilities of using each of the marked types of control for a specific type of research [26]. Computer holography makes it possible to register changes in the surface at low, not limiting loads, which pass at low rates of change in the state of the surface [27]. The image of a deformed body, represented in the form of colored interference fields, is recorded in the computer memory [28]. In the second case of holography, the object is captured in three-dimensional imaged on a film or glass photographic plate along with interference lines. It is used to study high-speed dynamic processes, sometimes associated with the destruction of the surface or the entire part [29]. The principle of operation of each type of holography is based on the double exposure method, when the body is observed before and after the application of the load. At the same time, changes in the state of the surface of the part are measured by comparing each of its sections with its changed state [30]. A defect or damage to a part manifests itself in a local anomalous placement of interference fringes. In the case of computer holography, these are colored stripes (each color corresponds to a certain amount of deformation), in another case, these are black and white stripes.

Purpose of research

The purpose of the research is to establish the feasibility of using holographic methods to determine the parameters of the technical condition of the working surfaces of parts, assemblies, aggregates and agricultural machines, to identify hidden defects, damages, to study the stress-strain state and related possibilities to increase the reliability of agricultural machines.

Materials and methods

To conduct research, low-pressure polyethylene was used as a sample material (representative of a wide range of agricultural machinery parts), for example pipes: diameter 120 mm, wall thickness 10 mm, pressure of the working area 0.3-0.4 MPa, product temperature at the moment the experiment was 0 °C. Cooled air was pumped into the initiation zone. The destruction was initiated by a scat pendulum. To trigger the laser, a Triger-system block device was used, the diagram of which is shown in figure 6.1.

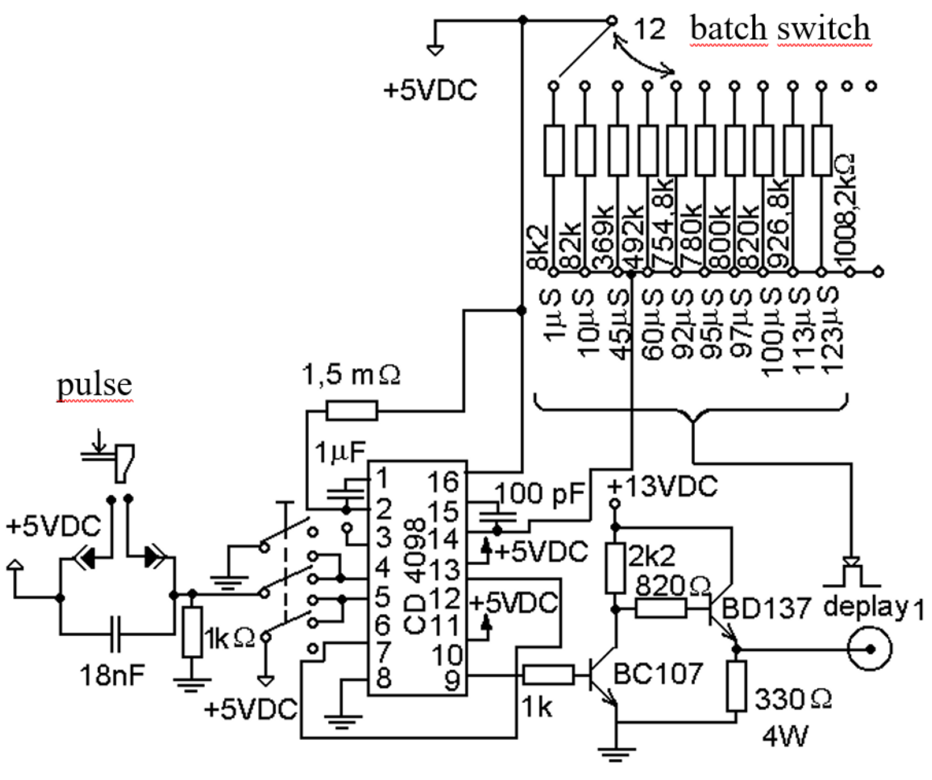

Fig. 6.1 The scheme of the device for launching the laser at set intervals Triger-system

Moreover, a switch was installed at a distance of 100 mm from the initiation zone, which was connected immediately before the start of the experiment to the Triger-system unit. The switch is a silver wire stretched between two electrodes in the zone of predictable destruction of the part. With the destruction of the part and the development of a crack along it at a speed of 500-560 m/s, the wire was broken and the retarder in the device block was set to a certain delay time. Then the laser was switched on and double fixation of the destroyed part was carried out. The interval between two laser flashes was constant at 20 ns.

During the experiment, two wave fields were recorded sequentially in the same hologram by the method of two exposures or frozen stripes. A pulsed ruby laser HLS-2 with a wavelength of 694 μm was used, and holograms were recorded on an AGFA film sensitive to red light. Examples of loading of parts are shown in figure 6.2. The laser was switched on not at randomly chosen times, but at precisely defined for each batch of investigated objects. The loads were carried out discretely, setting the weights of 50, 100, 200 for metal products and 5, 10, 20 N for non-metal ones, sequentially and repeatedly repeating the load until a clear interference pattern appears.

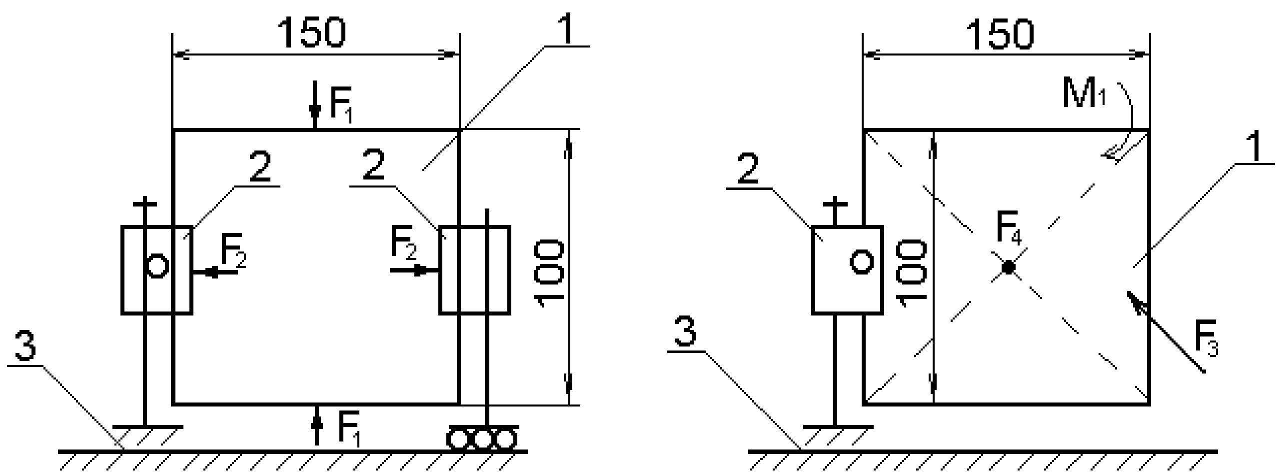

Fig. 6.2 Sample load diagram F_1 compression, F_2 tension, F_3 bending, M_1 torsion, F_4 thermal, 1 sample, 2 clamp bracket, 3 desktop.

Results and discussion

Table 6.1 given ranges of loads, at which the process of appearance of interference fringes occurs, depending on the adjustment of the optical system and the capabilities of computer support. The given load ranges cover the load range.

Table 6.1

Mechanical load when examining real parts.

Part name, material	Mechanical load type			
	F_1 (N)	F_2 (N)	F_3 (N)	M_1 (N m)
Internal combustion engine cylinder block, cast iron	400-500	-	-	40-60
Internal combustion engine piston pin, steel	-	800-900	750-900	80-100
Piston, aluminum	850-1100	500-620	-	-
Milk tap, steel	780-900	-	600-680	40-60
Cover, polyamide	150-200	-	18-22	-
Adapter connector, carbon fiber	30-34	25-27	18-20	-
Reducing sleeve, metal polymer	65-70	-	45-52	-

The calculation results were checked experimentally, which made it possible to refine the parameters of the holographic mode. As a disadvantage of the above methods, it should be noted that the experiments are carried out at clearly fixed discrete times. This does not make it possible to study the integral picture of the change in the microdeformation field of the product surface in time when the magnitude or nature of the load changes.

Disadvantages of the method of non-destructive testing by holography, which do not reduce the possibilities of its application. These include: all types of work should be performed only in a darkened room, the overall dimensions of the parts are limited by the actual dimensions of the film or photographic plates; the products must be painted prior to creating a matte surface (white or silver paint), protective screens must be used and there is no way to observe the products during the experiment in real time. It should be noted that some of the above disadvantages partially reduce the efficiency from the implementation of speckle interferometry - double-pulse holography, but do not completely eliminate it from the range of modern non-destructive testing methods. Under certain conditions, correctly selected nomenclature of parts, experimental conditions, as well as complex combination with other methods, for example; computer holography, acoustic, radiographic methods, it is possible to obtain results that correspond with a high measure of accuracy to real processes, and the obtained mathematical models adequately describe the physical phenomena that take place in the details of agricultural machinery.

The minimum of microdeformation and, accordingly, the minimum of stresses correspond to the areas with the minimum reproduced intensity of light bands. On computer holograms, these bands are in the middle range of the light spectrum, as indicated by the scales shown on the holograms. By studying holograms for each type of load and type of parts. Displacement of coloured stripes on the hologram indicates the presence of damage or defects. A sharp change in colour also gives information about the presence of damage or defects. As it was established experimentally, the presence of a stress concentrator, residual stresses indicate the location of the defect under a particular surface.

These definitions are the results of experimental studies that characterize the stability of computer holograms. When conducting research, for example: elements of the investigated part (figure 6.3) with 25-fold repetition, the computer hologram recorded almost a similar pattern, which allowed to say that this part is made without defects, has parameters within acceptable limits and can be operated effectively for a specified period of time. The study was conducted using ploughshares, which are used in tillage in agricultural enterprises. An important feature of computer holograms is that they provide information about the distribution of microdeformations on the surface of the part being studied. It is possible to study the physical processes of microdeformation not discrete, point by point, but integrally, assessing the state of the entire surface. The number of component models of the general description depended on the required accuracy given in advance, with an error of not more than β = 0.1-0.3.

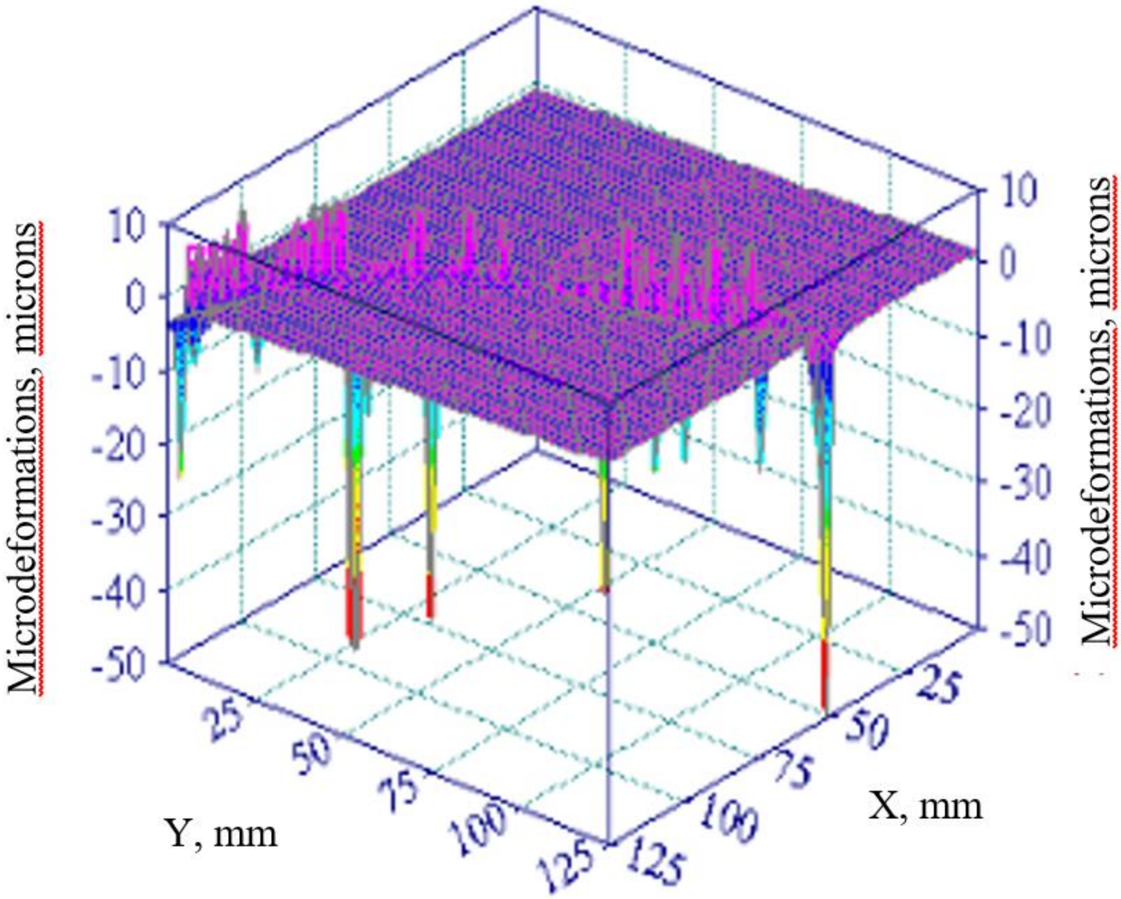

Fig. 6.3 Mathematical model of the process of microdeformation of the ploughshare surface at its allowable load.

The presence of a scale on each hologram allows to determine the magnitude of microdeformation at each point of the studied part, to develop a mathematical model that allows to adequately describe the process of microdeformation under the action determined by the formula, type and magnitude of the load. The mathematical model in the three-dimensional image, which adequately describes the dependence of the microdeformation at the allowable load, is shown in figure 3. This mathematical model is defined by a logarithmic polynomial:

$$M = f(X,Y) = (-1.0813 + 0.0104 \cdot X + 0.0255 \cdot Y - 0.6 \cdot X^2 - 0.0002 \cdot X \cdot Y) \times$$
$$\times (10 - 0.0097 \cdot X - 0.0081 \cdot Y + 1.139 \cdot X^2 - 0.5 \cdot Y^2)^{-1},$$

where M – microdeformations, microns, X, Y – coordinates on the surface of the hologram, mm.

Analysis of the correlation equation for a part with an existing defect shows that computer holography with a high degree of reliability describes the physical process, where: $G^2 = 0.877$ at $G^2 = 1$ => max; DF Adj $r^2 = 0.867$ at DF Adj $r^2 = 1$ => max. This indicates the adequacy of the result to the real process.

The hologram has a corresponding distortion related to the presence of the defect found. Characteristic peaks and troughs on the hologram characterize the presence, structure, and to some extent, the magnitude and nature of the subsurface defect: in our case, a crack, which significantly reduces the operational reliability of the working surface of the part.

Conclusions to Chapter 6

The important feature of computer holograms is that they provide information about the distribution of microdeformations on the surface of the part being studied. It is possible to study the physical processes of microdeformation not discrete, point by point, but integrally, assessing the state of the entire surface.

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Characteristic peaks and troughs on the hologram characterize the presence, structure, and to some extent, the magnitude and nature of the subsurface defect: in our case, a crack, which significantly reduces the operational reliability of the working surface of the part.

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CHAPTER 7. RESEARCH OF WEIGHT AND LINEAR WEAR FROM RESOURCE INDICATORS OF CULTIVATOR PAWS HARDENED BY COMBINED METHOD

Introduction

In the agricultural production [1], along with domestic tillage equipment [2], a significant amount of tillage equipment of foreign production has been accumulated [3], which is widely used [4]. The experience of using this technique indicates its advantages in reliability [5] and durability compared to domestic [6], in particular the operation of working bodies [7]. This is achieved by leading foreign companies through the use of new technologies in manufacturing [8]. It is established that in order to achieve an increased resource of working bodies [9], modern manufacturers use special alloying materials [10], design features [11] and technological methods of strengthening (heat treatment [12], application of wear-resistant materials [13]). The cost of such parts is quite high [14]. Therefore, there is a need to provide the agricultural consumer with relatively inexpensive spare parts for tillage equipment with high wear resistance [15].

The decision of this question is possible by creation of competitive technologies of strengthening of details of domestic and foreign agricultural machinery at their restoration [16] and manufacturing for increase in a resource of these details several times and introduction of the given developments at the enterprises [17] and at the repair enterprises of agricultural machinery [18]. Improving the performance of working bodies [19], in particular cultivator legs, is an urgent task in agricultural production [20]. This can be achieved by creating new technologies for manufacturing [21] and strengthening the working bodies [22], the use of new materials [23]. In particular, the development of technologies with the use of powder materials to strengthen the cutting edge is promising [24], which allows to increase the physical [25] and mechanical properties of the applied layers [26].

The most commonly used methods for strengthening the working surfaces of parts are conventional hardening of medium-carbon [27], high-carbon [28] and alloy steels [29]. The hardness of the metal can be obtained in the range of 45 HRC for steel 45 [30] and up to 65 HRC for steel 65G [31] and alloy steels [32]. The wear resistance of such working bodies is lower in comparison with other [33], strengthened special materials [34].To locally strengthen the surface of the part in places of probable wear [35], it is advisable to use other methods [36], namely [37], electric arc surfacing with artificial electrodes [38] or flux-cored wires [39], surfacing with powder materials on microwave units [40], plasma [41] and gas-flame surfacing with powders, rods [42].Electric arc surfacing and artificial electrodes and wires can be both continuous over the entire surface and point [43]. Studies have shown that spot surfacing has proven itself better on the working side, while slightly increasing the resistance, which sufficiently protects against wear of the base layer of metal [44].

Purpose of research

The purpose of the research is to increase the resource performance of cultivator paws when using a combined method of hardening.

Materials and methods

One of the ways to solve this problem is to increase the life of cultivator paws due to the combined three-layer formation of their working surfaces, namely (figure 7.1): 1 – base is steel 65 G; 2 – hardened layer of steel 65 G to a thickness of 1.9–2.3 mm using the electrocontact method of sharpening and hardening, thus obtaining a solid layer of wear resistance of which is 3.48 relative to unhardened steel 65 G; 3 – additional hardening performed on top of the hardened layer with powder abrasive-resistant materials, which are deposited with a special electrode T-590 Ø5×450 mm, the wear resistance of this layer is 4.49 relative to unhardened steel 65 G.

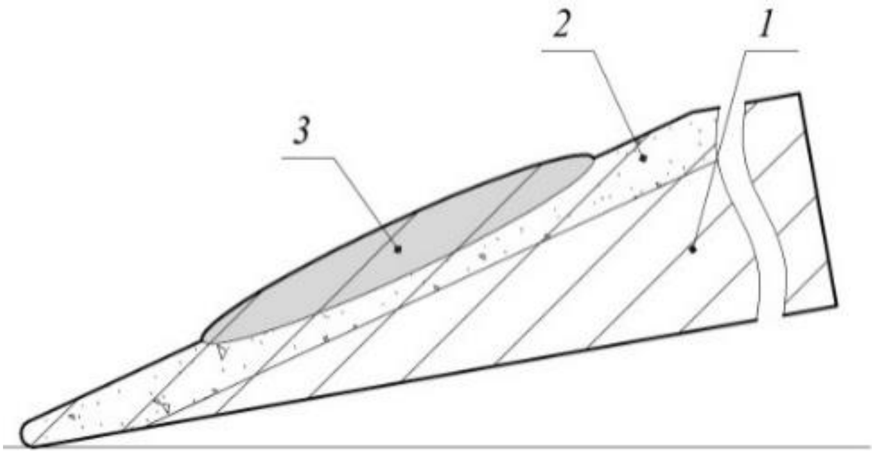

Fig. 7.1 Scheme of electrocontact sharpening and strengthening of the blade of the cultivator paw: 1 – the base is steel 65 G; 2 – zone of electrocontact sharpening and strengthening; 3 – zone of strengthening by powder abrasive-resistant materials.

As powder abrasive-resistant materials used powders PS-12NVK-01, PG-10K-01, sormite, FCB-1, PT-NA-01, PG-19M-01. According to the developed technology, a set of cultivator paws with a grip width of 420 mm was strengthened (figure 7.1). Manufactured and strengthened cultivator legs were installed on the cultivator QUANTUM-12 with a width of 12 m and were tested in production conditions (figure 7.2).

Fig. 7.2 Cultivator with the strengthened paws.

To determine the change in the geometric parameters of the blades of cultivator legs during production tests recorded linear wear, weight wear and the radius of rounding of the cutting edge of the working bodies with an operating time of 8, 23, 42 and 54 ha. According to the results of linear wear of the wings of experimental cultivator paws during production tests, the materials that provide the best performance against abrasive wear have been identified.

Results and discussion

As a result of the analysis of cultivator paws with a yield of 54 ha, it was found that the working bodies do not have visible damage and extreme wear and are suitable for further use. At the same time, measurements showed that the amount of wear on the width of the blade is 5.3–11.9 mm (figure 7.3).

Fig. 7.3 Reinforced cultivator legs.

It is established that the best results of wear resistance showed powder material PS-12NVK-01 applied by arc surfacing with an electrode T-590 Ø5×450 mm and is described by the following dependence:

$$w_x = 0.085 \cdot t_l - 0.059, \tag{7.1}$$

where w_x – the amount of linear wear of the blade of the cultivator paw, mm; t_l – operating time per paw of the cultivator, ha.

The results of comparative tests of weight wear of reinforced and serial legs showed that for the production of 54 ha of reinforced legs, the average weight wear is 425 g.

After processing the results of production tests of the legs of the cultivator QUANTUM-12 received mathematical models of the process of weight wear of serial w_c and hardened y from the legs (figure 7.5), depending on the operating time t_l:

$$w_{cs} = 0.017 \cdot t_l^{1.47} \text{ and } w_{ct} = 0.003 \cdot t_l^{1.8}, \tag{7.2}$$

where w_{cs} and w_{ct} – respectively, the weight of worn material serial and hardened cultivator legs, kg.

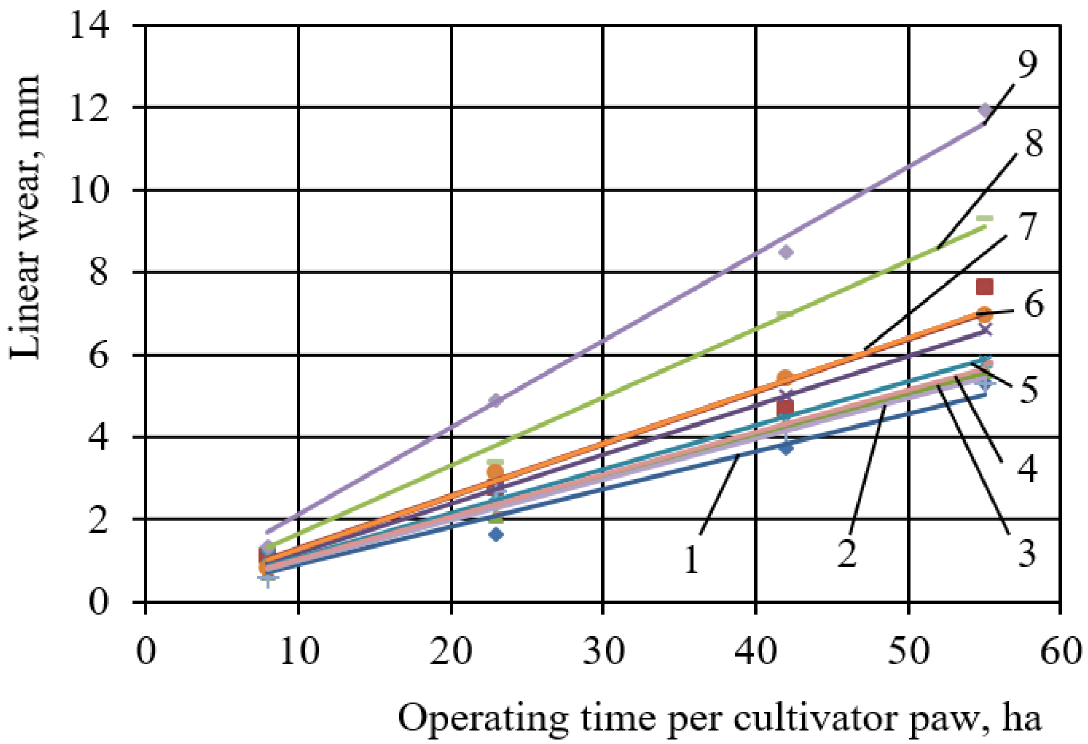

Fig. 7.4 Linear wear of the blade of the cultivator paw from the working time of the cultivator paw: 1 – PS-12NVK-01 + T-590; 2 – PG-10K-01 + T-590; 3 – FCB-1 + T-590; 4 – PG-10K-01 + Graphite; 5 – sormite + T-590; 6 – PS-12NVK-01 + graphite; 7 – FCB-1 + graphite; 8 – sormite + graphite; 9 – EDM treatment; 10 – serial cultivator paws KPE-410.

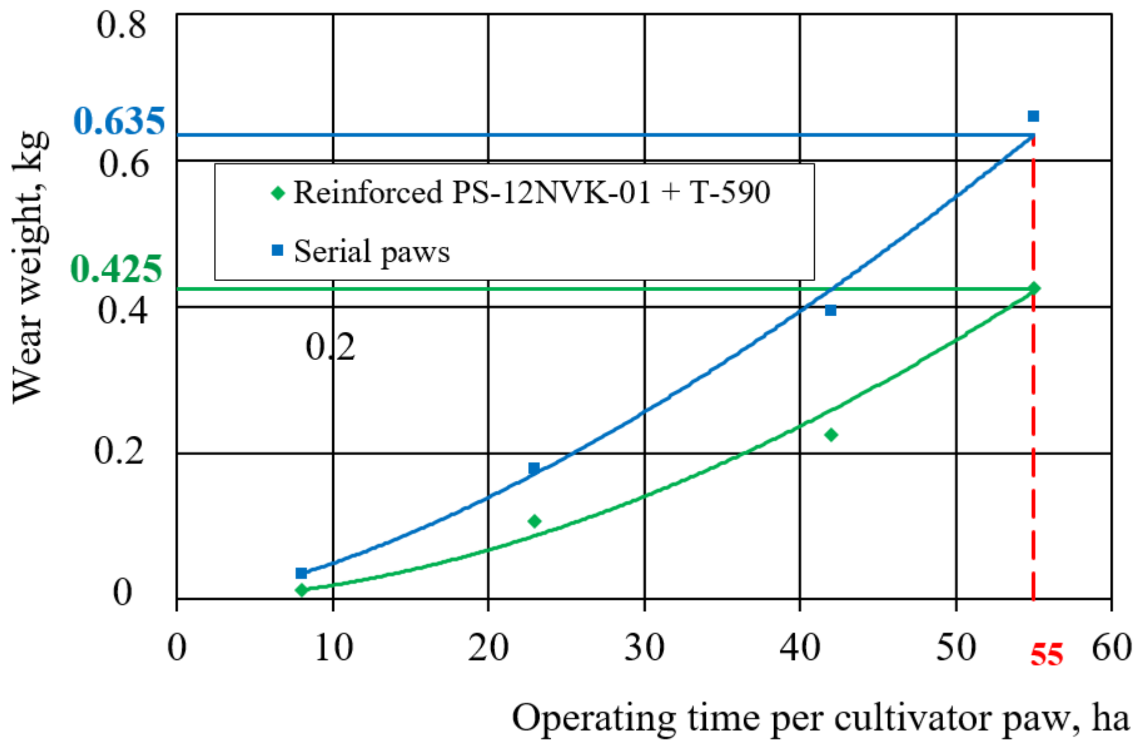

Fig. 7.5 Wear of cultivator legs by weight from operating time.

Mathematical models (2) allow to predict the amount of weight wear depending on the resource indicators of serial and harvested by the combined method of cultivator paws and found that when the weight wear reaches 635 g of hardened paws, their output will be 40% higher than serial.

The proposed method of hardening allows to increase the resource of cultivator paws by 35–45%.

Conclusions to Chapter 7

The best indicators of wear resistance showed powder material PS-12NVK-01 applied by the electrode T-590 after electrocontact treatment of the part, after 55 ha the amount of wear on the width of the blade is 5.3 mm, which is 43% less than electrocontact treatment.

The results of comparative tests of hardened and serial legs showed that for the production of 54 ha the average weight wear of serial legs is 635 g, and reinforced by the developed technology is 33% less and is 425 g. Achieving weight wear of 635 g of reinforced legs is will be 40% larger than serial.

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CHAPTER 8. RESEARCH OF PASSAGE CAPACITY OF COMBINE HARVESTERS DEPENDING ON AGROBIOLOGICAL STATE OF BREAD MASS

Introduction

The influence of the agrobiological state of the grain mass on productivity is stated in literary sources [1], and in practical conclusions it is declared by adjectives [2], but there is no analytical confirmation of the numerical values [3]. The implementation of the technological operation of harvesting the main condition is the quality of threshing of the grain mass and the stability of the passage of the technological process [4], through the implementation of the throughput indicator for combines of classes and design schemes [5]. The performance of combine harvesters depends on many objective and subjective factors and factors, the impact of which on the actual performance can be expressed by functional dependence:

$$W_f = W_{\text{A}} \cdot f(q_{\text{ф}}) = W_{\text{A}} \cdot f(k_1, k_2, k_3, k_4, k_{Bp}, k_{Vp}, k_A, k_{Ne}, k_q), \quad (8.1)$$

where: k_1, k_2, k_3, k_4 – coefficients that take into account the characteristics of the agrobiological state of the grain mass: straw moisture, straw content, grain moisture, dockage of grain [6]; $k_{Bp,}k_{Vp}$ – coefficients that take into account the average value of the header width and the average value of the working speed [7]; k_A, k_{Ne}, k_q – availability factors, the degree of reduction of engine power, from the nominal value and the coefficient of capacity reduction [8].

The development of technological maps for harvesting the values of the influence coefficients by specialists of agricultural enterprises are selected empirically [9], based on their own generalized experience or qualifications [10]. To reduce the influence of the subjective assessment of the influence of operating factors, characteristics, on the productivity of combines, the numerical values of their influence should be calculated [11]. Harvesting is characterized by certain quality indicators: technological standards (throughput) [12] and permissible deviations from them (technological tolerance for standards) [13]; the accuracy of the developed requirements [14] or the level of comparison of the quality indicators obtained in real production with the acceptable ones [15].

The uniformity of the grain mass feeding into the threshing apparatus depends on the influence of a significant number of factors [16]: uneven plant density, height and moisture of the crop, use of the header width, uneven mowing height, uneven feed by the header auger and floating inclined transporter, cultivation of a specific field, the degree of dockage of grain. In the literature [17], the throughput of the threshing-separating device of grain harvesters is shown as a constant value that depends on four starting design and operational characteristics and six empirical coefficients (0.458, 32, 0.26, 1.5, 0.8, 0.83). Practice shows [18] that in real production conditions, throughput and, accordingly, productivity is a variable value that depends on objective and subjective factors and characteristics. The objective factors are: soil and climatic conditions, relief and contours of fields, physical and mechanical properties of crops, design and operational characteristics of combines [19]. Subjective factors: dockage of grain, straw content, moisture content of the grain mass and grain, agricultural culture, qualification of combine operators (selection of the optimal working speed), cutting height, cutting width of header [20].

Purpose of research

The purpose of this work was to obtain the dependence of the actual throughput of the threshing-separating device and the performance of the combines on the characteristics of the grain mass during combining: the straw content of the grain mass, the moisture content of grain and straw, the degree of contamination of the field.

Materials and methods

The following characteristics are standard: straw content δ_c=1.5; yield capacity U=4 t/ha; straw moisture content M_s=17%; grain moisture M_g=15%; dockage of grain B_b<5%. The nominal hourly productivity with a yield of up to 4 t/ha can be determined using the relationship:

$$W_A = 0.36 \cdot \mathrm{B_b} \cdot \{Ne_{\text{н}} \cdot \xi - 2 \cdot q_{\text{н}}\} \cdot \left\{\mathrm{B_p} \cdot U \cdot (1 + \delta_c) \cdot (N_{\mathrm{M}} + N_{\mathrm{P}}) \cdot \eta_{\mathrm{T}} + 10 \cdot g \cdot f \cdot G_{\mathrm{T}} \cdot t\right\}^{-1}. \quad (8.2)$$

With a yield of more than 4 t/ha, taking into account the throughput of the threshing-separating device of the combine from the dependence:

$$W_A = 3.6 \cdot q_{\text{н}} \cdot \{U \cdot (1 + \delta_s)\}^{-1}. \quad (8.3)$$

To calculate the performance of the combine according to formulas (8.2), (8.3), it is necessary to determine the indicator of the throughput of the threshing-separating device, taking into account the influence of single agrobiological characteristics of the grain of the harvested crop. The throughput, in turn, is determined taking into account the numerical values of the individual coefficients. Obviously, the given characteristics of grain crops affect the increase in power consumption per unit of threshed grain mass. Experts in agricultural production know how the given characteristics of the grain stand, especially contamination, affects the change in the physical and functional parameters of the pitched board, the sieve mill, the inner surfaces of the straw walkers. The liquid that is squeezed out by the drum from the wet mass of weeds with a moisture content of 60-70% has high adhesion properties and falls on the working surface of the screen, sieves and the inner (working) surfaces of the straw walkers. This contributes to the adhesion of dust, chaff, chopped straw on them, and the creation on the surface (especially of the pitched board) of a monolithic hard surface, sometimes up to 50-70 mm thick. After which the pitched board loses its functional characteristics to separate grain from the chaff. To clean the pitched board from the adhesion of dirt, considerable physical effort and special technical devices are required, as well as an additional 4 hours of working time. The inhomogeneity of the thickness of the dirt around the perimeter of the pitched board is the cause of imbalance and possible breakage of the mountings. The accumulation of dirt on the working surfaces of the sieves, straw walkers leads to an increase in grain losses behind the harvesters of the threshing-separating device. In turn, weed residues have a greater mass, geometric dimensions and other aerodynamic properties than chaff and chopped straw and also causes increased grain losses. The influence of straw content, moisture and debris on the throughput of the threshing-separating device can be determined as follows: the influence of straw content (we use the inverse value of the coefficient):

$$qn(k_{1max})^{-1}, qn(k_{1min})^{-1}, \tag{8.4}$$

where k_{1max} and k_{1min} – respectively the maximum and minimum value of the coefficient of the influence of straw content on the throughput of the harvester of the threshing-separating device.

The influence of grain moisture (we use the inverse value of the coefficient):

$$qn(k_{2max})^{-1}, qn(k_{2min})^{-1}, \tag{8.5}$$

where k_{2max} and k_{2min} – respectively, the maximum and minimum value of the coefficient of the influence of grain moisture on the throughput of the threshing-separating device of the combine.

The influence of dockage of grain:

$$qn(k_{3max})^{-1}, qn(k_{3min})^{-1}, \tag{8.6}$$

where k_{3max} and k_{3min} – respectively, the maximum and minimum value of the coefficient of the influence of grain contamination on the throughput of the combine harvester of the threshing-separating device.

To obtain the resulting coefficient of influence of the state of grain, it should be taken into account that the direction of change in the value of the coefficient of influence must coincide with the direction of its influence on the throughput. That is, the minimum value of moisture, straw content and debris corresponds to the maximum value of the throughput of the threshing-separating device. Combining all the coefficients of influence of the state of grain, the throughput of the threshing-separating device can be expressed by the dependence $qnk_{3max}(k_{2max} \cdot k_{1max})^{-1}$, $qnk_{3min}(k_{2min} \cdot k_{1min})^{-1}$.

Results and discussion

To take into account the influence of straw content, the authors [7] proposes to use the coefficient k_1, the value of which is determined from the expression:

$$k_1 = (1 - c_0) \cdot (1 - c)^{-1} = (1 - 0.66) \cdot (1 - 0.64)^{-1} = 0.94, \qquad k_1 = 1 - 0.94 = 0.06 \approx 6\%.$$

where: c_0, c – calculated and actual grain content in straw, in unit fractions.

Machine test stations define this factor differently:

$$k_1 = \{0.6(1 + [\delta_c]^{-1})\}^{-1} = \{0.6(1 + +[1.5]^{-1})\}^{-1} = 1.$$

The calculated values of the coefficient, depending on the straw content, are shown in table 1. As can be seen from table 8.1, a change in straw content from δ_c = 0.9 increases the influence coefficient k_1 from 0.79 to 1.09, that is, by 30%. The influence of the moisture content of the grain mass on the value of the coefficient of influence k_2 for weed grain crops can be determined from the relationship:

$$k_2 = \left(\left\{100 - \mathrm{B}_g\right\} \cdot \left\{100 - \mathrm{B}_g^1\right\}^{-1} + \delta_c \cdot \{100 - \mathrm{B}_s\} \cdot \{100 - \mathrm{B}_s^1\}^{-1}\right) \cdot (1 + \delta_c)^{-1}, \quad (8.7)$$

where B_g, B_s, B_g^1,B_s^1 – respectively, the standard conditional moisture content of grain and straw and their actual value %.

Table 8.1

Influence of straw content on the value of the coefficient

δ_c	**0.9**	**1.0**	**1.1**	**1.2**	**1.3**	**1.4**	**1.5**	**1.6**	**1.7**	**1.8**	**1.9**
k_1	0.79	0.83	0.87	0.91	0.94	0.97	1.0	1.02	1.05	1.07	1.09
$(k_1)^{-1}$	1.266	1.208	1.199	1.093	1.063	1.030	1.0	0.98	0.952	0.934	0.917

For calculations, we will take a change in the moisture content of grain B_g from 20% to 13%, straw B_s from 22% to 15%. As evidenced by the data in table 8.2, a decrease in the moisture content of grain B_g from 20 to 13% and straw B_s from 22 to 15% reduces the influence coefficient k_2 from 1.063 to 0.976 or by ≈9%. In case of contamination of the grain mass, the value of the coefficient of influence k_3 is determined from the dependence:

$$k_3 = \left(100 - \mathrm{B}_g\right) \cdot \left(100 - \mathrm{B}_g^1\right)^{-1} \cdot (1 - \varepsilon)^2, \text{ or } k_3 = -0.025 \cdot \varepsilon^{0.538} + 1.037,$$

where ε – the content of weeds in the grain mass in unit fractions.

Table 8.2

Influence of grain and straw moisture on the value of the coefficient.

Indicators	Moisture content of grain and straw %							
Corn	20	19	18	17	16	15	14	13
Straw	22	21	20	19	18	17	16	15
k_2	1.063	1.048	1.037	1.024	1.012	1.000	0.988	0.976
$(k_2)^{-1}$	0.941	0.954	0.964	0.976	0.988	1.0	1.012	1.0245

The calculated values of the influence of contamination on the coefficient indicators are given in table 8.3 under standard conditions B_g= 15%, B_s = 17%. The work [7] shows a method for calculating the throughput of the threshing-separating device of grain harvesters using the example of combine harvesters Slavutich KZC-9F.

Table 8.3

Influence of grain mass contamination on the value of the coefficient.

Indicators	Relative contamination of the grain mass %									
	5	**10**	**15**	**20**	**25**	**30**	**35**	**40**	**45**	**50**
k_3	0.976	0.957	0.943	0.929	0.912	0.882	0.875	0.868	0.850	0.840

The calculated values of the influence of the coefficients on the performance indicators of the threshing-separating device are given in table 8.4. In particular, the data in the table indicate that the total effect of all coefficients on the throughput of the threshing-separating device can reach 43%. Relying on addictions stated above, the throughput of the threshing-separating device of combine harvesters Slavutich KZC-9F for example was determined under standard conditions: yield capacity U = 4 t/ha, grain crop – wheat, straw moisture B_s= 17%, dockage of grain – 0%, grain moisture B_g = 15%, grain: straw ratio – 1:1.5, engine power $Ne = 173$ kW, combine mass – 16.8 kN, transmission efficiency η_T = 0.88, rolling coefficient $f = 0.12$, specific threshing power $N_{\text{M}} = 9.1$ kW s/kg, $N_P = 2.1$ kW s/kg. As follows from the indicators in table 8.4 and figure 8.1, the straw content of grain stands has the greatest impact on reducing the throughput of the threshing-separating device.

Table 8.4

Calculated values of the coefficients of influence on the indicators of the throughput capacity of the threshing-separating device of combine harvesters Slavutich KZC-9F.

Characteristics of the grain	**Value**						
	Δ_{min}	Δ_{max}	k_{min}	k_{max}	q_{max}	q_{min}	%
Straw content	0.9	1.9	1.265	0.917	11.57	8.31	-27%
Humidity	15%	22%	1.026	0.94	9.38	8.60	-8.1%
Clogging	5%	50%	0.976	0.84	8.60	7.68	-16.1%
Total impact			1.265	0.724	11.57	6.62	-43%

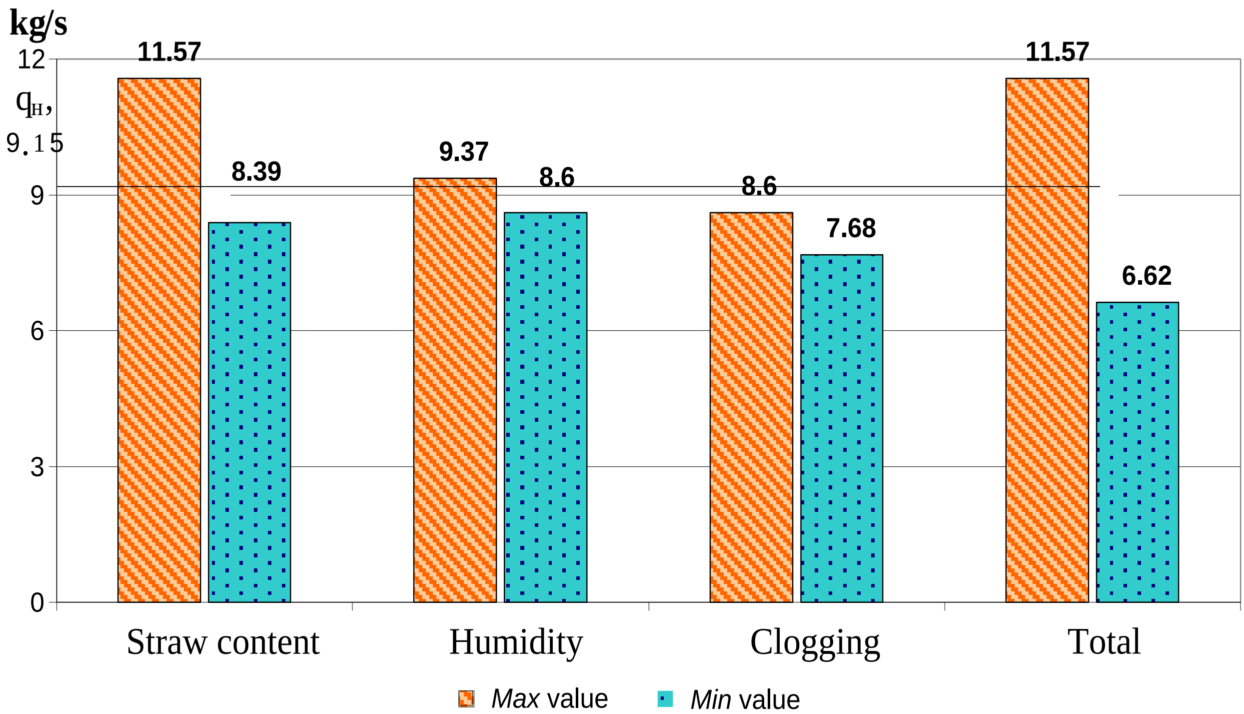

Fig. 8.1 Influence of individual and general characteristics of grain mass on throughput of harvester of the threshing-separating device.

The change in straw content $\delta_c = 0.9\ldots1.9$ accordingly leads to a change in the throughput from qn_{max} to qn_{min} kg/s.

In percentage terms, this influence is 27.4%. The influence of debris when changing from 5 to 50% reduces the throughput of the threshing-separating device by 16.1%.

The minimum impact on the throughput of the combine harvester is made by the moisture content of grain and straw and is 8.1%.

The calculation of the performance of combine harvesters Slavutich KZC-9F at the standard characteristics of grain according to the formula (8.2) showed the value $W_{\text{A}} = 4.05$ ha/h. A change in straw content from 0.9 to 1.9 affects productivity, changing it in inverse proportion from 5.12 to 3.71 ha/h, that is, it changes by 27.5%.

Humidity with a change from 22 to 15% affects productivity in the range from 4.15 to 3.8 ha/h or 8.4%, and a change in dockage of grain from 5 to 50% affects, respectively, in the range from 3.95 to 3.4 ha/h or 14%.

Conclusions to Chapter 8

The probable boundaries of the change in the numerical values of the throughput of the threshing-separating device of combine harvesters Slavutich KZC-9F are calculated, depending on the influence of single and total coefficients of influence. The greatest influence on the change in the throughput of the threshing-separating device is the straw content of the grain stand (within 27%), the lowest moisture content of the straw mass (within 8%).

Contamination significantly affects the throughput of the threshing-separating device and an increase in mechanical losses for the threshing-separating device (5-6 times) in comparison with the standard value (1%) of the gross tax.

Changing the cutting height of grain by 1-1.5 cm increases the throughput of the threshing-separating device of combine harvesters Slavutich KZC-9F by %, or by 0.1-0.12 kg/s. The productivity, respectively, by 0.04-0.05 ha/1%. The increased dockage of grain by 5% reduces the throughput of the threshing-separating device by 0.147 kg/s, and the productivity by 0.0065 ha/5%.

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CHAPTER 9. EXPERIMENTAL EVALUATION OF ENERGY PARAMETERS OF VOLUMETRIC VIBROSEPARATION OF BULK FEED FROM GRAIN

Introduction

The most complete theoretical studies of the operation of sieve surfaces that implement longitudinal oscillating motions are performed in the fundamental works [1, 2]. The first recorded a system of differential equations of spatial motion of the operating device of a vibrating machine with several mechanical vibrators, the axes of which are arbitrarily oriented in space [3, 4].

He solved the problem of moving agricultural materials on the working surfaces of the separators, provided that such materials are discrete solids; solved tasks of layer-by-layer processes and self-sorting, clogging and cleaning of sieve openings, sifting of seeds through sieve openings.

In the works [5, 6] the dependences of the quality of separation of a flat sieve on the main parameters are obtained: kinematic mode of operation and angle of inclination of the sieve to the horizon, angle of oscillation, shape and location of holes, sieve sizes and specific loading, humidity and clogging of grain material [7].

Kalivoda J. [8] determined the value of the critical velocity of sifting particles through the holes of the sieve, at an angle of inclination of the latter not more than 10° in the direction of particle rise, taking into account air resistance.

Golovin A. [9] found that biharmonic oscillations of the sieve are one of the effective ways to improve the technological process of grain separation.

He described the movement of grain material on the sieve, which performs biharmonic oscillations, developed a method for calculating the parameters of the eccentric drive mechanism of the sieve with an elliptical pulley, which obtained the biharmonic law of its oscillations, proposed and substantiated the separator scheme with biharmonic flat sieve drive.

Bahadirov G. [10] developed methods for calculating the intensification of vibrocentric separation processes according to technological indicators of productivity and quality, created and introduced into serial production dynamic, surface and volume intensifiers that increase productivity and quality of grain material separation by vibrocentric separators.

Astanakulov K. [11] found that the quality of the process of separation of grain raw materials on the non-perforated friction oscillating surface significantly depends on the structural and kinematic parameters of the vibrating grain cleaning machine: amplitude, frequency and angle of oscillation, as well as angles of inclination of the operating device to the horizon in transverse directions as a result, he substantiated the rational design and kinematic parameters and the actual working devices of the vibrating grain cleaning machine for processing wheat, barley, oats and rye.

Bulgakov V. proposed to intensify the process of separation of grain material at the sieve state due to the different angle of installation of the upper and lower sieves relative to the direction of the exciting force [12].

Elfverson C. [13] and Petre I. Miu [14] claim that the vibrating sieve separators of the traditional scheme of separation of grain mixtures by sieves have practically reached the limit of improvement and further increase of productivity is realized by increasing the working area of sieves. They investigated the influence of kinematic, technological and structural parameters on interlayer processes, the speed of particle movement in the grain layer and along the sieve surface in the presence of passive leavening agents.

Aim of the research

The aim of the research is to experimentally evaluate the influence of geometric parameters of the driving mechanism of vibrating screen of bulk combined fodder from masses of crushed grain raw materials under the conditions of spatial oscillating motion of sieve surfaces on energy consumption to create and maintain the processing.

Materials and methods

To achieve this aim, the following tasks were provided: development of basic schemes of vibrating screen and drive mechanism for providing its working devices with spatial oscillations; according to the presented schemes development of experimental installation of a vibrating screen for separation of bulk combined fodder from the crushed grain components; obtaining graphical dependences that reflect the influence of design parameters of the developed vibrating screen on the change of energy consumption on the implementation of the studied separation process.

During the experimental analysis and substantiation of energy regime characteristics of the developed vibrating screen (figure 9.1) with a sieve surface that performs spatial oscillations used German Robotron equipment and special electrical devices to determine energy and amplitude-frequency characteristics, methods of mathematical analysis and their processing in mathematical MathCAD environment to obtain the necessary graphical dependencies. When adjusting the parameters of the volumetric vibrations of the sieve and the intensity of the oscillating action in a given plane, the angle of inclination of the operating devices of the screen was changed, which was carried out with the help of a specially designed sleeve.

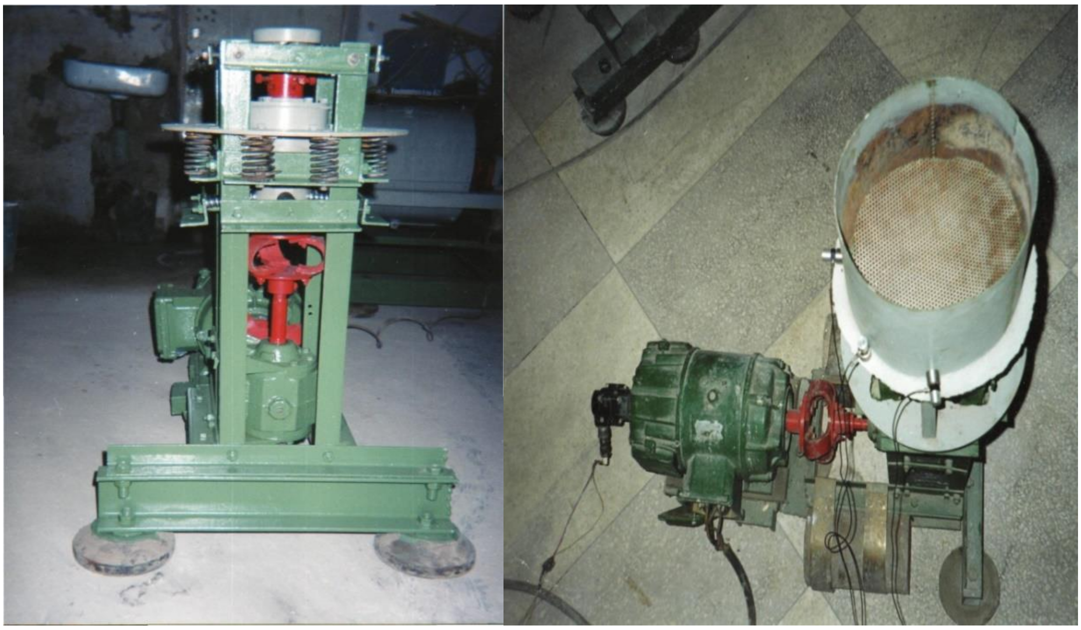

Fig. 9.1 Photos of developed experimental installation of vibrating screen and the drive mechanism of volume oscillations

The structure of the bulk material of feed consisted of crushed mass of wheat (45%), corn (30%), sunflower meal (8%), soybean meal (12%), premix (5%), which were completed on 5, 10, 15 and 20 kg. In the process of vibroseparation it was acceptable: the size of loose feed up to 3 mm, and the residue on the sieve did not exceed 10%.

The developed vibrating screen consists of a frame 9 (figure 9.2), on the base plate of which a vibrating platform 6 is mounted in a circle through cylindrical springs 7, on which a sieve is mounted, which can have one or several conical or cylindrical sieves 4.

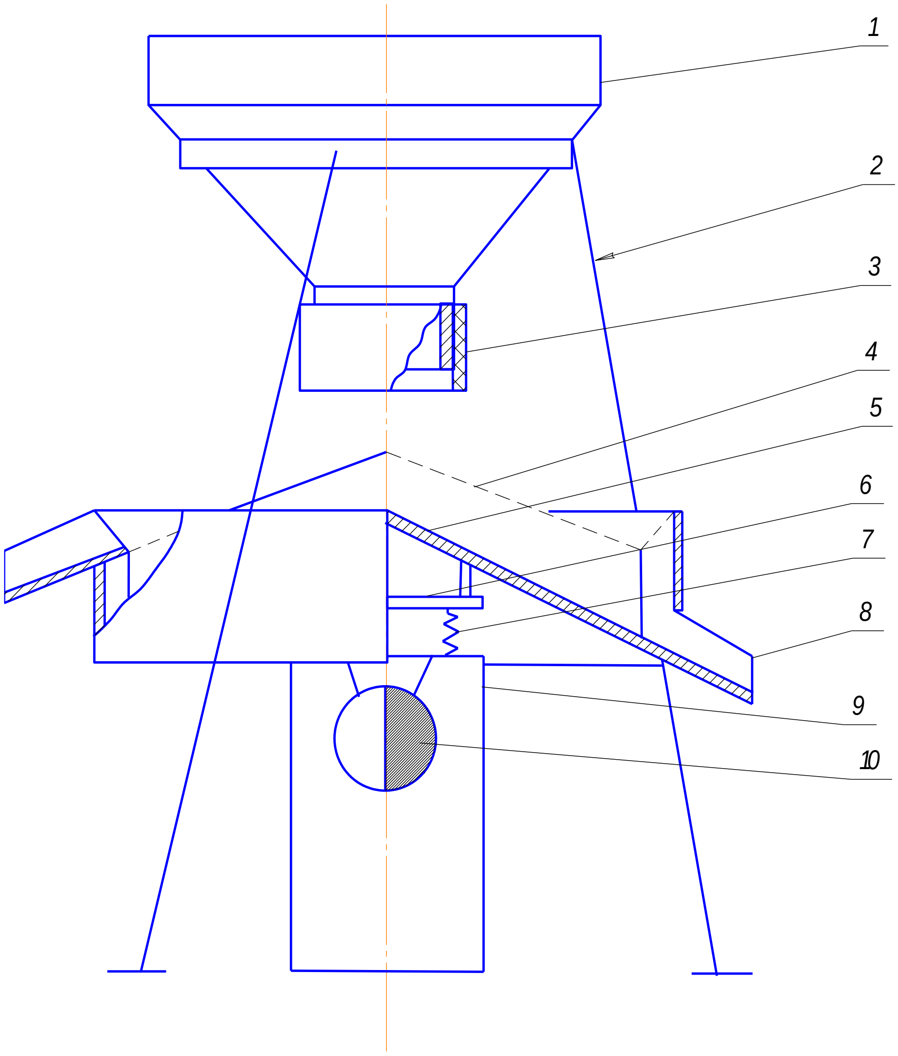

Fig. 9.2 Scheme of vibrating screen with a conical sieve surface: 1 – bunker; 2 – support; 3 – damper; 4 – sieve; 5 – sloping cone; 6 – vibroplatform; 7 – elastic element; 8 – launder; 9 – machine bed; 10 – vibrating exciter

To remove the separated fractions from the machine there is a sloping cone 5 (one or more depending on the number of sieves) and launders 8. In the lower part of the apparatus single-shaft vibrator 10 is attached to the vibrating table, which is driven by a DC motor. In the upper part of the machine, the hopper 1 is mounted on supports 2, which has an adjusting damper 3 for regulating the supply of products to the working space of the machine.

The vibrating exciter of this screen (figure 9.3) contains a drive of the vertical shaft 3, consisting of an electric motor 5, a conical reductor 6 and a sleeve 9, which is mounted on the shaft and kinematically connected to the separator through bearings 8. Placement of the sleeve on the shaft with eccentricity, and also execution of its outer surface in the form of a cylinder, the axis of which is with the axis of the shaft 3 – an acute angle φ, providing spatial oscillations of the working bodies of the machine.

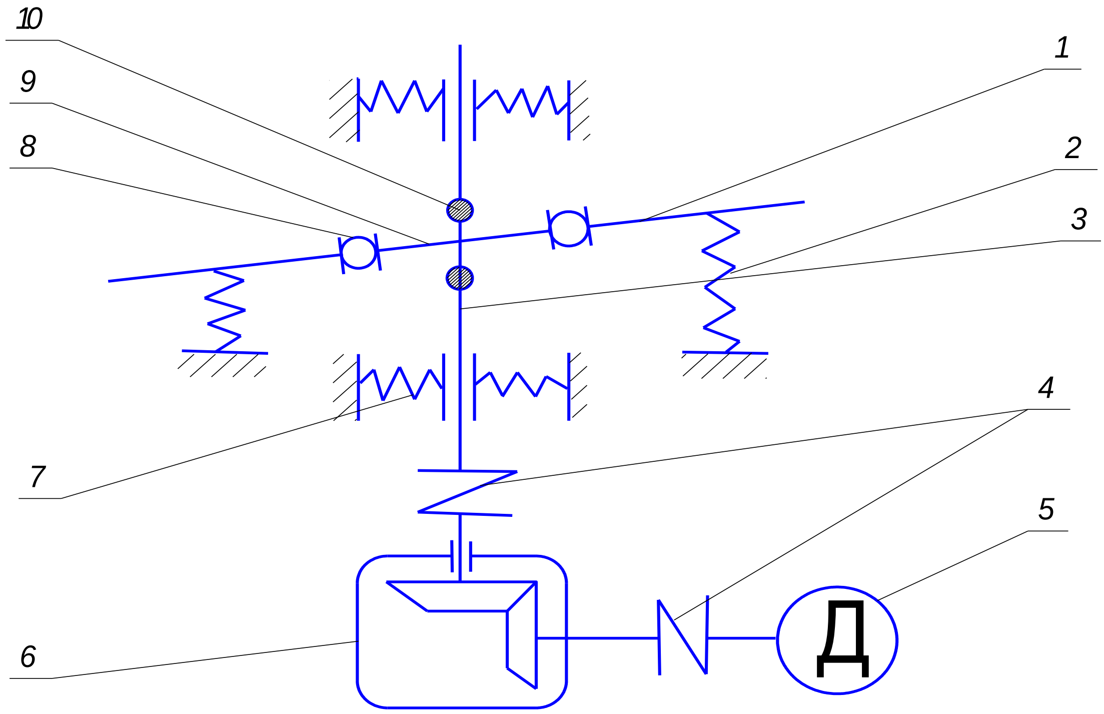

Fig. 9.3 Scheme of the vibrating drive of the screen: 1 – vibroplatform; 2, 7 – elastic elements; 3 – shaft; 4 – coupling; 5 – electric motor; 6 – conical reductor; 8 – bearing assembly; 9 – sleeve; 10 – counterweight

The drive of the shaft 3 (figure 9.3) from the motor 5 is carried out through the conical reductor 6. The inner rings of the bearings 8 are pressed on the sleeve 9, which is located on the shaft, so they rotate with it. The outer rings of the bearings 8, and with them the vibrating platform 1 and the separator itself does not rotate, but only moves in the radial and vertical directions.

The peculiarity of this vibratory drive is that in the horizontal plane the outer bearing races 8 and the separator perform a gyration movement, i.e. a gradual circular motion with a radius equal to the eccentricity of the inner surface of the sleeve relative to the axis of rotation of the shaft 3.

The springs 2, which are fixed at one end in a stated support, limit the rotation of the separator relative to its own axis. In addition, since the axis of the outer surface of the sleeve is inclined relative to the axis of the shaft at an angle φ, the outer race of the bearing and the separator perform oscillating motion in the vertical plane, which allows you to adjust the oscillation amplitude of the machine in a wide range.

In addition, the installation of the shaft 3 on the frame by means of bearings with adjustable elastic stops 6 allows to reduce the dynamic loads on all bearings of the drive and increase their service life.

Results and discussion

According to the results of experimental studies, the following energy dependences were obtained (figure 9.4 – figure 9.7). The increasing the load weight of the working container from 5 to 20 kg, the drive power of the machine changes by 14% (figure 9.4), with more stable operation of the drive motor is observed at higher loads of raw materials.

Figure 9.5 shows the graphical dependences of the power N_i on the angular velocity ω with a compatible change in the angle of the axis of the drive shaft φ and the value of its eccentricity e.

Changing the angle of the drive shaft axis φ from 0 leads to spatial oscillations in the studied system, which improves the separation conditions and with increasing energy consumption for the process decreases.

As the value of the eccentricity e of the drive shaft of the vibrator increases, the amplitude of oscillations increases and, accordingly, the permeability of vibrations in the mass of the bulk medium increases.

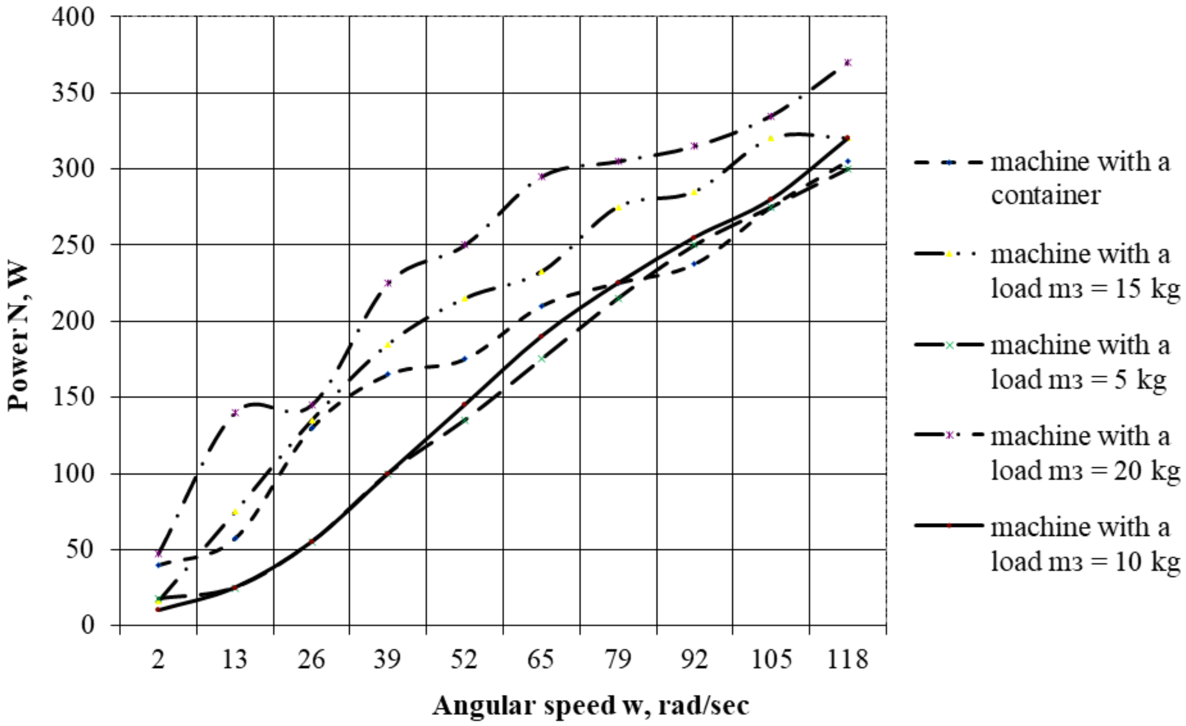

Fig. 9.4 Graph of the dependence of power consumption N on the angular speed of the drive shaft of the screen w when changing the load mass of the working tank

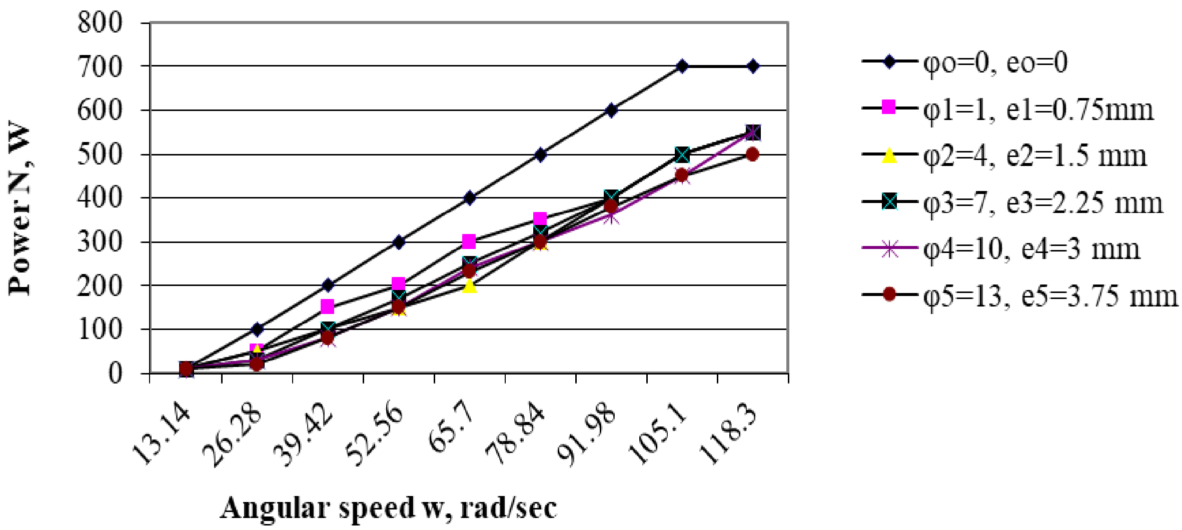

Fig. 9.5 Graph of the dependence of power N and the angular speed w when changing the angle of the axis of the drive shaft φ and the value of its eccentricity e

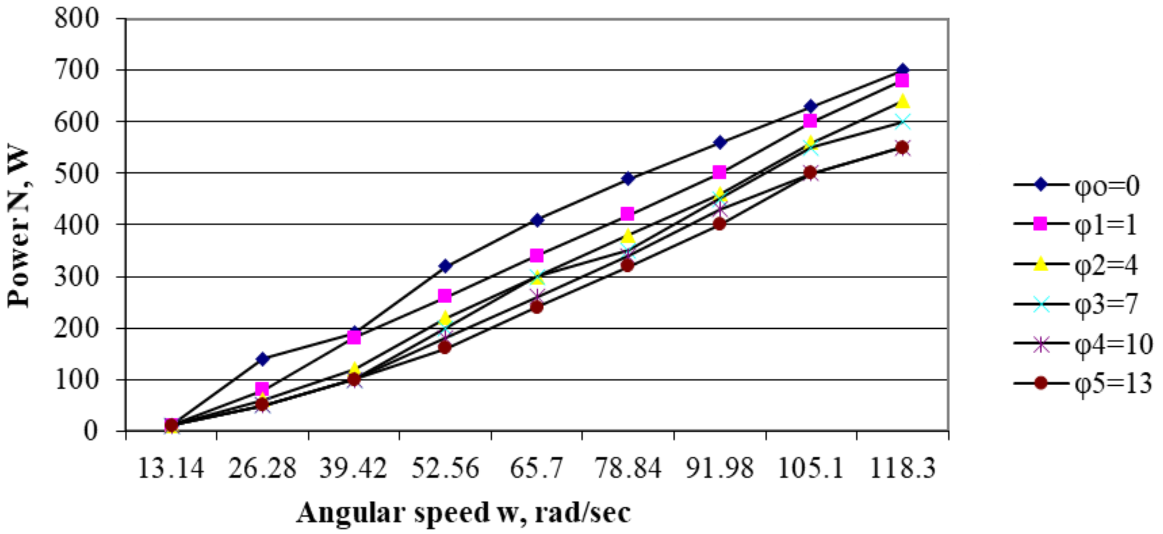

Fig. 9.6 Graph of power N and the angular velocity of the drive shaft of the screen w when changing the angle of the shaft axis φ

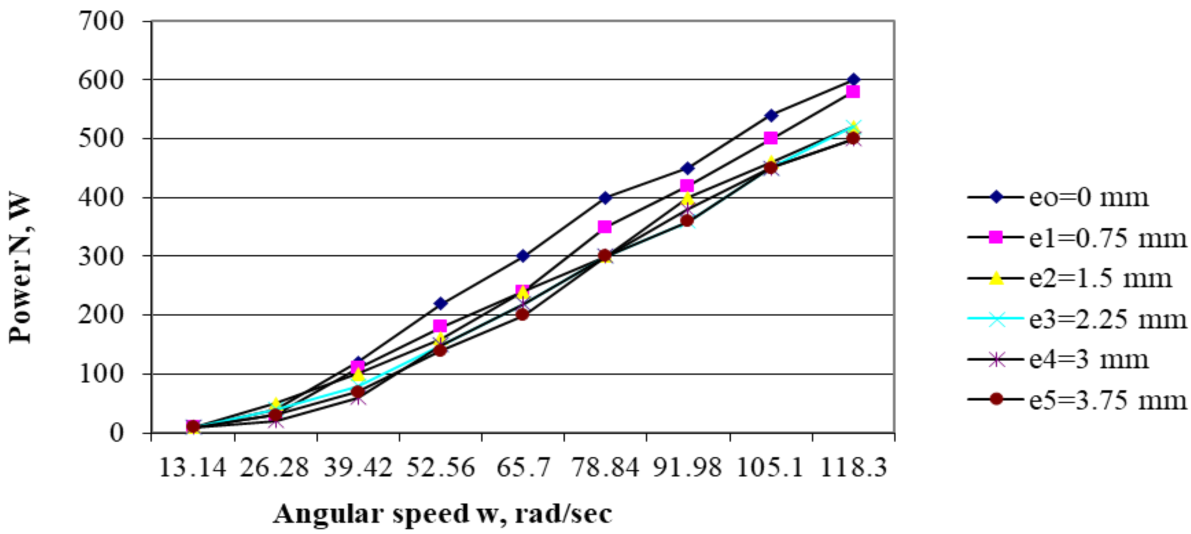

Fig. 9.7 Graph of the dependence of power N and the angular velocity w when changing the value of the eccentricity of the drive shaft of the screen e

Figure 9.6 shows the graphical dependences of the power N_i on the angular velocity ω only when changing the angle of the drive shaft axis φ, which confirms the effect of reducing energy consumption to create spatial motion of the working bodies of the machine. Figure 9.7 shows the graphical dependences of the power N_i on the angular velocity ω only when the value of its eccentricity e changes from 0, which leads to the creation of kinematic vibration excitation, which reduces the oscillating masses in the system and reduces energy consumption for the process.

At the value of the angle of the axis of the drive shaft of the screen $\varphi = 0^0$ and the value of its eccentricity e = 0 mm, there is a transfer of the container of flat oscillations, which increases energy consumption by 28.6% compared to spatial oscillations (figure 9.5). The rest presented in figure 9.5 operating parameters of the vibrating screen correspond to the transmission of spatial oscillations, in which energy consumption is reduced by 24%. When changing the angle of the drive shaft axis φ by 130, energy consumption for the separation process changes by 21.4% (figure 9.6), and when changing the value of the eccentricity of the drive shaft of the screen e by 3.75 mm energy consumption decreases by 16.7% (figure 9.7).

One of the progressive directions in the use of low-frequency mechanical oscillations for the separation of bulk agricultural products is the creation of devices with operating devices of spatial oscillatory action, the intensity of which depends not only on the amplitude and frequency of their source, but also on the method of excitation, transmission of oscillations and installation geometry. The developed experimental setup of the vibrating screen allowed to model the regimes of separation of bulk feed during the transmission of flat and spatial oscillations, changing the eccentricity of the drive shaft and the amplitude of oscillations, varying the angle of the working screen and the geometry of oscillating motions.

Conclusions to Chapter 9

1. Energy consumption during separation in the conditions of flat oscillations increases by 28.6% in comparison with the modes of realization of the spatial motion of sieve surfaces.

2. Energy consumption for the separation process changes more significantly when changing the geometry of oscillatory motions by 21.4%, than when changing the amplitude of oscillations by 16.7%. With increasing load weight of the working container, the operation of the drive mechanism is more stable, although there is an increase in energy consumption by 14%.

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CHAPTER 10. DESIGN OF LANDING OF ASSEMBLY MACHINE BUILDING UNITS WITH CIRCULATING LOAD ROLLING BEARING RINGS

Introduction

In today's economy [1], the quality of training of graduates of technical and agricultural higher education institutions are determined mainly by the degree of formation of future specialist's skills and ability to produce competitive products [2]. Therefore, the higher school should integrate into its educational standards [3], curricula and the structure of the main disciplines [4], the aggregate requirements of production and business [5], and preferably taking into account the peculiarities of staffing of enterprises in the region [6]. In the process of manufacturing any product of all types of machine production, ultimately [7], the problem of ensuring its assembly with the provision of the appropriate type of geometric interchangeability [8]. Typical technological processes of assembling machine-building products [9] and the main approaches to ensuring manufacturability [10] are presented in the literature, for example, in [11], and taking into account the peculiarities of heavy engineering also in [12], where, however, has no recommendations for training specialists in this field [13].

In the process of manufacturing any product for all types of production [14], the problem of ensuring the manufacturability of assembly with the provision of the appropriate type of interchangeability is solved [15]. In the special technical literature is widely used the term machinability of the part, which means the ability of materials to be processed by cutting or, alternatively, a set of properties of materials that ensure their processing by cutting to achieve optimal values of basic technological indicators (cutting speed, surface quality, cutting force, etc.) [16].

The problem of ensuring the manufacturability of the assembly is complex and includes, above all, the question of determining the optimal tolerances, limit deviations and landings in the design of the product with justification for choosing the type of interchangeability (complete, incomplete, group, fit or adjustable) [17]. The design of design and technological documentation is associated with the choice of the required accuracy of surfaces and their roughness [18], justification of landings taking into account the technological features of processing [19] and, in particular, with reasonable rationing of deviations in the shape and location of surfaces [20].

Researchers need to take into account current standards for the basic norms of interchangeability, as this primarily forms the quality of machine-building products and their competitiveness [21]. Therefore, it is very important that the scientific literature reflects modern ideas and methods of accuracy rationing [22], the purpose of landings in the joints, including, for example, in complex units with standard rolling pads [23], which are most common in engineering products, regardless of their series [24].

Aim of the research

The aim of the research is to study the general factors that ensure the accuracy of folding units with rolling mills with the achievement of the desired nature of the connection of the rings with the surfaces of the shaft and the hole.

Materials and methods

Despite the fact that the assembly of units with standard rolling mills (general specifications) are the most common in machine-building products, regardless of their series, the process of connecting them with parts is insufficiently studied – comprehensive studies have been conducted only on radial two-row roller rollers type 3182100 with a conical hole, when installing which the radial clearance in the roller is regulated by axial movement of the inner ring relative to the conical neck of the spindle [25].

The problem of ensuring folding is complex and includes, first of all, the question of assigning optimal tolerances, maximum deviations and landings in the design. Landing of rolling mills on the shaft and in the body should be selected taking into account the type and size of the rolling mill, its operating conditions, value and nature of loads acting on it, but, above all, the type of ring load: local, circulating or oscillating. According to the most common guide among practitioners [26], as well as all textbooks and manuals without exception, the landing of the circulating load ring is determined by the so-called intensity of the radial load by the formula:

$$P_R = R \cdot k_D \cdot k_1 \cdot k_2 \cdot (B - 2 \cdot r)^{-1}, \quad (10.1)$$

where P_R – intensity of radial loading, kN/m; R – constant radial load in the direction, kN; k_D – dynamic landing coefficient depending on the load (for overload up to 150%, moderate shocks and vibrations $k_D = 1$, for overload up to 300%, strong shocks and vibrations $k_D = 1.8$); k_1 – coefficient that takes into account the degree of relaxation of the landing tension for the hollow shaft and the thin-walled body (for the hollow shaft $k_1 = 1 - 3$, solid $k_1 = 1$, for the body $k_1 = 1 - 1.8$); k_2 – coefficient of non-uniformity of load distribution R between rows of rollers in two-row conical roller rollers or between double ball rollers in the presence of axial load on the support ($k_2 = 1 - 2$; in the absence of axial load $k_2 = 1$); B – bearing width, m; r – radius of curvature of the chamfer of the ring, m.

To select locally loaded rings in the mentioned sources the recommended fields of tolerances of openings and shafts depending on type of a rolling mill, working conditions and character of loading are resulted. If the dynamic coefficient (k_D) is difficult to determine, the fit can be determined by the minimum tension between the circulating ring and the surface of the part. Determine the smallest tension of the circulation-loaded ring by the formula:

$$N_{min} = 13 \cdot R \cdot k \cdot (B - 2 \cdot r)^{-1} \cdot 10^{-6}, \quad (10.2)$$

where N_{min} – the lowest design tension that provides the required strength of the connection of the circulating load ring with the shaft, mm; k – design factor, which depends on the series of felling ($k == 3.5$ for especially light series; $k = 2.8$ for light series; $k = 2.3$ for medium series; $k = 2$ for heavy series).

Choose the required standard fit that meets the condition:

$$N_{min\_s} \geq N_{min}, \quad (10.3)$$

where N_{min_s} – the lowest tension of a standard landing.

We check correctness of a choice of landing, proceeding from a condition of durability, for this purpose we define admissible tension:

$$N_{max} = 11.4 \cdot d \cdot k \cdot \sigma_p \cdot (2 \cdot k - 2)^{-1} \cdot 10^{-6}, \quad (10.4)$$

where N_{max} – the greatest design tension providing necessary durability of connection of a circulating loaded ring of a rolling pin with a shaft, mm; d – nominal diameter of the connected ring of a rolling mill, mm; σ_p – permissible tensile stress (for steel rolls $\sigma_p = 400$ MPa).

Check the strength of the connection, following the condition:

$$N_{max\_s} \leq N_{max}, \quad (10.5)$$

where N_{max_s} – the greatest tension of a standard landing.

Results and discussion

Determine the landing of the circulating loaded inner ring of a radial single-row bearing № 205 of accuracy class 6 ($d = 25$ mm, $D = 52$ mm, $B = 15$ mm,) on a rotating solid shaft. Estimated radial reaction of the support $r = 1.5$ mm, $R = 3$ kN. Impact load, overload 200%. There is no axial load. Determine the load intensity by formula (10.1):

$$P_R = 3 \cdot 1.8 \cdot 1 \cdot 1 \cdot (15 - 2 \cdot 1.5)^{-1} \cdot 10^3 = 450\ kN/m.$$

Here we take the values of the coefficients under the specified operating conditions of the bearing: $k_D = 1.8$, $k_1 = 1$, $k_2 = 1$. Under the specified conditions for the shaft corresponds to the tolerance field $k6$, ie $\varnothing 25k6\binom{+0.015}{+0.002}$. The maximum deviations of the diameter of the hole of the inner ring of the bearing are found by: $EI = -0.01$ mm, $ES = 0$. Then the landing of the inner ring of the bearing on the shaft $\varnothing 25\frac{L0}{k6}$. Limit tensions: $N_{min} = ei - ES = 0.002 - 0 = 0.002$ mm, $N_{max} = es - EI == 0.015 - (-0.01) = 0.025$ mm. The tolerance field of the hole in the housing under the outer locally loaded bearing ring is assigned a hole diameter of $D = 52$ mm for the specified conditions of the rolling mill, we accept the tolerance field $JS7$ (for the bearing of the sixth accuracy class), i.e. $Ø52JS7(\pm0.015)$. The maximum deviations of the outer diameter of the outer ring of the bearing are: $ei = -0.013$ mm, $es = 0$. Then fit the outer bearing ring in the housing $Ø52\frac{JS7}{l0}$. Limit tensions and gaps: $S_{max} = ES - -ei = 0.015 - (-0.013) = 0.0028$ mm, $N_{max} = es - EI = 0 - (-0.015) = 0.015$ mm. By formula (10.2) we find:

$$N_{min} = 13 \cdot 3 \cdot 2.8 \cdot (15 - 2 \cdot 1.5)^{-1} \cdot 10^{-3} = 0.0091 \text{ mm}.$$

Tolerance field of the inner ring of the bearing $Ø25m6\binom{+0.025}{+0.009}$. Then fit the inner bearing ring on the shaft $Ø25\frac{L0}{m6}$. Limit tensions: $N_{min} = ei - ES = 0.009 - 0 = 0.009$ mm, $N_{max} = es - EI = 0.025 - (-0.01) = 0.035$ mm. Condition (10.3) is fulfilled. By formula (10.4) we find:

$$N_{max} = 11.4 \cdot 2.8 \cdot 25 \cdot 400 \cdot (2 \cdot 2.8 - 2)^{-1} \cdot 10^{-6} = 0.1064 \text{ mm}.$$

Condition (10.5) is fulfilled. Determine the landing of the single-row tapered roller bearing №7209 on the shaft. The most loaded is the bearing of the right support. The load on the inner ring is circulating. Radial load 4788 N. The expected difference between the temperature of the furnace and the ambient air 20 °C. The sizes of the specified rolling mill: $d = 45$ mm, $D = 85$ mm, $B = 19$ mm, $r = 2$ mm, $r_1 = 0.8$ mm. Minimum allowable tension according to formula (10.1): $N_{min} = 0.0124$ mm. For clarity of the choice of landing we will make the table 10.1.

Table 10.1

The value of probable gaps and tensions.

Deviation of an opening of an internal ring, μm	Shaft, μm		Probable, μm			
	tolerance field	*es/ei*	clearance		tension	
			min	*max*	*min*	*max*
	js6	+8/-8	-	1.2	-	18.8
0	*k6*	+18/+2	-	-	8.8	28.8
-12	*m6*	+25/+9	-	-	15.8	35.8
	n6	+33/+17	-	-	23.8	43.8

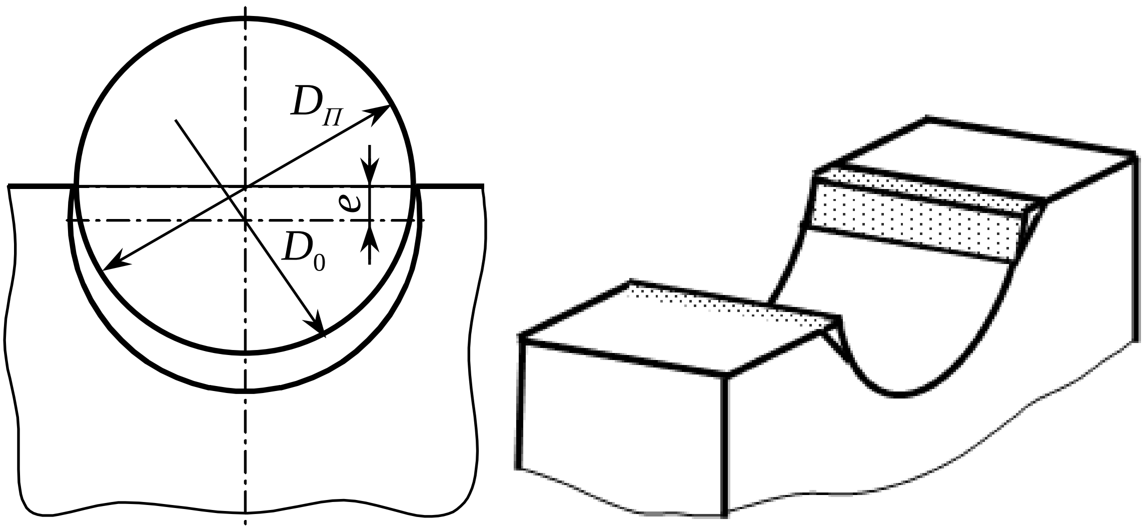
From table 10.1 it follows that the closest tolerance field of the shaft, which provides the connection with the inner ring of the rolling mill the required tension, is *m6*. Estimated durability under the given working conditions makes 9500 hours. Therefore, the bearing operation mode is normal. The inner ring of the rolling pin has a circulating load mode.

Instead, not only the methodological basis for the purpose of planting ring rings, but also detailed tables for their selection, taking into account, above all, the mode of its operation (depending on the ratio of the current radial load and dynamic load capacity), the type of load, type and diameter and even with numerous examples of machines and folding units. By the way, with reference to this standard in [7] is a compiled table with a simple list of recommended fields of tolerances and landings of rings of different types of furnaces depending only on the type of load, but without examples of sound selection of landings. A comparative analysis of both methods of assigning the considered plantings or tolerance fields shows the following differences. In the tables for the selection of plantings of rings with circulating load according to the method [18] there are no fields of tolerances of shafts, namely p6, r6, r7, recommended for many classes of machines and units operating in difficult conditions and tolerance fields with the main deviation h, provided by the standard for precision machines (hydraulic motors, small electric machines, in-pour spindles, etc.) and rolling mills on fixing sleeves.

Locally loaded rings usually require landings with a gap or transients with a higher probability of a gap – such a landing under the action of starting torque, shocks and vibrations from time to time scrolls relative to the connected surface, thus ensuring uniform operation of the raceway and the possibility of axial movement way of temperature deformations. To select the landings of such rings, in contrast to the materials in [11], specific landings are given, of course, taking into account the required accuracy class of the feller and the mode of operation of the machine. But the disadvantage of the method of selection according to [19] and other above sources is that they do not take into account the peculiarities of the manufacture of folding units with detachable housings.

According to the experience of mechanical engineering, the nominal (estimated) durability of joints with rolling mills in real conditions can be greatly reduced due to deformation of rolling pin rings, insufficient area of their adhesion to surfaces (less than 70…75 %) due to unreasonable technical requirements for connection accuracy. and surfaces of the connected details, and also deformations of both parts of the case after processing of planes of the socket and apertures (here after their preliminary assembly). The latter is due to technological heredity associated with deformations that occur during machining of parts, especially non-rigid holes (which are body parts), due to the redistribution of internal residual stresses in the metal thickness.

To minimize this phenomenon in the manufacture and assembly of detachable housings perform a number of measures aimed at ensuring the quality of the folding units under consideration. For example, the displacement e of the axis of the hole relative to the plane of the connector is limited by tolerances (figure 10.1, a), and before installing large shafts in semi-holes fit its landing surfaces in areas adjacent to the plane of the connector, performing the so-called collapse, the dimensions of which are regulated by a special normative document depending on the dimensions of the hole (figure 10.1, b).



a) b)

Fig. 10.1 Scheme of installation of the outer ring of the rolling pin in the semi-opening of the housing (a) and fitting the surfaces (b).

It is easy to see from figure 10.1 and that the condition of assembly of the outer ring of the rolling mill with a semi-finished product can be expressed by the condition:

$$2 \cdot \sqrt{4^{-1} \cdot D_0^2 - e^2} \geq D_п, \tag{10.6}$$

where $D_п$ and D_o – the actual diameters of the outer ring, respectively, the roller and the hole of the housing, respectively, m; e – eccentricity, m.

After simple calculations we obtain that theoretically the assembly is provided under the condition that:

$$e \leq 2^{-1} \cdot \sqrt{D \cdot (ES_0 - \Delta D_m)}, \tag{10.7}$$

where D – nominal joint diameter, m; ΔD_m and ES_0 – respectively, the lower deviation of the outer diameter of the ring of the roll and the upper deviation of the hole of the housing, m.

Calculations taking into account 6 and 7 qualities for holes and showed that, for example, for the diameter range of 100…500 mm, the most common in large gearboxes, the allowable value of the offset of the hole axis relative to the plane of the housing connector is 1…4 mm, which, taking into account the economically achievable accuracy of calibration of boring rods of boring machines practically does not limit the folding of the folding unit of the feller. To ensure a gap in the joints of the outer locally loaded rolling pin rings in the holes of the detachable housings, it is recommended to assign tolerance fields H6, H7, G6, G7 regardless of the rolling pin type, dimensions and operating conditions. The layout of the tolerance fields of the joints of the outer ring of the rolling mill according to the recommended options is shown in figure 10.2.

Note that the tolerance fields JS7, K7 and M7, which are given among others in figure 10.2 and other above-mentioned sources for holes here are generally unacceptable, because, first, with the tolerance fields of the rings l0, l6. They will give transitional, not with a gap, landings, and secondly, do not take into account the described production phenomena of technological heredity.

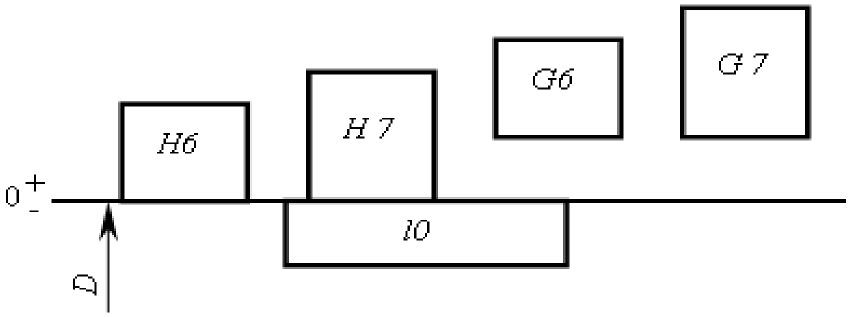


Fig. 10.2 The layout of the tolerance fields of the outer ring of the rolling pin (*l0*) and the hole (H6, H7, G6, G7) in the housing.

As modern production experience shows [27], the diameters of the holes of the detachable housings are assigned tolerance fields with the main deviation *H*. This, in our opinion, is due to the usual traditional principle of assigning tolerance fields to the size of all internal surfaces in the body part, i.e. in this case zero line, which is technologically rational [28]. Despite this, the quality of assembly, especially of large detachable housings, does not always meet the required standards [29]. Production observations have shown that often for the installation of rolling mills during fitting it is necessary to remove a layer 2…3 times larger than the normalized.

The calculations of the average gaps in the joints in the case of using the tolerance fields of the hole H6, H7, on the one hand, and the tolerance fields G6, G7, on the other hand (according to figure 10.2) showed that in the latter case the gap in joints with nominal diameters of 100…500 mm is 1.2…1.7 times larger.

It should be noted that in the case of assigning to the holes of such housings tolerance fields H6, H7 and characteristic due to the psychological factor of the operator negative asymmetry when processing the hole by test passes, gaps in joints (especially taking into account shape deviations and location of connected surfaces) generally close to 0, and in some cases (with unfavourable summation of deviations of surfaces in the process of assembly) instead of the gaps required for operation in such connections, in fact, even tension can be formed [30].

Conclusion to Chapter 10

The calculations of the average gaps in the joints in the case of using the tolerance fields of the hole H6, H7, on the one hand, and the tolerance fields G6, G7, on the other hand showed that in the latter case the gap in joints with nominal diameters of 100…500 mm is 1.2…1.7 times larger.

On diameters of openings of demountable cases under external rings with local loading it is expedient to assign fields of tolerances H6, H7, G6, G7, and in large folding units – fields of tolerances G6, G7 that will allow to create a backlash in connection with a ring, and, hence, the possibility of periodic rotation of the latter during operation of the folding unit of the rolling mill and reduce the uneven wear of raceways and the associated increase in durability of folding units with rolling mills.

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CHAPTER 11. RESEARCH OF SLIDING BEARINGS WITH REVERSE FRICTION PAIR AND INLAID LINERS MADE OF THERMOPLASTIC COMPOSITE MATERIALS

Introduction

The efficiency of composite polymeric materials is determined not only by high physical [1] and mechanical [2] and other properties [3], but also by the rational design of the friction unit [4]. Sliding bearings made according to the shaft-sleeve scheme with a sleeve made of thermoplastic materials have become widespread [5]. This design provides ease of manufacture and assembly of the bearing [6], interchangeability and ease of repair [7], which in this case is to replace the worn sleeve with a new one [8]. In this case, the sleeve is usually pressed into the seat of the bearing assembly [9], which does not allow to obtain the required tension or clearance during operation [10] and makes it difficult to take into account the compensation in the connection of plastic and metal parts [11]. At the same time, the sleeve receives pressure from the shaft only on the surface [12], which is determined by the angle of contact (girth) [13]. As a result of such interaction the plug wears out in one place [14]. In addition, the spike of the shaft, which rotates in the sleeve [15], when the abrasive particles enter the gap [16], wears out, and in some cases its wear may exceed the wear of the sleeve [17]. Known designs of plain bearings in the form of inverted friction pair [18], where the plastic sleeve is mounted on the shaft and rotates with it inside the outer holder [19]. The advantage of this design is the absence of wear of the shaft when the abrasive enters the friction zone [20]. The disadvantage is unsatisfactory heat dissipation from the friction surface into the environment, which is caused by the continuity of the sleeve [21]. Known design of the inverted plain bearing, which consists of a housing with a clamp installed in it with rod inserts, which are located in the longitudinal grooves [22]. Landing of the rod inserts is carried out on a cylindrical surface so that the axes of the inserts do not coincide with the inner surface of the holder [23], so that the rods are fixed from falling out [24]. Any shaft design can be attached to the shaft [25]. In this embodiment, the bearing operates according to the scheme of the inverted friction surface [26].

The disadvantage of this design is the complexity of manufacturing individual elements – split locking rings, clips. Uneven wear along the height of the liners leads to an uneven change in the clearance in the bearing throughout the service life [27]. During the initial operation of the bearing, the increase in the gap in the connection is more intense than in the next [28]. Uneven wear of the inserts on the contact area causes a proportional increase in temperature in the friction zone. In this case, the calculation of the temperature regime, which is an important element in the design of plain bearings with plastic inserts, is a task of some complexity [29].

The aim of the research is to increase the efficiency of the use of inverted plain bearings with type-setting inserts made of thermoplastic composite materials by determining their rational parameters.

Materials and methods

At mutual sliding of the corresponding plastic inserts there is a wear caused by fatigue of material which is shown thanks to adhesive forces of coupling at inevitable heat formation. In the design of the bearing it is irrational to increase the working thickness of the liners, as the size of the liner is regulated by the size of the inner holder of the bearing assembly. In addition, the increase in the diameter of the cylindrical liners leads (due to the low thermal conductivity of the plastic) to a change in the thermal mode of operation of the bearing. Heat accumulates in the polymer and leads to deterioration of the mechanical properties of plastics. To eliminate these shortcomings, an experimental design of the inverted plain bearing was developed, which differs from the existing ones in that the plastic inserts of the plain bearing are made in the form of prisms with a rectangular cross section, one side of which is an arc. The diameter of the arc is determined by the size of the fixed holder, as a result of which the surface area of contact of the liner with the fixed holder always remains constant, regardless of their linear wear. The plain bearing (figure 11.1) consists of a housing *1*, which is fixedly connected to the holder *2*, which is fixed by a locking screw *5*. Inside the holder *2*, coaxially with it, is a holder *4*, in the longitudinal grooves on the cylindrical surface of which are fixed plastic liners *3* by means of screws *9*. The holder *4* (figure 11.1) is fixed on the shaft *11* by a key *6*, and the thrust rings *8* on the shaft *11* and the seal *10* are held by the covers *7*.

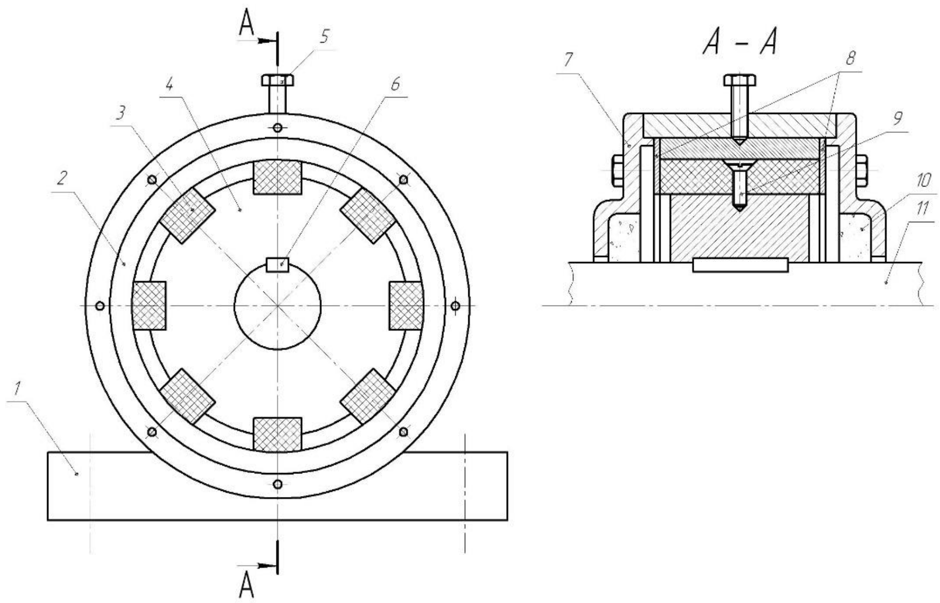

Fig. 11.1 The scheme of e sliding bearing with type-setting inserts from thermoplastic plastics.

The operation of the bearing shaft *11* (figure 11.1) with the holder *4* and fixed inserts *3* is carried out due to rotation inside the holder *2*. The presence of local wear of the holder *2* (figure 11.1) provides the possibility of rotation relative to the pressure zone of the shaft *11*. The force of possible axial movement of the holder *4* (figure 11.1) in the form of rectangles made by cutting along the generating cylinder. These sections also have a spherical surface of friction of the liners on the holder *2* (figure 11.1), create obstacles to the rotation of the liners around its own axis. The effort when installing the inserts in the holder *4* (figure 11.1), can further increase with the inevitable increase in temperature during operation of the bearing due to the increased coefficient of linear expansion of the polymer inserts. This fact, as well as the design solution contributes to the reliable fixation of the inserts in the holder *4*.

To simplify the fixation of the inserts in the holder *4* (figure 11.1), you can use a groove in the form of a dovetail, but this is due to the complexity of the technology of making a movable bearing holder.

Results and discussion

Determination of the load capacity of the sliding bearing in the form of an inverted pair is based on the calculation of the load distribution between the inserts (inserts), which is as follows. To the inner bearing holder of the sliding radial load application F, the inner holder is shifted relative to the outer on and the liners will be in position 0, 1, 2, $\ldots n$ (figure 11.2). In the absence of an initial radial clearance in the bearing, the liners absorb the load Q_0, Q_1, …, Q_n. At the same time, deformations will occur in the places of contact of sliding bodies with the outer holder δ_0, δ_1, …, δ_n. Placing the inserts relative to each other at an angle γ the equilibrium equation is as follows:

$$F = Q_0 + 2 \cdot Q_1 \cdot \cos\gamma + 2 \cdot Q_2 \cdot \cos 2\gamma + ... + 2 \cdot Q_n \cdot \cos n\gamma \,. \tag{11.1}$$

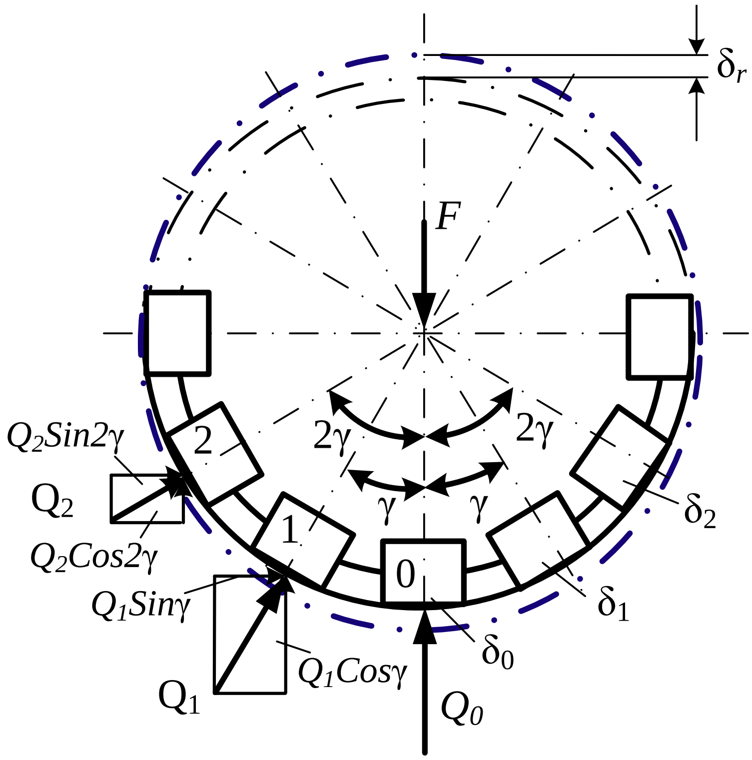

Fig. 11.2 The scheme of distribution of loading capacity of the sliding bearing with type-setting inserts.

According to Sribeck's theory, the relationship between load and deformation in this case can be expressed by the following relations:

$$\frac{Q_0^2}{\delta_0^3}=\frac{Q_1^2}{\delta_1^3}=\frac{Q_2^2}{\delta_2^3}=...=\frac{Q_n^2}{\delta_n^3},\ \text{or}\ \frac{Q_1^2}{Q_2^2}=\frac{\delta_1^3}{\delta_2^3};\ \frac{Q_0^2}{Q_1^2}=\frac{\delta_0^3}{\delta_1^3} \quad (11.2)$$

$$Q_1=Q_0\cdot\left(\delta_1\cdot\delta_0^{-1}\right)^{1.5},\ Q_2=Q_0\cdot\left(\delta_2\cdot\delta_0^{-1}\right)^{1.5},\ Q_n=Q_0\cdot\left(\delta_n\cdot\delta_0^{-1}\right)^{1.5}. \quad (11.3)$$

$$2\cdot Q_1\cdot\cos\gamma=2\cdot Q_0\cdot\cos^{1.5}\gamma\cdot\cos\gamma=2\cdot Q_0\cdot\cos^{2.5}\gamma,$$

$$2\cdot Q_2\cdot\cos 2\gamma=2\cdot Q_0\cdot\cos^{1.5}2\gamma\cdot\cos\gamma=2\cdot Q_0\cdot\cos^{2.5}2\gamma,$$

$$2\cdot Q_n\cdot\cos n\gamma=2\cdot Q_0\cdot\cos^{1.5}n\gamma\cdot\cos\gamma=2\cdot Q_0\cdot\cos^{2.5}n\gamma. \quad (11.4)$$

Substituting the ratio (11.2), (11.3), (11.4) to (11.1) obsessed

$$F=Q_0+2\cdot Q_0\cdot\cos^{2.5}\gamma+2\cdot Q_0\cdot\cos^{2.5}2\gamma+...+2\cdot Q_0\cdot\cos^{2.5}n\gamma. \quad (11.5)$$

$$F\cdot Q_0^{-1}=1+2\cdot\cos^{2.5}\gamma+2\cdot\cos^{2.5}2\gamma+...+2\cdot\cos^{2.5}n\gamma. \quad (11.6)$$

Since most radial bearings have the number of rolling elements in one row $z=8...20$, then by analogy for a plain bearing with plastic liners determine the value z within this range z = 8, 10, 12, 15, 20, that corresponds γ = 24°, 45°, 36°, 30°, 18°. Calculations show that the largest value Q_0 acquires at z =15. Thus, the load Q_0, which receives the most loaded liner of the sliding bearing placed in the plane of action of external radial loading (γ = 0) will be determined from the formula:

$$Q_0=4.37\cdot F\cdot z^{-1}. \quad (11.7)$$

These dependences allow to determine the load that is perceived by the most loaded liners under the simplest load condition: a purely radial load is applied to a radial bearing having a contact angle γ = 0. It should be noted that the load Q_0 within the angle γ does not remain constant, but changes relatively little $Q_0=(4.3...4.43)\cdot F\cdot z^{-1}$ and this change in engineering calculations can be neglected and take into account that the coefficient in the formula (11.7) constant and equal to 4.37. The appearance of a radial gap significantly affects the load distribution Q on the liner. In the case of a gap-free bearing, all inserts are loaded, which are located below the horizontal axis and the angle of the loaded area is $2\gamma=\pi$. With a radial gap $\delta_r>0$ the angle of the loaded zone decreases and the radial load is distributed over a smaller number of liners, reloading each liner within the loaded area. The influence of the radial gap in engineering calculations is taken into account by increasing the coefficient in the formula:

$$Q_0 = 5 \cdot F \cdot z^{-1}. \tag{11.8}$$

More accurate calculations can use the formula:

$$Q_0 = 4.37 \cdot \left(0 + 40 \cdot \delta_r \cdot h^{-1}\right) \cdot F \cdot z^{-1}. \tag{11.9}$$

where h – liner height; δ_r – radial clearance.

In order to clarify the application of dependences (11.1), (11.2), (11.4) - (11.9) to calculate the number of inserts of inverted plain bearings, experimental studies were conducted, which consisted of the following. In the laboratory bearing (figure 11.3) were made several versions of the inner clips, which differed in the number of inserts and their sizes. In the process of testing the load in all variants of the internal clamps was applied the same: $F = 800$ H. According to formula (11.9), the values of the loads on the most loaded liner were determined at their variable number in the inner holder z = 4, 6, 8, 12. The tests were performed in grease. The sliding speed under these test conditions was constant and was Θ = 0,392 m/s. The results of experimental studies of comparative tests of metal-polymer plain bearings showed (figure 11.4) that when the number of liners z = 4…6 there is intense heat generation and loss of stability of polymer liners (melting and partial application of material on the sliding surface – the outer holder).

This is due to the fact that the allowable gap δ = 0.1…0.15 mm in the bearing is insufficient when the number of inserts z =1…6. Due to the relatively high coefficient of linear expansion of the plastic liners at the specified gap, even with a slight increase in temperature in the friction zone, there is a process of adhesion with partial jamming. Increasing the operating gap leads to an increase in the load on the liner. It is established that the most rational number of inserts in the inner holder of the inverted plain bearing is z = 8…12. From figure 5 shows that under the same conditions at z =8…12 the moment of friction during running-in is less than at z = 4…6. Evaluation of the effectiveness of graphite-modified coprolon-B in bearing assemblies was performed on wear resistance. For comparative evaluation, the bearing assembly of the initial cleaning apparatus of the Moreau corn harvester was taken. Four 11237K rolling bearings are mounted on the shaft. Experience of operation of the combine showed that installation of sliding bearings is not economic and does not reduce the vibration loadings which arise at work of the initial cleaning device.

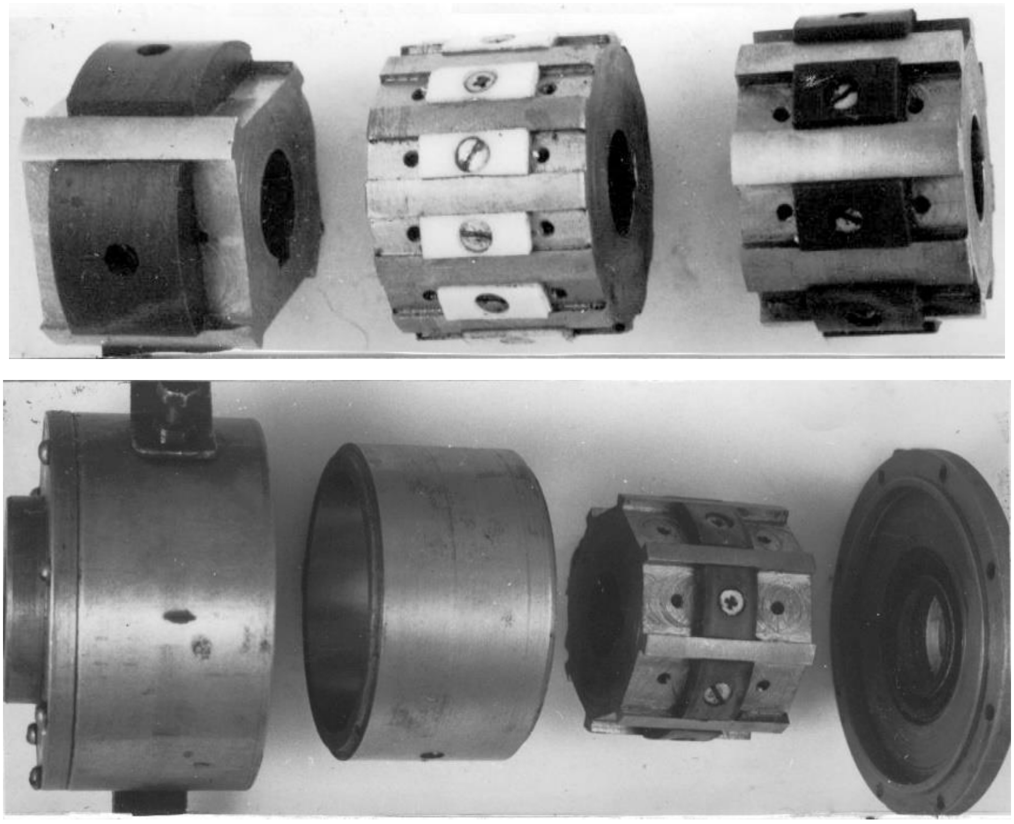

Fig. 11.3 Design of the laboratory sliding bearing.

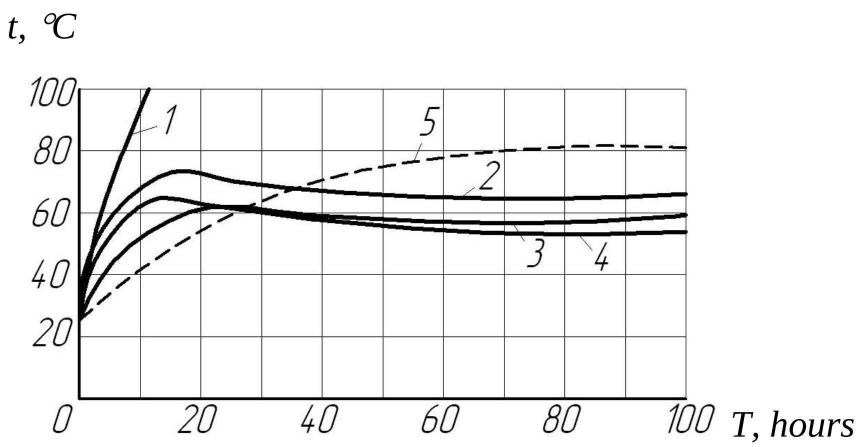

Fig. 11.4 Dependence of temperature in the friction zone of different design of the plain bearing on the test time T, hours: 1 – z = 4; 2 – z = 6; 3 – z = 8; 4 – z = 12; 5 – direct friction pair.

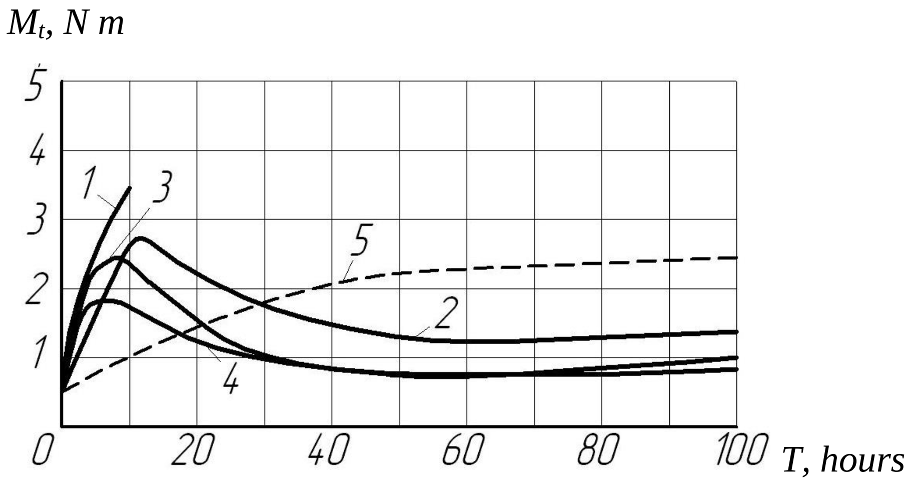

Fig. 11.5 Dependence of the moment of friction M_t of various design of the sliding bearing on test time T, hours: 1 – z = 4; 2 – z = 6; 3 – z = 8; 4 – z = 12; 5 – direct friction pair.

Frequency of rotation of a shaft of the initial cleaning device n =200 min^{-1} at nominal working loading on one bearing F =600 N. According to [30, 31], the design dimensions of the plain bearing in the form of a direct friction pair are calculated.

Diameter of a basic site of a shaft (neck) d_s =35 mm; wall thickness of the sleeve $S = (0.45...0.5) \cdot \sqrt{d_s} = 3$ mm; sleeve length $l = (1.0...1.3) \cdot d_s = 45.5 \approx 46$ mm.

The experimental sliding bearing in the form of the inverted friction pair is installed on the basis of the standard case of the rolling bearing 11237K (figure 11.6). The outer friction holder *7* (figure 11.6) is pressed into the housing *8*. The holder *7* (figure 11.6) can be installed in the housing by sliding landing, but in this case it must be provided to fix it from turning.

The outer side surface of the holder *7* (figure 11.6), as well as the inner sliding surface is heat treated. The inner holder *5* (figure 11.6) is mounted on the shaft by means of a split conical sleeve with a clamping nut on the holder *4* are fixed eight inserts *6* of fluoroplastic-4 with a paraffin content of 6%.

The thrust ring *2* (figure 11.6) prevents the displacement of the shaft with the holder *5* and is remote when determining the lateral gap between the ends of the inserts and the inner holder *5*. In the cover *1* (figure 11.6) and the housing *8* are rubber-metal sealing sleeves.

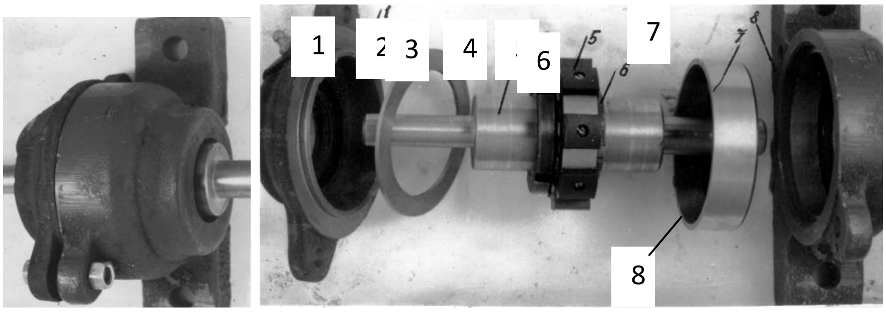

Fig. 11.6 General view and elements of the laboratory sliding bearing in the form of an inverted friction pair based on a rolling bearing 11237K.

The results of laboratory tests showed that under the same conditions, the total linear wear of the bearings of the sliding bearing in the form of inverted friction pair at 100 hours of continuous operation in a grease environment is 20…25% less than the bearing in the form of direct friction pair. This confirms the effectiveness of the use of plain bearings in the form of inverted friction pairs in the loaded components of machines and mechanisms for various purposes.

Conclusion to Chapter 11

The largest value of the initial load Q_0 acquires at z =15, namely the initial load receives the most loaded liner of the sliding bearing placed in the plane of action of external radial loading at γ =0.

The magnitude of the initial load Q_0 within the angle γ do not remain constant, but change relatively little $Q_0 = (4.3...4.43) \cdot F \cdot z^{-1}$ and this change in engineering calculations can be neglected and take into account that the coefficient is 4.37.

The appearance of a radial gap significantly affects the load distribution Q on the liner, so with a radial gap $\delta_r > 0$ the angle of the loaded zone decreases $2\gamma \le \pi$ and the radial load is distributed over a smaller number of liners, reloading each liner within the loaded area.

The total linear wear of the inserts of the sliding bearing in the form of inverted friction pair at 100 hours of continuous operation in the environment of grease is 20…25% less than the bearing in the form of direct friction pair.

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CHAPTER 12. MODELING OF NORMATIVITY OF CRITERIA OF TECHNICAL LEVEL OF FORAGE HARVESTERS COMBINES

Introduction

Carrying out such a comparison in practice contains additional difficulties [1]. This is due to the fact that the indicators of the object are compared [2], as a rule, which differs in the beginning and resource of its life cycle from other objects of analogues [3]. In addition, the indicators of the evaluated object may be stable throughout the resource [4] or only slightly change within the design or technical solution embedded in the object [5], and indicators, in general, determine the level of world achievements (this type of forage harvesters combines), acquire significant changes over time [6], reproducing the continuous process of selecting optimal technical solutions [7].

Thus, the ratio of relatively stable indicators of the object being evaluated and continuously updated over time indicators of the level of world development of science [8] and technology will always be a function of time [9]. This means that the assessment of the technical level of the object can be reliable only at a specific point in time, which refers to the existence of objects of analogues, selected as a comparison [10]. At any time, the assessment of the technical level of the object will not be completely reliable due to the error caused by changes in the technical level during the time that has elapsed since the existence of the analogue object [11]. In the general case [12], the error in estimating the technical level of the object δ will be equal to [13]: $\delta_i = k_i \cdot \Delta t$, where k_i – the coefficient of change of the *i*-th indicator of the technical level; Δt – time discrepancy between the moment of determining the technical level and the moment of assigning the object of analogue to the best world achievements. A similar type of error can occur when the moments of selection of several objects of analogues do not match (not the coincidence of the moments of their development in production). Its value $\overline{\delta}$ will be equal to [14]: $\overline{\delta} = \sum_{i=1}^{n}(k_i \cdot \Delta t \cdot n^{-1})$, where n – is the number of analogous objects accepted in the evaluation. The difficulty of taking into account this type of error is also due to the fact that to assess the technical level of the object have to operate with a set of unit indicators of the types of objects analogs [15].

According to the values of the time indicators located in time, it is necessary to find at each given moment such a calculated state of the generalized indicator, which would reproduce the general trend of its change over time [7], which will express through the regularity of this type of objects [16]. One of the acceptable methods is to solve such problems by comparing the economic efficiency of analogous objects, taking into account the combined effect of all parametric and functional properties of the object [17]. However, such methods have clearly limited properties within the comparison of already optimized variants within themselves and it is impossible to extend them to the synthesis of new variants [18]. Thus, before determining one of the two objects that do not coincide according to the principle of technical solution of analogues, there is a problem of preliminary optimization of design, manufacturing technology and methods of operation of each object to solve the evaluation of their properties and efficiency. Using the dependence [19], which determines the economic efficiency of the ratio of savings to costs. It is considered the most appropriate in the overall set, which characterizes the object of indicators. This methodological approach provides the possibility of system-integrated analysis of the technical level, the content of which is to summarize the individual indicators through a set of unit indicators of the object, which evaluates the economic performance of the entire object. The complexity, and sometimes the impossibility of such an analysis, given the need to use data on world counterparts, require justification of other approaches [20].

Purpose of research

From the author's point of view, it would be rational to assess the technical level as a function of the rate of change of indicators that determine the development of this type of equipment over a limited period of time, for example, over the life of the object. During this period, as a rule, you can consider the performance of two or three generations of this type of equipment (predecessor, existing and projected). Develop methodological provisions for assessing the technical level of agricultural machinery with their subsequent standardization of forage harvesters combines.

Materials and methods

In the general case, the task of obtaining for each fixed point in time a generalized indicator of the technical level of the object is to determine the regression function of time:

$$K_{TPj} = \sum_{j=1}^{m} \left[k_{m_j} \cdot K_j(t) \right], \tag{12.1}$$

where K_{TP_J} – the generalized value of one j-th type of indicators at a given time t_r; $K_j(t_r)$ – values of unit indicators; k_{m_j} – weighting factor of each indicator j-th type; m – the number of indicators of one j-th type.

At small intervals of evaluation, covering two or three changes of generations of this type of technology, the function $K_j(t_r)$ for each one j-th type of unit indicator K can be approximated with sufficient accuracy by a linear relationship:

$$K_{jir} = K_{ji0} + k_i \cdot t_r, \tag{12.2}$$

where K_{jir} – the value of the *i*-th indicator of the technical level of the j-th type at a given time $t_r = \tau$, which is taken as the base, ie $K_{jr} = K_{баз}$; K_{ji0} – the value of the *i*-th indicator of the technical level of the j-th type at the initial time $t_r = 0$.

Results and discussion

Using the mathematical apparatus of the least squares method, we obtain:

$$K_{ji0} = \sum_{i=1}^{n}(K_{ji} \cdot t_r) \cdot \sum_{r=}^{w} t_r - \sum_{i=1}^{n} K_{ji} \cdot \sum_{r=}^{w} t_r^2 \cdot [\left(\sum_{r=}^{w} t_r\right)^2 - w \cdot \sum_{r=}^{w} t_r^2]^{-1}, \tag{12.3}$$

$$k_i = [\sum_{i=1}^{n} K_{ji} \cdot \sum_{r=}^{w} t_r - w \cdot \sum_{i=1}^{n}(K_{ji} \cdot t_r)] \cdot [\left(\sum_{r=}^{w} t_r\right)^2 - w \cdot \sum_{r=}^{w} t_r^2]^{-1}, \tag{12.4}$$

where K_{ji} – the value of the *i*-th indicator of the technical level of the j-th type at a given time t_r; t_r – time value from the total set w.

If the accuracy of the calculation is not required or if the duration of the evaluation (number of analogues and time) is insufficient to perform calculations by the method of least squares, the corresponding calculations can be performed with less accuracy by a simplified formula $K_{ji0} = \overline{K_j}$:

$$k_i = \sum_{i=1}^{n} \Delta K_{ji} \cdot (\sum_{r=1}^{w} \Delta t_r)^{-1}, \tag{12.5}$$

where ΔK_{ji} – change the value of the *i*-th indicator of the technical level of the j-th type at the time Δt_r. Due to the fact that the change ΔK_{ji} in time is determined by analogous objects (A_1–A_9) in the sequence, obtained for the durability of combine harvesters (figure 12.1).

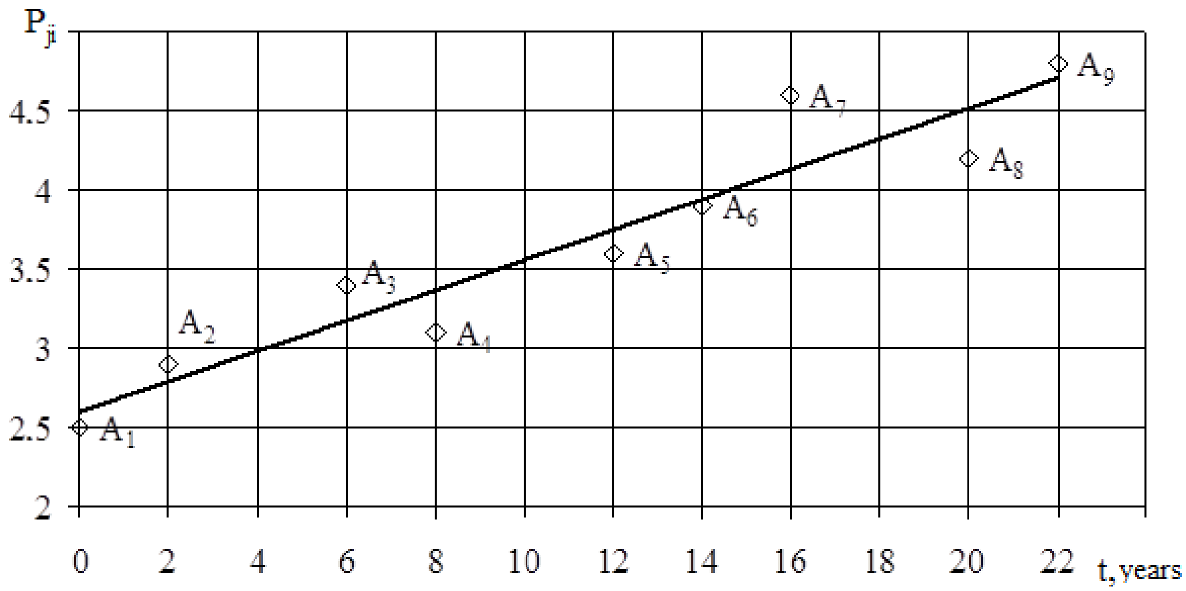

Fig. 12.1 Changing the value of a single indicator of objects of analogues in retrospect.

Substituting the value of figure 1 in expression (5) we obtain:

$$K_{ji0} = (412.6 \cdot 100 - 33 \cdot 1584) \cdot (10000 - 9 \cdot 1584)^{-1} = 2.35,$$

$$k_i = (33 \cdot 100 - 9 \cdot 412.6) \cdot (10000 - 9 \cdot 1584)^{-1} = 0.098.$$

These data allow you to get the most likely value K_{ji} at a given time t_r. If we calculate at five-year intervals, then according to formula (12.4) we obtain (figure 12.2). The obtained regularities (figure 12.1 and figure 12.2) bear the middle position of the approximation in retrospect of the parameter K_{ji} within its actual values.

Therefore, it is necessary to consider these methodological provisions not as a replacement for mathematical optimization of the technical level, but the integration of analog objects with the location of the optimal curve, which is approximated within the laws at the moment. In this regard, the task is to select and process or summarize the individual indicators of the assessed object. According to the results of our own research, we find that successfully competing in the market objects that represent the best achievements of technical progress, as a rule, have no absolute analogues in terms of values. Each of these objects has advantages in one or another single indicator or even a combination of several of them, which will satisfy the needs of certain consumers or are more rational in solving certain constructions of the object. Thus, the identification of the optimal object by the dominance of a single indicator, which is accepted in the assessment of the main for a particular consumer, has a subjective context in relation to another consumer.

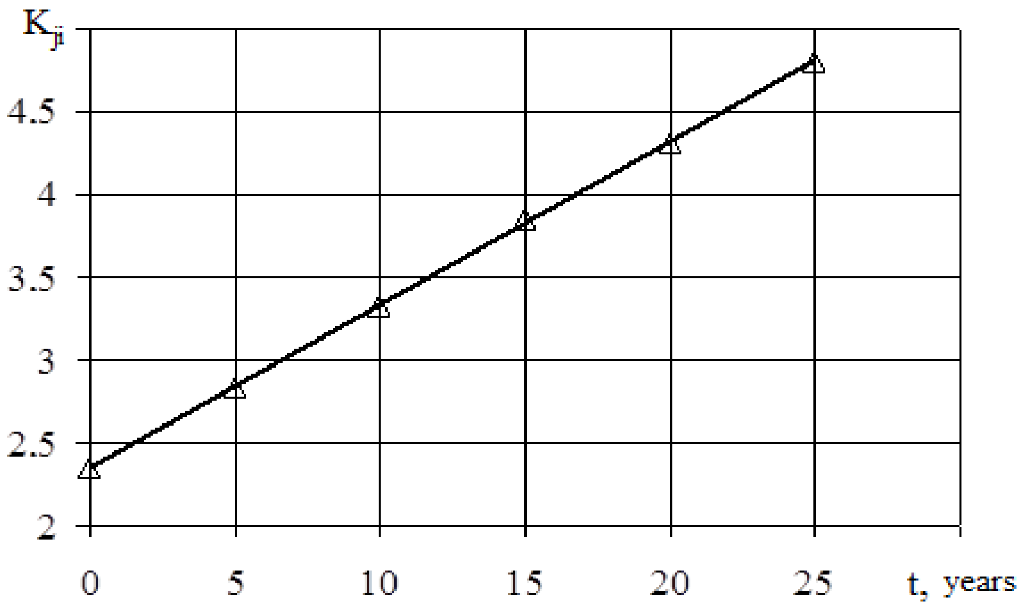

Fig. 12.2 Changing the value of a single indicator of analog objects in five-year periods.

Investigating the rate of change of parameters of a certain set (chronological series) of analytical objects as an indicator of technical level, we can conclude that the difference in the rate of change of an indicator has the degree of influence of each indicator on the formation of initial properties that determine the ability of this technique at each stage of its development to meet the specific requirements of consumers. That is, only important indicators are subject to improvement or development. If the indicator is improving at an accelerated pace, it means that the importance of factors such as manufacturability or world conditions. Therefore, the value of the weight m_{ji} of each indicator K_{ji} is determined by the expression through the coefficient v_{ji} of the rate of its change over time, attributed to the original value of the indicator K_{ji0}: $m_{ji} = v_{ji} \cdot K_{ji0}^{-1}$, or, using expressions (12.5) and (12.6) by the method of least squares, we obtain:

$$m_{ji} = [\sum_{i=1}^{n} K_{ji} \cdot \sum_{r=1}^{w} t_r - n \cdot \sum_{i=1}^{m} (K_{ji} \cdot t_i)] \cdot [\sum_{i=1}^{m} (K_{ji} \cdot t_i) \cdot \sum_{r=1}^{w} t_r - \sum_{i=1}^{n} K_{ji} \cdot \sum_{r=1}^{w} t_r^2]^{-1}. \qquad (12.6)$$

By expression (12.6), determining the values m_{ji} in fractions relative to l a number i of indicators K_{ji}, we obtain the value of the weighting factor $M_{ji} = m_{ji} \cdot (\sum_{i=1}^{l} m_{ji})^{-1}$. The obtained data K_{ji} and M_{ji} are sufficient to determine the generalized value of the technical level $\overline{K}_{ji}$ indicator at an arbitrary time t_r: $\overline{K}_{ji} = \sum_{i=1}^{l} (M_{ji} \cdot K_{ji})$. According to the data obtained, the indicator K_{jir} according to formula (4), take as a baseline, then: $q_{ji} = K_{ji} \cdot (K_{jir})^{-1}$. Let at the time of development of the technical task the objects which characterize a modern level – objects analogs are known (table 12.1). To determine the technical level, which is set by the following parameters: estimation accuracy – 0.25%, temperature error – 0.20%, number of channels – 6 units, temperature range – 120 °C, gas consumption – 1.8 l/min. Substitute the values K_{ji} (table 1) and summarize them. The value of the technical level according to formula $\overline{K}_{ji} = \sum_{i=1}^{l} (M_{ji} \cdot K_{ji})$ is equal to 1.14, i.e. the value of exceeding the technical level of 14%. However, the application of the proposed method in solving various types of practical problems of determining the technical level, such as the formation of the technical task for the development of the object, assessing the quality of this development, assessing the compliance of the object to the criteria of quality category ISO series, we can demolish (1year), which will include the technical level of the assessed object. One of the solutions can be to determine the value of the time indicator of the technical level of the object by determining the time intervals of growth of the technical level indicator by the value Δ, i.e.: $\Delta t_r = \overline{K}_{ji} \cdot (\sum_{i=1}^{l} M_{ji} \cdot k_i \cdot \{K_{ji}\}^{-1})^{-1}$, where Δt_r – the time difference from the moment when the value of the technical level was determined $\overline{K}_{ji}$.

Table 12.1

The main indicators of the technical level of the maintenance tool for controlling the heating temperature and crankcase gas flow.

Indicator	K_{ji}	Objects					
		C_1	C_2	C_3	C_4	C_5	C_6
Estimation accuracy, %	K_{11}	1.0	1.0	0.5	0.3	0.5	0.3
Temperature error, %	K_{12}	0.6	0.6	0.5	0.3	0,3	0.25
Number of channels, units	K_{13}	1	2	2	2	3	5
Temperature range, °C	K_{14}	45	45	60	90	105	110
Gas consumption, l/min	K_{15}	8.0	7.5	5.5	6.0	2.5	4.0

K_{ji}		K_{ji0}	k_i	K_{jir}	m_{ji}	M_{ji}	q_{ji}	$\overline{K}_{ji}$	$(M_{ji} \cdot k_i)/K_{ji}$
K_{11}	0.25	0.98	0.045	0.17	0.046	0.098	0.68	0.067	0.0176
K_{12}	0.20	0.60	0.025	0.16	0.042	0.089	0.75	0.067	0.0110
K_{13}	6	0.91	0.193	4.39	0.212	0.450	1.36	0.610	0.0145
K_{14}	120	36.5	4.850	123.8	0.133	0.283	0.97	0.275	0.0114
K_{15}	1.8	8.3	0.306	2.80	0.037	0.078	1.55	0.121	0.0133

K_{ji}	Object D			Object F		
	K_{ji}	q_{ji}	$M_{ji} \cdot q_{ji}$	K_{ji}	q_{ji}	$M_{ji} \cdot q_{ji}$
K_{11}	0.25	0.68	0.067	0.25	0.68	0.067
K_{12}	0.20	0.75	0.067	0.20	0.75	0.067
K_{13}	12	2.73	1.230	3	0.68	0.306
K_{14}	120	0.97	0.275	120	0.97	0.275
K_{15}	3.6	0.78	0.061	0.9	3.11	0.242
$\overline{K}_{ji}$	1.700			0.957		

Conclusions to Chapter 12

1. Since the annual variable value of the technical level is 0.0678, at $\overline{K}_{ji}$ 14%, we have Δt_r=2.06 years. Thus, exceeding the complex indicator of the technical level of 0.14 is a period of two years to catch up with world counterparts. Than applying the developed analytical provisions for the assessment of promising solutions under conditions of changing indicators, we obtain the following tabular data (table 1). Thus, with the proposed technical solutions and the calculated dynamics of changes in the technical level of 0.678, the introduction of object D provides an advance in the technical level for 10.96 years.

2. These examples allow us to assess the correctness of the methodology from the practical significance of the proposed method of assessing the technical level of forage harvesters combines, the advantage of which is the mathematical justification of weighting factors when used in complex assessment, reducing the impact of subjective expert assessments. In the future, the development of industry standardized provisions of the methodology for assessing the technical level of forage harvesters combines products is envisaged.

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CHAPTER 13. RESEARCH OF MIXED CARBON SORBENTS FOR REMOVAL OF OIL PRODUCTS FROM WATER AND SOIL FOR PRESERVATION OF ENVIRONMENTAL INFRASTRUCTURE

Introduction

Fuels and lubricants, even with careful storage and use in large quantities, enter the environment causing significant environmental damage [1]. Various methods are used to clean water bodies [2] and soils from oil [3] and oil products [4]. The most widely used in practice method of purification from petroleum products – absorption (sorption) [5]. Recently, mixed carbon-carbon [6] and carbon-mineral sorbents have been widely used to remove petroleum products from the environment [7] and preserve the infrastructure of the environment [8]. Mixed are mixtures obtained from two or more individually pure sorbents, which acquire different properties from the source materials [9]. The structural-sorption properties of mixed sorbents are not a simple set of properties of the source components [10], there is always more or less deviation from the original analogues [11], and, in some cases, the resulting sorbent may acquire completely new properties [12].

The need for the use of mixed sorbents is associated with: reducing the use of more expensive (due to the cost of carbonization) carbon sorbents [13]; limited amount, in certain areas, of raw materials for carbon sorbents [14]; reduction of energy costs; obtaining cheaper sorbents [15]; improvement of sorption and technical and economic indicators; efficiency and success of removal of oil products from the environment [16]; relevance of environmental quality monitoring [17].

Petroleum products are common pollutants in the biosphere [18]. Once they enter the environment, as a result of chemical and biological decomposition, harmful toxic substances are formed, which also have a negative impact on living organisms [18]. Studies of carbonization processes, taking into account the nature and structure of raw materials, aimed at obtaining effective sorbents with high sorption capacity and absorption capacity of petroleum products are important [19]. Obtaining and determining the system of the most efficient use of mixed carbon-carbon and carbon-mineral sorbents [20], taking into account their physical and chemical, structural-sorption characteristics [21], in comparison with the relevant indicators of raw materials [22], for removal of petroleum products from the environment and preservation of environmental infrastructure, has important scientific [23] and practical significance and needs further research [24].

Purpose of research

The purpose of our research was to study technologies for obtaining carbon sorbents from by-products of agriculture, woodworking industry and utilities under the influence of low-temperature single-stage carbonization, as well as structural, physicochemical, absorption properties of mixed carbon-carbon and carbon-mineral and carbon-mineral economic feasibility in the removal of petroleum products from the environment and the preservation of environmental infrastructure.

Materials and methods

The use of carbon materials in vegetable raw materials is becoming more common due to their high sorption properties. The indicators characterizing sorbents of carbon origin, first of all, are defined both by the nature of initial raw materials, and conditions of their reception.

We studied the structural and absorption characteristics of sorbents for the removal of petroleum products and the preservation of environmental infrastructure: plant origin; carbon sorbents based on vegetable raw materials obtained under the influence of low-temperature one-stage carbonization on the raw material; mixed carbon-carbon and carbon-mineral sorbents. The study of the process of carbonization of sawdust of coniferous trees (pine) revealed that at each studied carbonization temperature we have the optimal value of its duration, at which the oil content of carbonate is maximum, as shown in figure 13.1.

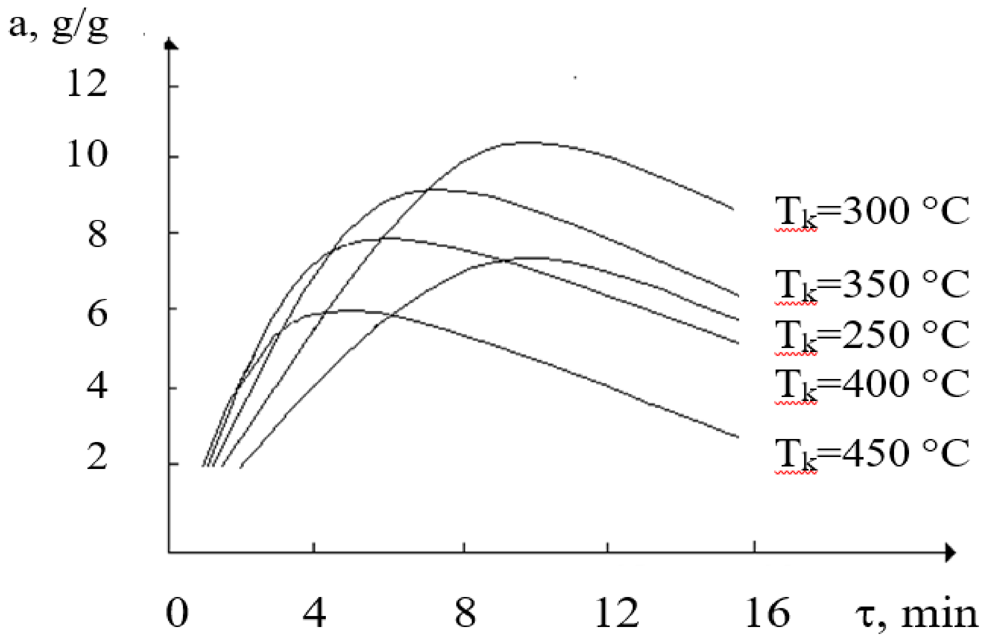

Fig. 13.1 Dependence of oil content of carbon materials a (g/g) from pine sawdust on time τ (min) and temperature of carbonization T_k (°C).

The appearance of maxima in figure 13.1 can be explained by the change in the porous structure and surface chemistry of carbon materials during carbonization.

Thus, at the initial stage of the process of carbonization of pine sawdust (up to 350 °C), due to the removal of moisture, volatile and resinous components, pores are formed, which leads to an increase in its sorption capacity for oil and petroleum products to some maximum value.

Further carbonization (350-450 °C) leads to carbon burnout and, consequently, an increase in the number of macro pores, which changes the structure of carbonates in the direction of reducing mesa porosity. As you can see, under the influence of temperature, the porous structure is partially degraded, which leads to a decrease in the sorption capacity of the carbonate.

The optimal mode of carbonization of pine sawdust is 300 °C (±10 °C) and a time of 8-10 minutes, which is confirmed by the data in table 13.1.

In the case of purification of oil and petroleum products from water, mainly from its surface, an important characteristic of the sorbent is buoyancy. Because, at loss of buoyancy, in the presence of oil agglomerates the bottom of reservoirs is polluted and repeated pollution of a surface of water in case of desorption of oil is possible. We conducted studies of carbonates for buoyancy, obtained at different values of temperature and heat treatment time, depending on the duration of their stay on the water surface. Mixed carbon-carbon sorbents are obtained by mechanical mixing of carbon materials in certain predetermined ratios. Obtaining mixed sorbents using, as one of the components, pine sawdust carbonate (as a base), which has the best sorption properties compared to other sorbents, with optimal carbonization parameters, allows you to have cheaper sorbents due to lower energy costs.

Table 13.1

Change in the structural and sorption characteristics of carbon materials from pine sawdust depending on the conditions of heat treatment.

T_c (°C)	τ (min)	Sorption volume of pores V_s (C_6H_6), (cm^3/g)	Sorption capacity (sorption value) $A(ESR)$ (mg-eq/g)
250	5	0.08	1.5
250	10	0.10	2.6
250	15	0.09	1.7
300	5	0.09	1.8
300	10	0.10	3.6
300	15	0.07	2.4
350	5	0.07	2.3
350	10	0.06	2.5
350	15	0.06	1.7
400	5	0.06	2.3
400	10	0.05	1.9
400	15	0.04	1.6

The structural sorption capacity of various mixed carbon-carbon sorbents, based on vegetable raw materials, which have the most widespread use for purification from petroleum products, as well as carbon-mineral sorbents was investigated. The following carbon-carbon sorbents have been proposed.

Pine sawdust carbonate: walnut sawdust carbonate (optimal carbonization conditions) in percentage 90:10, 80:20, 70:30, 60:40, 50:50, 40:60, 30:70, 20:80, 10:90; expanded graphite in the percentage 90:10, 80:20, 70:30, 60:40, 50:50, 40:60; since the expanded graphite has a fairly low bulk density (0.012 g/cm^3), the percentage of 30:70, 20:80, 10:90 should not be used; rapeseed straw carbonate (optimal carbonization conditions) in a percentage ratio of 50:50.

Carbon-carbon sorbents, within one mixture, differ from each other only in the ratio of components. Mixed carbon-mineral sorbents were obtained by mechanical mixing of carbon and mineral components in certain predetermined ratios. The value of the sorption capacity of natural mineral materials in relation to oil and oil products decreases in the following series: saponite → ash → bentonite → crucible → → clinoptilolite. Using pine sawdust, with optimal carbonization parameters, having high values of sorption capacity and natural sorbents, carbon-mineral sorbents were obtained: saponite, ash, bentonite; in percentage 90:10, 80:20, 70:30, 60:40, 50:50, 40:60, 30:70, 20:80, 10:90. Mixed sorbents are a mixture of two individually pure materials. The determined structural sorption characteristics of mixed sorbents were compared with the corresponding additive values calculated on the basis of the known composition of the sorbent. Structural and sorption characteristics of mixed materials are not a simple sum of the properties of the original components, there is always a deviation from the additivity.

Results and discussion

Structural-sorption and physico-chemical characteristics of mixed carbon-carbon materials change with the change of mass fraction of components in the composition of the mixture. Thus, V_s and $A(ESR)$ capacity of petroleum products for carbon-carbon sorbents increase and reach a maximum at a ratio of components close to 50:50% of the mass (figure 13.2).

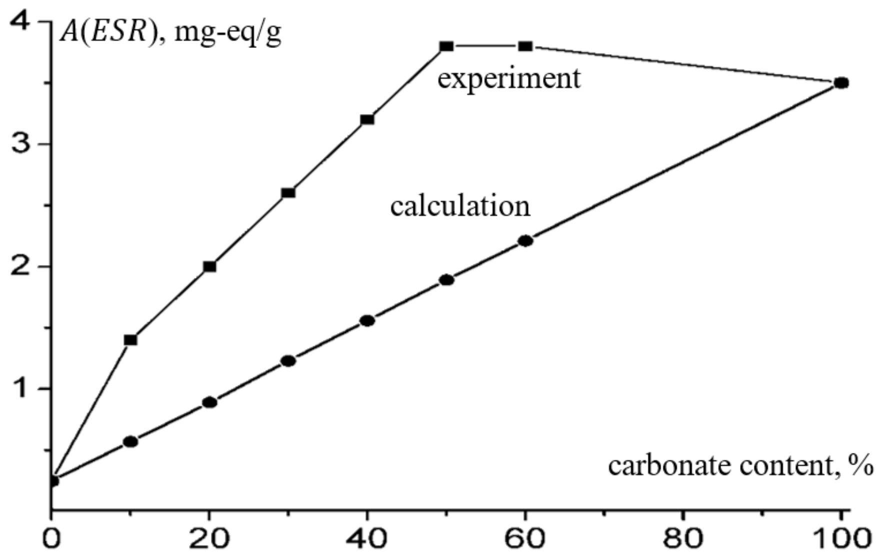

Fig. 13.2 Dependence of the value of the exchange capacity on the composition of carbon-carbon mixtures (pine sawdust carbonate: expanded graphite).

Theoretical (calculated) values of the sorption capacity were obtained based on the actual values of each sorbent and its mass fraction in the composite mixture. Manifestation of the synergistic effect in the mixed material, apparently, is a consequence of the determining role of the surface chemistry of carbon materials of plant origin, as the static exchange capacity of graphite in comparison with carbon sorbents are quite low. A further reduction in the size of the synergistic effect is due to a reduction in the carbon sorbent content of the mixture. The manifestation of the synergistic effect in the case of the sorption volume of the pores of mixed carbon-carbon sorbents is due to both the porous surface of carbon materials of plant origin and the sorption volume of the pores of expanded graphite. Since the expanded graphite has a low value of bulk density (0.012 g/cm^3), the further decrease in the size of the synergistic effect is associated with a decrease in the content of expanded graphite in the mixture.

Inadequate nature of sorption of oil and oil products is observed, first of all, on carbon-carbon sorbents of mixed type. Oil consumption increases with the change in the ratio of components in materials of mixed type and reaches a maximum at such values at which the maximum $A(ESR)$ and sorption volume of the pores are observed. Oil capacity and absorption capacity of petroleum products of pine sawdust carbonate:

expanded graphite, at a ratio (within 50:50%), 2.5-3.0 times higher than the capacity of pine sawdust carbonate, which makes it extremely effective and promising in use for cleaning from petroleum products of the environment and preservation of environmental infrastructure. Therefore, the combination of sorption pore volume (due to expanded graphite) and high exchange capacity (due to pine sawdust carbonate) can cause a high value of oil-carbon-carbon mixed materials. Mixtures of expanded graphite with sawdust carbonates of other trees or straw have high sorption properties. They have higher sorption rates compared to the source materials, so in the absence of sawdust from coniferous trees, you can use to make a carbon-carbon mixture of expanded graphite and carbonates from sawdust from other trees or straw.

Manifestation of antisynergistic effect in the case of sorption volume of pores and $A(ESR)$ of mixed carbon-mineral sorbents is due to changes in the nature of active centers, namely, by reducing the content of carbon components in the mixture (figure 13.3).

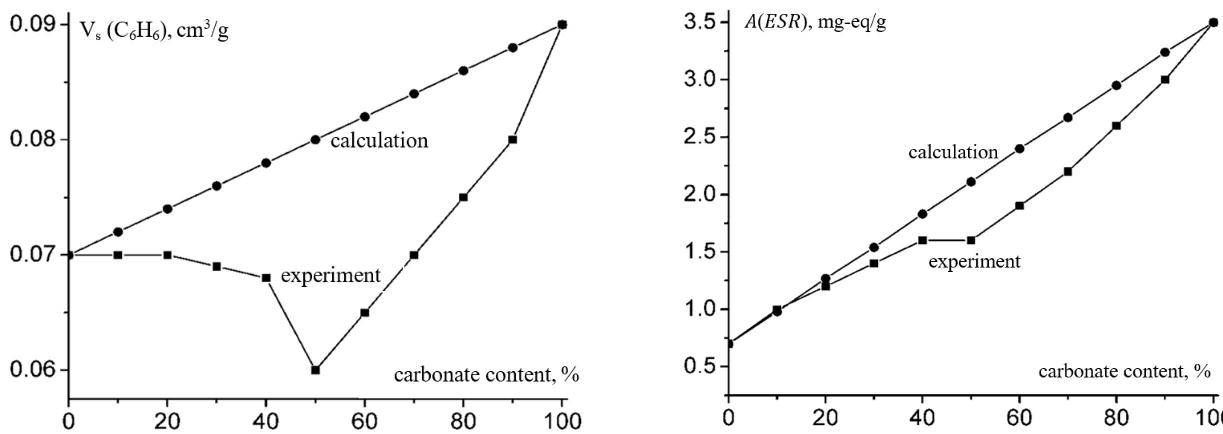

Fig. 13.3 Dependence of the value of the sorption volume of the pores and the exchange capacity on the composition of the carbon-mineral mixture (pine sawdust carbonate: saponite).

The obtained mixed carbon-mineral materials were investigated for buoyancy. It turned out that the mineral component (saponite) sinks when it is on the surface of the water for one hour, the carbon component has a buoyancy of 99% for 6 months.

Therefore, as the ratio of the mass fraction of the components in the mixture changes and buoyancy. As the mineral component in the mixture increases, the buoyancy of the obtained materials decreases.

Inadequate nature of sorption of petroleum products is also observed on carbon-mineral sorbents of mixed type.

The oil content decreases as the ratio of components in materials of mixed type changes and reaches a minimum at such values at which the minimum value of $A(ESR)$ and the volume of sorption pores are observed.

The manifestation of the antisynergistic effect in the case of mixed carbon-mineral materials is due to a change in the nature of the active centers, namely, by reducing the content of the carbon component in the mixture.

The obtained data on the influence of the qualitative composition of mixed materials on their sorption capacity in relation to oil and oil products allow us to suggest the feasibility of using carbon-carbon materials as effective sorbents of oil products from water and soil, and carbon-mineral, for example, as effective barriers to preventing the migration of oil and petroleum products into groundwater.

Carbon-carbon sorbents (one of the components of which, pine sawdust carbonate (basic), as it has the best sorption properties) have better sorption and absorption properties, relative to the corresponding indicators of the materials that are part of them.

Conclusions to Chapter 13

1. Oil content and absorption capacity of petroleum products of pine sawdust carbonate: expanded graphite, at a ratio (within 50:50%), 2.5-3.0 times higher than pine sawdust carbonate. Its use is the most promising and cost-effective in removing petroleum products from the environment and preserving environmental infrastructure.

2. Carbon-carbon materials should be used as effective sorbents to remove petroleum products from the water surface and soil, and carbon-mineral as effective barriers to prevent the migration of oil and petroleum products into groundwater.

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CHAPTER 14. RESEARCH OF INTERACTION PROCESS OF SHANKS OF CONCAVE DISC SPRINGS OF TILLAGE MACHINES

Introduction

The interaction process of an elastically fixed concave disk [1] and the soil can be generalized according to a system based on parallel-opposite connection of two parts [2]: elastic fixing that characterizes elastic deflections of an operating element [3] and draft force energetics [4] of the technological process (figure 14.1). The uncertain influence of a soil medium is generally viewed according to two components regular [5] and random [6] that are transmitted to a shank as load [7]. The additive effect of soil heterogeneity determines the external influence on the interaction process of a disk operating element on a spring shank [8] and a soil medium [9]. It is sizing up the impact of the design parameters of shanks on the motion stability of a disk operating element in a vertical plane under the action of the components R_z and R_x of soil drag [10], the dependences of elastic deflections δ_z were obtained with the use of Maxwell-Mohr method [11]. For a coil spring shank [12]: $\delta_z = \delta_{zR_z} + \delta_{zR_x}$, where δ_{zR_x} – the deflection of a shank from the horizontal component of drag forces, mm; δ_{zR_z} – the deflection of a shank from the vertical component of drag forces, mm. For a C-shape shank [13]: $\delta_z = (2 \cdot E \cdot I)^{-1} \cdot r^3 \cdot (\pi \cdot R_z + 4 \cdot R_x)$, where r – shank bending radius, mm; E – elasticity modulus (for steel $E = 2 \cdot 10^5$ N/mm); I – shank section inertia moment, mm^4. For a helical shank [14]:

$$\delta_z = 2.355 \cdot (E \cdot I)^{-1} \cdot F_{dr} \cdot r^3 + sin\big(arctg\big\{2.355 \cdot [E \cdot I \cdot l^2]^{-1} \cdot F_{dr} \cdot l` \cdot r^3\big\} + \{E \cdot I\}^{-1} \cdot F_{dr} \cdot l`\big).$$

The movement of a concave disk in the reference system [16], which is rigidly connected with a unit frame [17] and moves translationally together with [18], it is a three-dimensional oscillation about an equilibrium position [19]. Such oscillations take place under the action of the moments applied to the mechanical system that consists of a spring shank [20], a bearing unit [21], a disk [22] and the soil on a disk [23] (here after the system "soil – disk – spring shank"). These moments include: the moment of elastic forces [24]; the moment of attractive forces [25]; the moment of disk drag reaction forces [26].

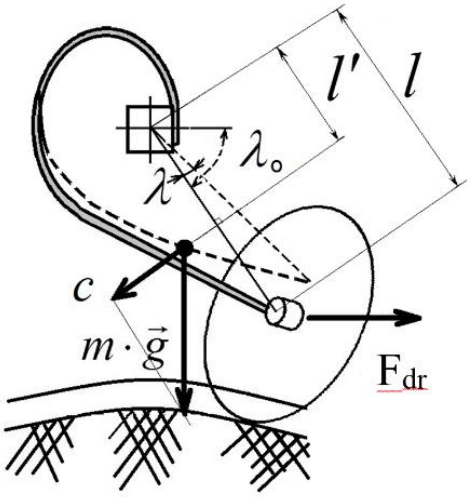


Fig. 14.1 Computational scheme of the dynamic model: λ – generalized coordinate, deg.; c – coefficient of shank rigidity, N/m; $m \cdot \vec{g}$ – weight force (reduced mass), N; F_{dr} – soil drag force, N; l – reduces shank length (a line from a reference-starting point, which is rigidly connected with a unit body and moves translationally together with it to the point of intersection with the axis of disk rotation in a vertical plane [15]), m; l' – reduced length distance to the intersection with the perpendicular to l from the center of shank and disk weight, m.

Purpose of research

Providing reasonable accuracy for practical purposes, the movement of a concave disk on a spring shank can be presented as one-degree-of-freedom movement that is explicit due to the deviations of a spring shank. Such movement is described with the help of one generalized coordinate λ, which is the angle of shank deviation from the position it takes in the state of statistic equilibrium.

Materials and methods

The movement of the oscillating system "soil – disk – spring shank" (figure 14.2) is uniquely defined if the generalized coordinate is expressed as the function of time.

The moment with respect to the mount point of a shank and a frame bar, taking into account the resultant of the elastic forces caused by unit weight on a shank and the external influence (soil drag), is equal to:

$$M_p = c \cdot l^2 \cdot sin\lambda,$$

where c – coefficient of shank rigidity, N/m.

During shank movement, when the parameter λ is different from the initial one, weight force is equal to $m \cdot \vec{g}$ and its perpendicular to the beam drawn along the reduced shank length l to the center of weight $l^{`}$ is equal to $m \cdot g \cdot cos(\lambda_0 + \lambda)$ (on the theorem of the angles with mutually perpendicular sides). The moment of weight forces relative to the mount point of a shank and a frame bar is equal to:

$$M_v = l^{`} \cdot m \cdot g \cdot cos(\lambda_0 - \lambda).$$

The principal moment of disk drag forces:

$$M_s = -l \cdot F_{dr}(t, V) \cdot sin\lambda,$$

where $F_{dr}(t, V)$ – soil drag force (spring shank load), N.

Fig. 14.2 The movement of the oscillating system “soil – disk – spring shank”.

Based on d'Alembert's principle, the differential equation of spring shank movement was obtained:

$$S \cdot \ddot{\lambda} = -c \cdot l^2 \cdot sin\lambda + l^{`} \cdot m \cdot g \cdot cos(\lambda_0 - \lambda) - l \cdot F_{dr}(t, V) \cdot sin\lambda, \quad (14.1)$$

where S – inertia moment relative to the mount point of a shank and a frame bar.

Having divided the two parts of the equation in (14.1) by S and having denoted the coefficients in the obtained equation at $cos\lambda$ and $sin\lambda$ by A and B, we obtain the equation (14.1):

$$\ddot{\lambda} = A \cdot cos\lambda + B \cdot sin\lambda, \qquad (14.2)$$

The non-linear differential equation (2) is not possible to solve using elementary functions, thus, let us handle the solution with the help of Jacobian elliptic functions.

Let us develop the new required function $\psi = \psi(t)$, assuming $\dot{\psi} = \ddot{\lambda}$. Then:

$$\ddot{\lambda} = d\psi/dt = (d\psi/d\lambda) \cdot (d\lambda/dt) = \psi \cdot (d\psi/d\lambda), \qquad (14.3)$$

Taking into account (14.2) and (14.3), the function $\psi = \psi(t)$ satisfies the equation:

$$\ddot{\lambda} = \psi \cdot (d\psi/d\lambda) = A \cdot cos\lambda - B \cdot sin\lambda, \qquad (14.4)$$

Thus, the second-order differential equation (14.2) was reduced to the first-order equation (14.4), which could be completely integrated against elementary functions. In order to determine the physics of the equation (14.3), let us apply the function, which is defined by the equality:

$$U = -(A \cdot sin\lambda + B \cdot cos\lambda), \qquad (14.5)$$

The function $U = U(\lambda)$ depends on the coordinate λ and it is also a potential energy. From (14.4) and (14.5) we obtain:

$$0.5 \cdot \psi^2 + U = const. \qquad (14.6)$$

The equality (14.6) shows that the sum of potential and kinetic energy is constant and it is the first integral of spring shank motion equations. Thus, the chosen dynamic model is validated.

Let us insert the parameter μ_0 in order to determine the initial oscillation phase, which is defined by the initial conditions and is expressed as: $\mu_0 = arctg(A \cdot B^{-1})$, $0 \le \mu_0 < \pi/2$.

Thus: $\psi^2 = 2 \cdot B \cdot sec\mu_0(cos\{\mu_0 - \lambda\} - cos\mu_0) = 4 \cdot B \cdot sec\mu_0(sin^2 \mu_0/2 - sin^2 \{\mu_0 - \lambda\}/2)$.

Let us insert the new function $\varepsilon = \varepsilon(t)$ and find $\dot{\varepsilon} = \dot{\lambda} = \psi$ based on (14.1). It follows that:

$$-\mu_0 \leq \varepsilon \leq \mu_0.$$

Thus, there is oscillating motion pattern and the amplitude of oscillations is equal to $2 \cdot \mu_0$. Let us insert the new required function $\eta = \eta(t)$, that is related with $\varepsilon = \varepsilon(t)$ by:

$$sin\, \varepsilon/2 = k \cdot sin\eta. \qquad (14.7)$$

Results and discussion

Having analyzed the expression (14.7), the following conclusions can be drawn: the function $\eta = \eta(\varepsilon)$ is continuous in the segment $[-\mu_0; \mu]$; if $\varepsilon \to \mu_0 - O$, then $\eta^{`} \to \infty$, here, $\eta^{`}(\varepsilon) = O \cdot (\mu_0 - \varepsilon)^{-1/2}$ for $\varepsilon \to \mu_0 - O$, where O – Landau symbol; analogously, if $\varepsilon \to -\mu_0 + O$, then $\eta^{`} \to \infty$, here, $\eta^{`}(\varepsilon) = O \cdot (-\mu_0 + \varepsilon)^{-1/2}$.

According to the previous dependences, it is determined that if there is an increase of ε from $-\mu_0$ to μ_0 (if there is a decrease from μ_0 to $-\mu_0$, respectively), the function $\eta = \eta(\varepsilon)$ increases (decreases) from $-\pi/2$ to $\pi/2$ (from $\pi/2$ to $-\pi/2$).

The whole process timespan can be divided into the complementary intervals $]t_{i-1}, t_i[$, $i \in N$, during each interval the direction of spring shank movement is not changed, that is, during each interval there is only concave disk deepening or lifting from the soil.

For any $n \in N$ the sign is «+» at $t_{2n-1} < t < t_{2n}$ and it is «–» at $t_{2n} < t < t_{2n+1}$. Let us determine the law of motion of a disk on a spring shank in operation. For this purpose, let us choose the arbitrary t that meets the condition (n-arbitrary): $t_{2n} < t < t_{2n+1}$.

Let us apply the elliptic sine from the theory of elliptic functions, that is denoted by sn and is determined from the equality $snu = sin(amu)$. Then:

$$sin\eta = snC_0(t - t_0), \qquad (14.8)$$

and applying $-\mu_0 \leq \varepsilon \leq \mu_0$ and (14.8) it follows that

$$\lambda = \mu_0 + 2 \cdot arcsin(k \cdot snC_0\{t - t_0\}), \qquad (14.9)$$

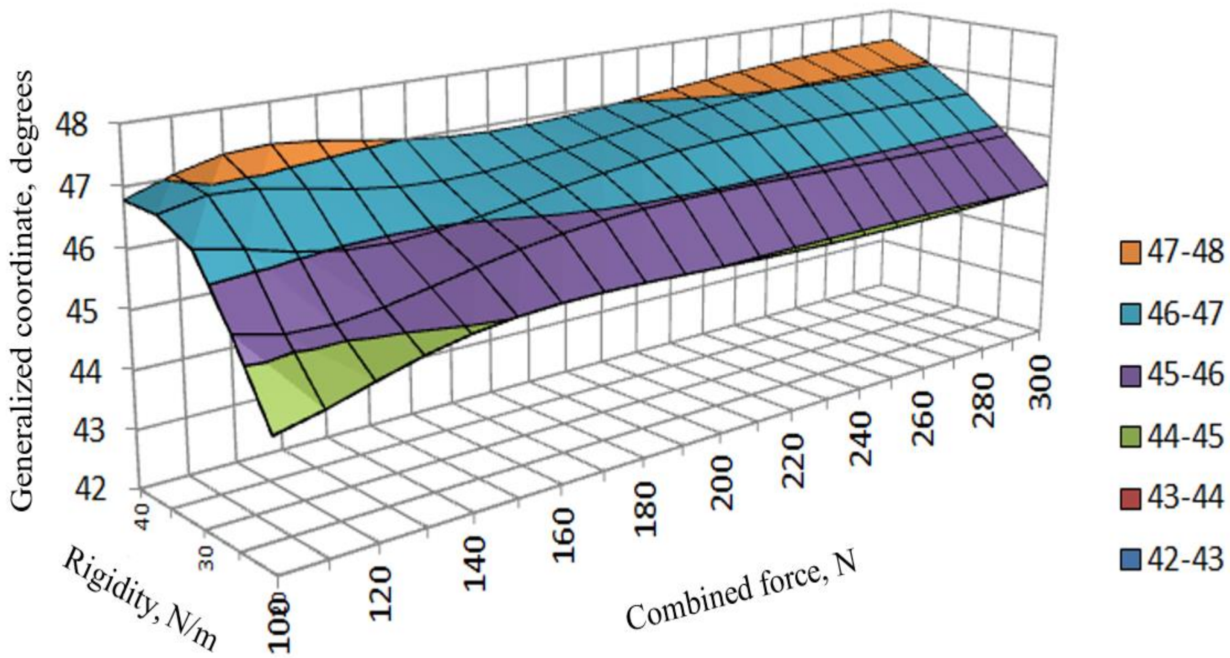
The equality (14.9) describes the law of motion of the system "soil – disk – spring shank" during the process of soil tilling and shows that the oscillations are not harmonic, not stochastic, their period is $4 \cdot C_0^{-1} \cdot k$, their amplitude is μ_0. According to the analytical equation (14.9), the characteristic curves (figure 14.3 and figure 14.4) were obtained.

According to spring shank oscillations, the quality index of disk header process performance was observed – the steadiness of its running, according to the depth of an operating element, that can be provided on condition:

$$H \cdot (1 - sin\lambda) \leq \Delta, \tag{14.10}$$

where Δ – agrotechnical tolerance on steady running, according to the depth of an operating element (for disks the mean square deviation is equal to 15 mm); H – vertical distance between a horizontal plane at the mount point of a spring shank and a unit frame to the furrow bottom of the cultivated soil, mm.

The exceedance of the acceptable value of oscillation amplitude of the system breaks the steadiness of soil cultivation.



Figu. 14.3 Change of the generalized coordinate depending on the value of the reduced mass and the rigidity of a spring shank.

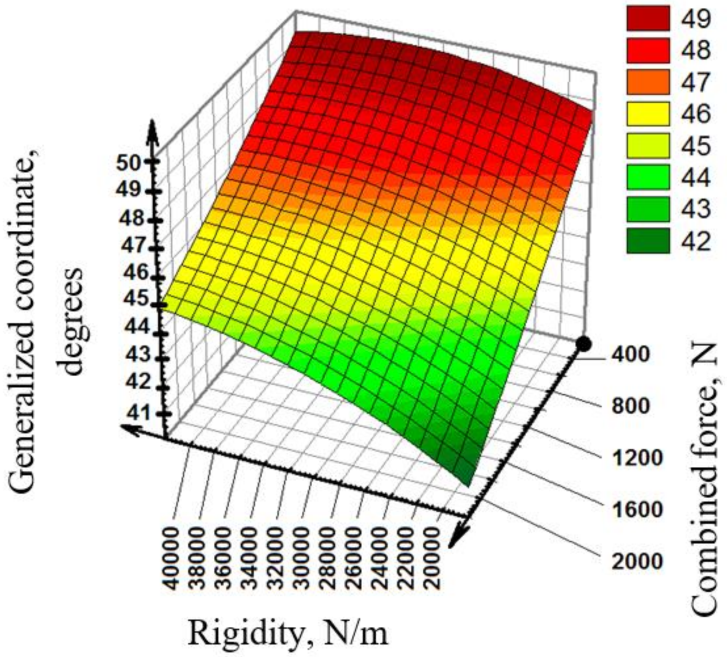


Fig. 14.4 Change of the generalized coordinate depending on load value.

The mathematical model has been developed and a theoretical investigation on the change in the drag forces of a disk operating element on a spring shank has been conducted. The dynamic characteristics of a spring shank in the course of unit operation have been determined within the range of the rigidity variation limits from 20 kN/m to 40 kN/m, the reduced mass limits from 100 N to 300 N, the draft force variation limits from 200 N до 2000 N.

Conclusions to Chapter 14

The structural diagram of the interaction process of a concave disk and the soil, where oscillations of an operating element on a spring shank with a certain amplitude and frequency create a feedback in the system “soil – disk – spring shank”, has been suggested.

The mathematical model has been developed and a theoretical investigation on the change in the drag forces of a disk operating element on a spring shank has been conducted. The dynamic characteristics of a spring shank in the course of unit operation have been determined within the range of the rigidity variation limits from 20 kN/m to

40 kN/m, the reduced mass limits from 100 N to 300 N, the draft force variation limits from 200 N до 2000 N.

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CHAPTER 15. CONCEPTUAL BASES OF SYSTEM TECHNOLOGY OF DESIGNING OF LOGISTIC SCHEMES OF HARVESTING AND TRANSPORTATION OF GRAIN CROPS

Introduction

The probabilistic nature of the harvesting and transport process causes downtime of combines and vehicles during the harvest of cereals [1] and the use of direct transportation [2]. Thus, with a distance of 8-9 km [3] and optimal combinations of the number of harvesters and vehicles idle harvesters waiting for transport reach 20% [4], and downtime of vehicles – 30-36% of the shift time [5]. The use of trucks (N3 wheeled vehicles) on direct grain transportation [6], the loss of time associated with waiting for loading and moving across the field [7], increases to 47% of the shift time [8]. In addition, the use of trucks leads to additional soil compaction [9]. The use of an intermediate compensation link in the technological chain between combines [10] and vehicles allows to significantly reduce the duration of harvesting [11] and transport operations in comparison with direct road transport [12]. The role of interoperable compensators is performed by tractor trailers [13], interchangeable bodies [14] and other devices [15]. The use of compensators, first, allows you to organize the work so that the harvesters can be unloaded immediately after filling the hopper [16], and the vehicle – to be loaded on arrival at the field [17]. Secondly, such compensators as reloading trailers have the ability to solve the problem of eliminating compaction in the ground by heavy trucks that are effective in transporting grain [18]. Theoretical studies of the use of interoperative compensators were initiated in works [19, 20], where the main patterns of their use were considered. However, this problem requires further study in relation to the natural production conditions of farms, areas, grain yields, road conditions and distances and, thus, becomes systemic in nature, which, of course, requires appropriate system-analytical tools to solve it [21]. In the process of designing flexible logistics technologies for grain harvesting complexes, we used the scientific and analytical conclusions of previous studies on these issues [22, 23]. Since logistics technology is considered by us as a complex system, the problem of designing such technology and formulating criteria for its effectiveness is expected to be solved using the methodology of systems analysis [24] and in particular using inductive technologies of systems information-analytical research [25]. At the stages of modeling, in particular the classification of technologies, construction of statistical and econometric models, the application of modern methods of inductive modeling of complex systems described in the numerous scientific literature, in particular [26, 27].

The aim of research is to build a conceptual position on the creation of technology for the design of transport and logistics schemes for harvesting, transshipment and transportation of grain crops with flexible adaptation to specific agronomic, technical, natural, economic and other numerous conditions during their harvest. The result of system-analytical research of the problem should be a practical consulting document that would allow the decision-maker (decision maker) in specific conditions to determine the choice of optimal transport and logistics schemes in order to most effectively perform the stages of harvesting and transporting crops to granaries.

Materials and methods

The problem of creating a flexible logistics technology for harvesting and transporting grain crops during their harvest will be considered as a complex system, a complex object to be studied. The result of solving this problem will be the creation of a comprehensive document of information and recommendation (consulting) nature, which would contain certain mandatory sections and their content in relation to the flexible choice of a set of technical means with optimal economic performance, parameter groups, etc. depending on specific natural, economic, legislative and other numerous conditions during the harvest of cereals.

Since there is no sufficient information base for the content of such a document at the beginning of such a comprehensive study, in our opinion, the solution of this system problem should be performed from the standpoint of inductive approach to modeling complex systems, which has proven itself in solving many complex uncertain problems.

One of the effective tools of this approach to the creation of this type of document is the inductive technology of system information-analytical research, which applies the main principles of the theory of inductive modeling of complex systems. It will be recalled that the result $R^*(I_b^*)$ of complex inductive technologies of system information-analytical research is a specific document $D\{R^*(I_b^*)\}$, which reflects the results of system subject analysis of a complex object (process, phenomenon or problem in general), which is based on the optimal information base I_b^* constructed in the research process. Requirement has an information-recommendatory (consulting) nature, endowed with a certain official status and level of access. $D\{R^*(I_b^*)\}$ means a document prepared and executed in accordance with the requirements, based on a set of optimal research results $\{R^*(I_b^*)\}$, which may still have a certain sketch character. This is the only difference between the results $R^*(I_b^*)$ and the resulting document $D\{R^*(I_b^*)\}$.

Results

Here are the main stages that must be passed in accordance with the inductive technology of system information and analytical research to create a meaningful document.

Stage I. Formation of the top-level expert commission and construction of the primary information base, selection of analytical groups *A* and *B* to perform system-analytical research and synthesis of the optimal information base I_b^*, which should generate results $R^*(I_b^*) \in \{R^*(I_b^*)\}$.

Stage II. Creating a matrix of reference (target) result:

$$E = E\{R^0(I_b^0)\} = (e_{ij}) = \begin{pmatrix} e_{11} & \cdots & e_{1n} \\ \vdots & \ddots & \vdots \\ e_{m1} & \cdots & e_{mn} \end{pmatrix}, \quad (15.1)$$

in which the *i*-th line, $i = 1, 2, \ldots, m$, reflects one of the indisputable types of requirements for the target result of the study from the standpoint of the expert commission, and the *j*-th column, $j = 1, 2, \ldots, n$, possible gradations of estimates of the *i*-th element. The line in our case is one of the necessary sections of the future document, and the elements of the line are the formalized values of expert assessments (requirements) to its subdivisions. The elements (e_{ij}) of the matrix $E\{R^0(I_b^0)\}$ are formalized according to a certain algorithm based on the assessments of top-level experts. For example, this may be the median on the set of expert tolerances (conclusions) regarding such an element (e_{ij}).

It is important to note that the matrix $E\{R^0(I_b^0)\}$ concerns only the form of the future result and the importance of reflecting in it the most important positions and their parts. In addition, in contrast to the known analytical technologies such as the Delphi method, the key point is that members of the top-level expert commission who agreed with the generalized estimates of matrix $E\{R^0(I_b^0)\}$ tolerances (e_{ij}) cannot change their conclusions about the shape of future results throughout the project. That (e_{ij}) is, there are constant estimates. The semantic content of all these positions is performed by analytical groups and tested by experts from the top-level expert commission at each step of the next stage III.

Stage III. Execution of information-analytical project by iterative procedure.

Step 1 of stage III. Groups A and B synthesize analytical results $R_k(I_b^1)^{(A,B)}$, $k = 1, 2, \ldots, K$, which include only the initial information base I_b^1, and for each such result, experts make estimates, ie build matrices that are formalized on the same principle as for the matrix $E\{R^0(I_b^0)\}$.

$$W_k^{(A,B)} = (w_{ij}) = \begin{pmatrix} w_{11} & \cdots & w_{1n} \\ \vdots & \ddots & \vdots \\ w_{m1} & \cdots & w_{mn} \end{pmatrix}, \quad (15.2)$$

That is, the matrix W^A – reflects the formalized *k*-th result $R_k(I_b^s)$, achieved by the analytical group *A* in the *s*-th step of the study ($s = 1, 2, \ldots, S$) and, accordingly W^B –

reflects the formalized k-th result $R_k(I_b^s)$ achieved by the analytical group B in the same s-th step of the study.

Each synthesized result is evaluated according to the criteria of systemic correlation:

$$CR_{corel} = \sqrt{\sum_{i=1}^{m}\sum_{j=1}^{n}\left(\delta_{ij}^2\right)_{W^A W^B}}, \tag{15.3}$$

and systemic relevance:

$$CR_{rel} = \sqrt{\sum_{i=1}^{m}\sum_{j=1}^{n}\left(\delta_{ij}^2\right)_{W^{(A \wedge B)}E}}, \tag{15.4}$$

where δ_{ij}^2 – the elements of the matrices

$$\Delta_{(*)}^2 = \begin{pmatrix} \delta_{11}^2 & \cdots & \delta_{1n}^2 \\ \vdots & \ddots & \vdots \\ \delta_{m1}^2 & \cdots & \delta_{mn}^2 \end{pmatrix}, \tag{15.5}$$

and are equal to the squares of the differences of the corresponding elements of the pairs of matrices $W\{R_k(I_b^s)\}^{(A \wedge B)}$ and $E\{R^0(I_b^0)\}$ and $W\{R_k(\delta_{ij}^2)\}^A$, $W\{R_k(I_b^s)\}^B$, respectively, and all these matrices have dimensions $n \times m$. The sign * means to which pairs of matrices, and therefore to which criterion, the object belongs δ_{ij}^2 (relevance or correlation).

Therefore, these system criteria require minimal differences both between the achieved two groups of results (15.2) and between such results and the reference. Then the information monitoring system is activated, an additional target portion I_b^+ of monitoring information is formed, which should complement the already existing ensemble I_b^s, $s = 1, 2, \dots, S$, in order to improve the results $R_k(I_b^s)$, bringing them closer to the reference $R^0(I_b^0)$.

Step 2 of stage III,…, S. Analytical results $R_k(I_b^s)$, $k = 1, 2, \dots, K$, $s = 1, 2, \dots, S$ are synthesized, which are based on the results of previous steps and information I_b^+ of purposeful information monitoring. Again, each synthesized result is evaluated by criteria of systemic correlation (15.3) and systemic relevance (15.4).

Criteria (15.3) and (15.4) should theoretically have minima, which indicate the cessation of the inductive procedure for the synthesis of the optimal result $R^*(I_b^*)$ (or a certain limited set of such results $\{R^*(I_b^*)\}$). The last step of the procedure of inductive technologies of system information-analytical research is one in which:

- the result is obtained, which is objectively the best according to the system criteria (15.3) and (15.4) and satisfies the customer;
- the result is obtained, which can still be improved, but it already satisfies the customer;
- exhausted research resources (time and money, for example).

Stage IV. Formation of the optimal result $R^*(I_b^*)$ and the corresponding consulting document $D\{R^*(I_b^*)\}$. Both analytical groups should already work here under the control of the project manager.

Stage V. Protection of the document $D\{R^*(I_b^*)\}$ and its transfer to the customer. Obviously, the described technology does not require complex, long-term and costly field experiments to achieve the result (although in the general case it is not denied). In addition, it is also obvious that such an approach should be used to solve complex problems, to implement intelligent projects of high complexity. The problem undoubtedly belongs to this class of problems. Thus, comparing the above, a consulting document that would contain simple and at the same time sufficient for decision-making procedures (schemes) for the synthesis of flexible logistics technologies for harvesting and transporting grain to granaries and optimally adapted to specific natural production, economic, technical and other conditions during the collection period, it is advisable to create, based on the described inductive technology of system information and analytical research. The concept of developing such a document for the synthesis of logistics technologies for harvesting and transporting grain involves the following stages.

Step 1 of stage V. Creation of a matrix of the target (reference) result $E\{R^0(I_b^0)\}$ and the primary information base I_b^1 by the previously formed expert commission of the top level. As a result of the substantive analysis of the problem and, based on previous studies, the experts proposed conclusions that allowed to obtain the following formalized matrix of the target result $E\{R^0(I_b^0)\}$:

$$E\{R^0(I_b^0)\} = \begin{pmatrix} 9&9&7&8&7&8&5&4&0 \\ 7&8&7&8&7&7&5&0&4 \\ 6&8&8&8&6&5&6&0&0 \\ 9&7&5&6&9&8&6&7&8 \\ 9&8&9&9&9&9&9&7&8 \\ 8&7&8&8&8&8&7&8&0 \\ 9&8&9&9&9&9&8&8&0 \\ 3&3&2&2&3&1&1&1&0 \end{pmatrix}, \quad (15.6)$$

Table 15.1

The primary information base in the project of synthesis of flexible.

Number and names group	Parameter in group		
	number	symbol	name and its unit of measurement
I. Combine harvester	I.1	W_k	Nominal productivity, t/h
	I.2	N_k	Engine power, kW
	I.3	G_k	Fuel consumption, kg/(kW year)
	I.4	C_k	The cost of the combine, UAH
	I.5	V_k	Capacity of the grain bunker, m^3
	I.6	$W_к$	Productivity of unloading grain auger, t/h
II. Reloader trailer	II.1	V	Capacity of the loader trailer, m^3
	II.2	N_n	Tractor engine power, kW
	II.3	g_p	Specific fuel consumption, kg/(kW h)
	II.4	C_n	Cost of the trailer of the reloader, UAH
	II.5	W_n	Productivity of the unloading auger of the trailer-reloader, t/h
III. Wheeled vehicle of category N3	III.1	g_a	Load capacity of the vehicle, t
	III.2	$Q_н$	Standard fuel consumption, l/(100 km)
	III.3	C_a	Vehicle cost, UAH
	III.4	v_a	Technical speed of the vehicle, km/h
	III.5	D_a	Fuel type (diesel, petrol, gas, biodiesel, biogas)
IV. Natural	IV.1	K_p	Coefficient of complexity of natural

and			production conditions
production	IV.2	K_o	Weather coefficient
conditions	IV.3	T_a	Agroterms (harvesting period), days
	IV.4	U	Yield, t/ha
	IV.5	S	Field area, ha
	IV.6	L	Length of the run, m
	IV.7	l	Distance of transportation, km

The first column of the matrix (15.6) shows agreed estimates of the importance of a section of the final document. The following columns reflect the experts' assessments of certain group parameters from the optimal information base I_b^*. The first four rows of the matrix (15.6) to some extent coincide with the groups of the primary information base. From the 5th line to the 8th – the requirements of experts from the expert commission of the top level concerning indisputably necessary processing of the following sections of the final document (on lines) are reflected:

- 5th – schemes of synthesis of harvesting and transport technological processes;
- 6th – technical and economic analysis of optimal harvesting and transport technological processes;
- 7th – forecasting the economic efficiency of the selected by the accident harvesting and transport processes;
- 8th – appendices (instructions, nomograms, numerical information, etc.).

The experts included and classified the following parameters into groups in the primary information base I_b^1 (table 15.1).

Step 2 of stage V. Formation of analytical groups *A* and *B*, which should perform system-analytical research in the direction of content filling of matrices $W\{R_k(\delta_{ij}^2)\}^A$, $W\{R_k(I_b^s)\}^B$ and in order to achieve the optimal result $R^*(I_b^*)$ (or a certain limited set of such results $\{R^*(I_b^*)\}$) and minimize criteria (3) and (4). Execution of the project in parts of rows (lines) 5-th to 7-th of the matrix $E\{R^0(I_b^0)\}$ to obtain numerous analytical models and dependencies involves the use of inductive methods for modeling complex systems.

Step 3 of stage V. Formation of the source document $D\{R^*(I_b^*)\}$ and its transfer to the customer.

Discussion

The article presents the conceptual principles of creating a system technology for designing logistics schemes of harvesting and transport technological processes of schemes in a very short period of harvesting and transportation of grain crops [28]. Such schemes are obviously technology in turn. Despite the conceptual nature of the work, it already contains at least the results of the first stage of inductive technologies of system information and analytical research, because, based on the complexity and significant lack of initial information base, the project is expected to perform such inductive technology [29]. This was done due to the already conducted theoretical studies of the processes of application of interoperative compensators, which obtained the basic patterns of their use [30].

Based on the agreed conclusions of top-level experts, a matrix of the target result (document) is synthesized, which should become a clear guide in the form of further system-analytical research in the direction of semantic content of the final document. Such a document, according to experts, should allow a person who makes decisions that are optimal according to the criteria constructed in the research scheme of harvesting and transport processes in a short time of harvesting and transport work at harvest [31]. The concept stipulates, and this is reflected in the matrix of the target result, that the effectiveness of information base schemes in the project of flexible synthesis should be evaluated both by technical parameters and by economic factors [32].

Conclusions to Chapter 15

The matrix $E\{R^0(I_b^0)\}$ concerns only the form of the future result and the importance of reflecting in it the most important positions and their parts. In addition, in contrast to the known analytical technologies such as the Delphi method, the key point is that members of the top-level expert commission who agreed with the generalized estimates of matrix $E\{R^0(I_b^0)\}$ tolerances (e_{ij}) cannot change their conclusions about the shape of future results throughout the project.

Creation of a matrix of the target (reference) result $E\{R^0(I_b^0)\}$ and the primary information base I_b^1 by the previously formed expert commission of the top level. As a result of the substantive analysis of the problem and, based on previous studies, the experts proposed conclusions that allowed to obtain the following formalized matrix of the target result $E\{R^0(I_b^0)\}$.

Formation of analytical groups *A* and *B*, which should perform system-analytical research in the direction of content filling of matrices $W\{R_k(\delta_{ij}^2)\}^A$, $W\{R_k(I_b^s)\}^B$ and in order to achieve the optimal result $R^*(I_b^*)$ (or a certain limited set of such results $\{R^*(I_b^*)\}$) and minimize criteria (15.3) and (15.4).

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CHAPTER 16. RESEARCH OF MACHINES FOR MULCHING NEAR-TRUNK STRIPS IN PERENNIAL FRUIT PLANTATIONS

Introduction

The most common methods of soil retention in gardens are tillage between rows and black steam in the stem strips 1.0-1.6 m wide [1]. Application of the latter in the first 2 years after planting [2] is recommended to carry out mechanical tillage [3], and then use herbicides 2-3 times per season [4]. However, the use of herbicide vapor, along with the positive aspects, is characterized by a number of significant disadvantages: rapid loss of moisture in the dry season from an unprotected surface [5]; formation of a surface crust after rain [6], which prevents the enrichment of the soil with air [7]; the need for regular fertilization of plants with mineral fertilizers due to the lack of replenishment of nutrients due to the decomposition of plant residues [8], which occurs, for example, when using mechanical means of cultivation [9]; pollution of the environment with chemicals that [10], falling on the leaves of trees [11], can inhibit their development [12], and accumulate in the fruit, harm the health of consumers [13].

Today, an alternative way to maintain the soil in optimal condition in terms of plant physiology, reduce environmental pollution and improve fruit quality is to cover the interstitial strips with mulch substrate [14]. The material for this can be sawdust, shavings, flax trusts, chopped branches, grass, straw or a mixture thereof [15], used fungal substrate, litter for keeping animals [16]. Mulch reduces moisture evaporation [17], protects plant roots from freezing in winter, improves soil structure, enhances microbiological processes in it, reduces daily temperature fluctuations [18], inhibits weed germination and as a result improves the marketable quality of fruits [19], has a positive effect on their storage [20]. One of the reasons holding back the spread of this technology is the lack of special means of mechanization for its implementation [21].

The aim of research is to increase the productivity and quality of mulching substrate application in the stem strips of perennial orchards by developing an effective means of mechanization and substantiation of the optimal parameters of its work during this operation.

Materials and methods

Laboratory field experiments were carried out taking into account agrotechnical technological and technical and economic requirements [22], a priori information and conclusions made on the basis of engineering calculations [23]. Processing and analysis of the obtained data were performed using the methods of mathematical statistics [24].

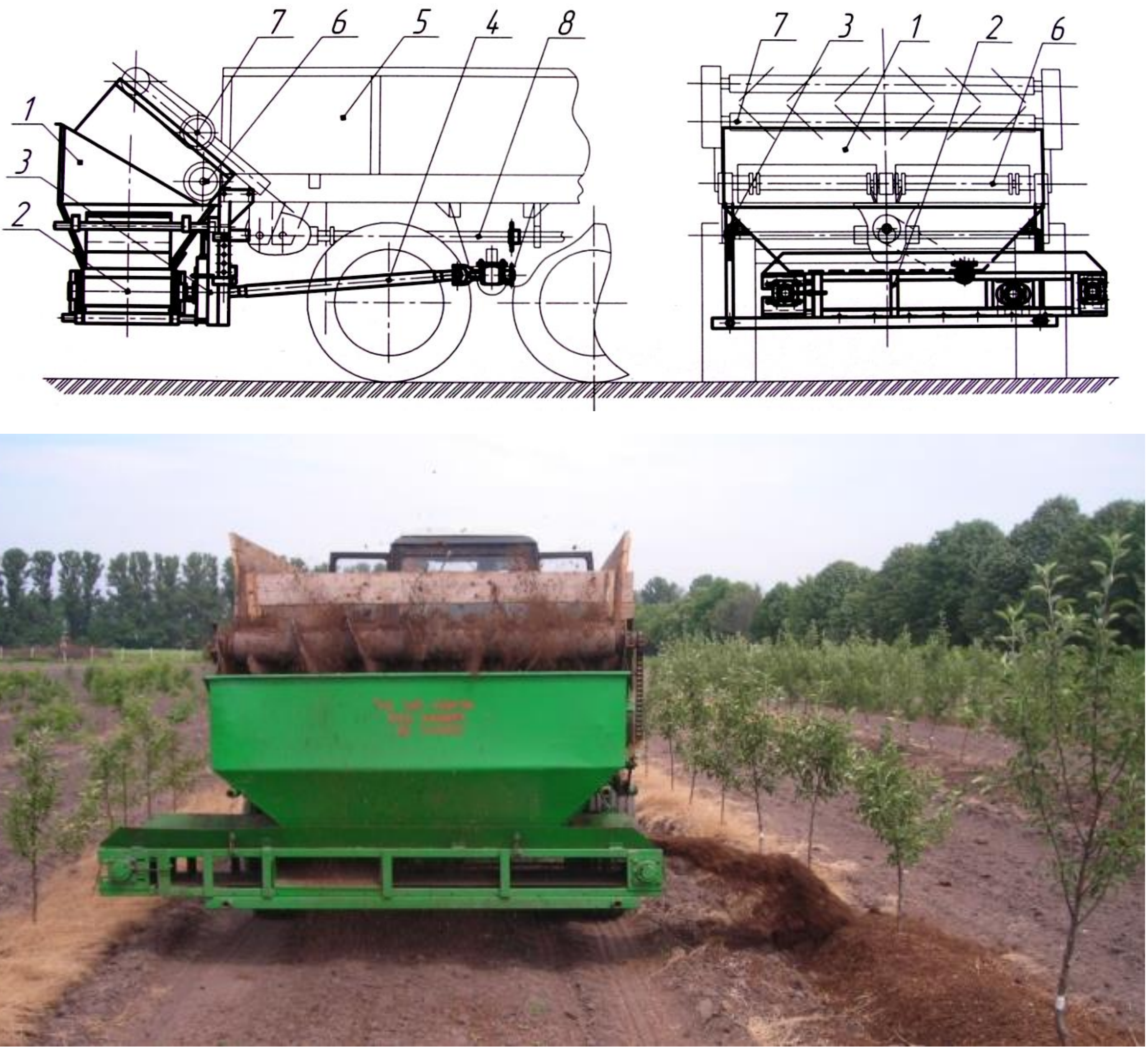

Fig. 16.1 Schematic representation and model of the machine for making mulch substrate in the stem strips of perennial orchards.

The program of experimental researches provided definition: productivity of nutritious and cross conveyors [25], geometrical parameters of the last, distance of laying of mulch in a roll depending on height of its placement over a soil surface and speed of rotation of a drive drum [26], norms of mulch introduction at change of a mode of operation of the unit and its components [27].

To conduct research, a prototype machine was developed and manufactured, which is mounted on a spreader of solid organic fertilizers type RS-6M (figure 16.1). The machine (figure 16.1) consists of a receiving hopper *1*, a belt cross conveyor *2*, a hanging device *3*, a torque transfer mechanism *4* and auxiliary equipment, and a spreader – a body *5*, a longitudinal chain-bar (feed) conveyor with a drum of its displacement *6*, a crushing and dosing mechanism *7* and the mechanism for actuating the working bodies *8*.

The design of the hanging device allows to hang and dismantle the machines without special efforts that does the spreader more universal in use. During the operation of the unit, the mulch is fed by a feed conveyor through the crushing and dosing mechanism into hopper of machine. Then the crushed mass is moved by belt conveyor to the area of tree trunks where a roll of the required size is formed from it.

Results

The analysis of the obtained results shows that the productivity of the feed and cross conveyors can vary in the range from 1.2 to 9.0 m^3/min and from 2.0 to 5.3 m^3/min. Accordingly, when increasing the speed of rotation of the power take-off shaft of the tractor, as well as when moving the ratchet regulator to a higher mark (the first conveyor) and change the position of the dosing valve (second) (figure 16.2 and figure 16.3). Therefore, to prevent the undesirable excessive accumulation of substrate in the receiving hopper of the machine, the productivity of the feed conveyor should not exceed 5 m^3/min, which is possible with the appropriate position of the ratchet mechanism.

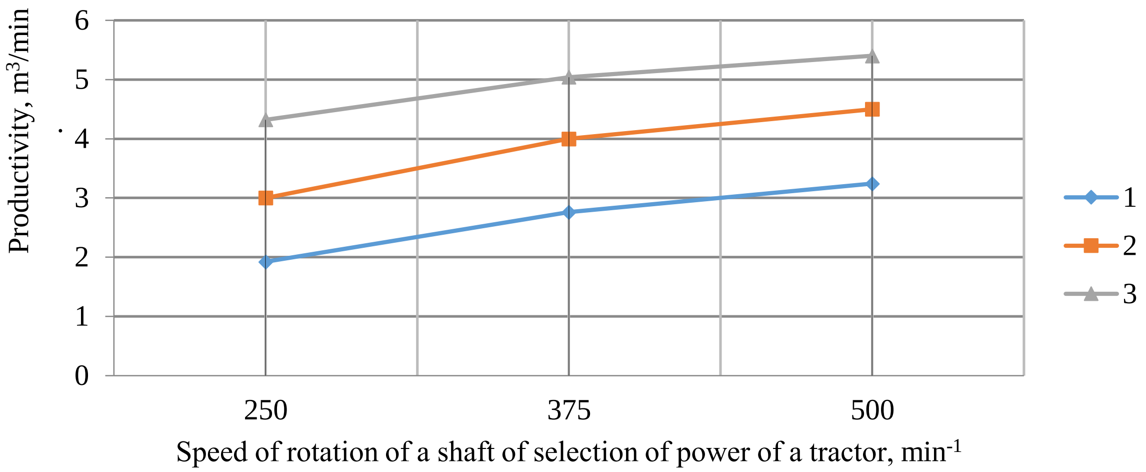

Fig. 16.2 Change of productivity of the cross conveyor of the tractor depending on position of a dosing gate and speed of rotation of a shaft of selection of power of a tractor: curves 1, 2, 3 – positions of change of position of a gate.

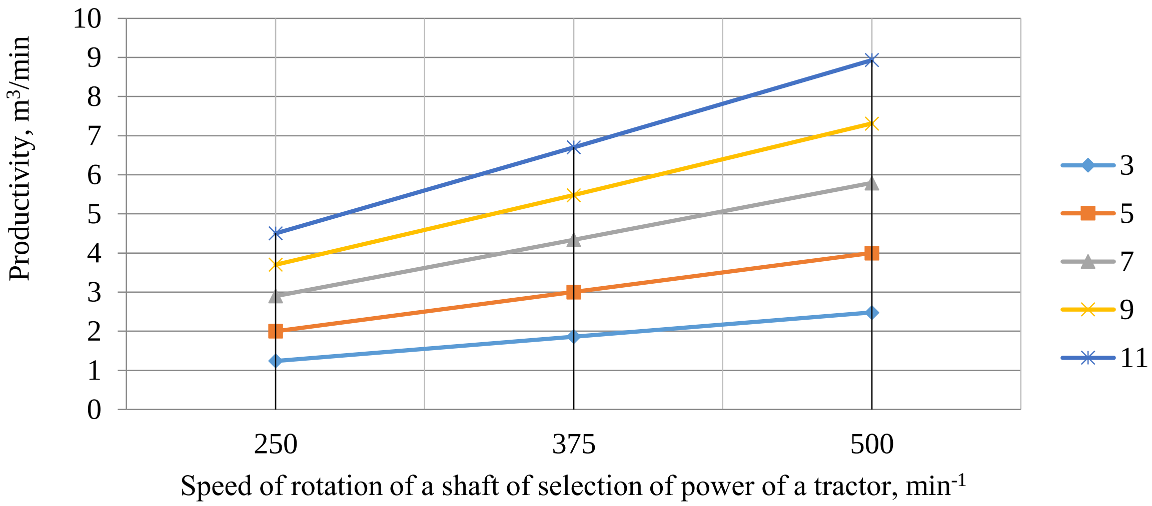

Fig. 16.3 Change of productivity of the feeding conveyor of the basic tractor depending on position of the ratchet mechanism and speed of rotation of a shaft of selection of power of a tractor: curves 3, 5, 7, 9, 11 – position of an eccentric of the ratchet mechanism.

The distance of laying mulch in the roll relative to the transverse conveyor (figure 16.4 and figure 16.5) can vary from 0.40 to 1.30 m depending on the height of the latter above the soil surface and the speed of rotation of the shaft of selection of power of a tractor.

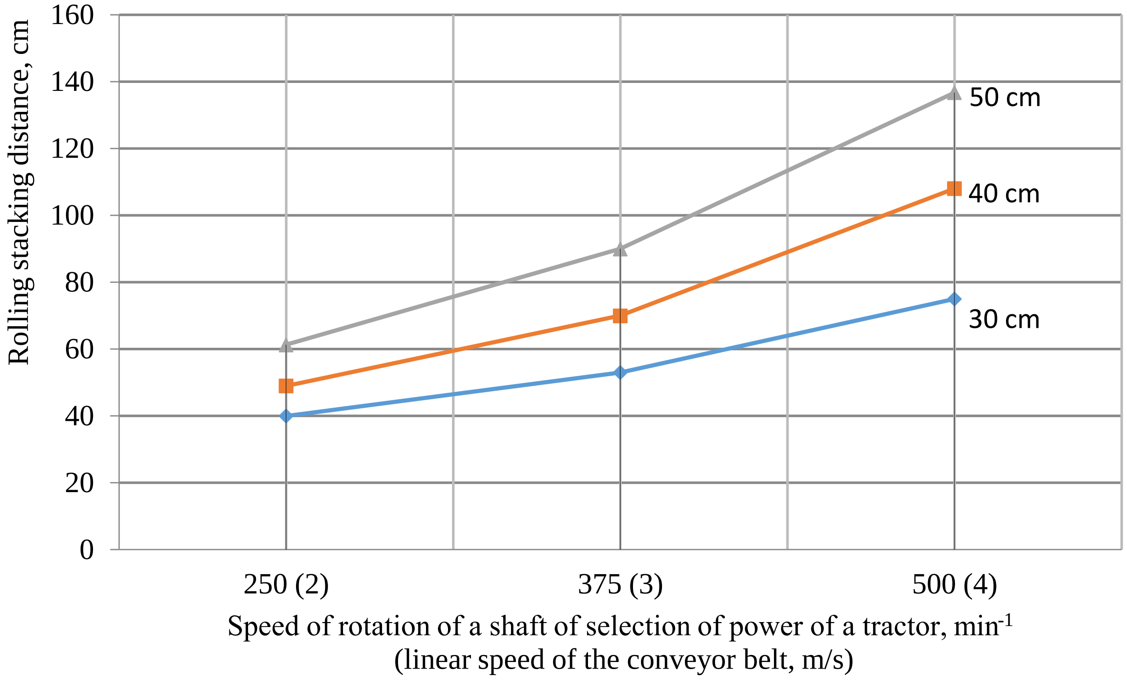

Fig. 16.4 Changing the distance of laying the roll relative to the transverse conveyor depending on the height of its placement above the soil surface and the speed of rotation of the shaft of the power take-off of the tractor: curves 30, 40, 50 cm – the height of the conveyor.

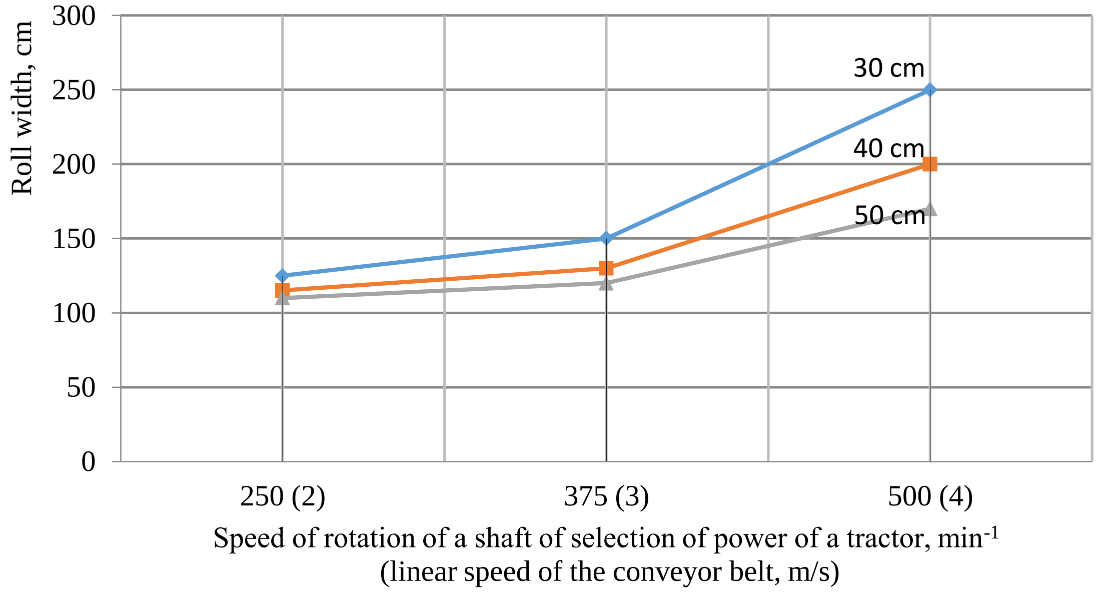

Fig. 16.5 Change of width of a roll of a substrate depending on speed of rotation of a shaft of selection of power of a tractor and height of placement of the cross conveyor over a soil surface: curves 30, 40, 50 cm – change of height of placement of the conveyor above the soil surface.

This dependence can be used to compensate for possible deviations of the trajectory of the unit relative to the centerline of a number of trees during the operation in order to ensure the required accuracy of laying the roll.

The width of the roller also depends on the change in the speed of the tractor power take-off shaft, and hence on the speed of the cross conveyor belt. The latter can vary in the range from 0.8 to 2.5 m when changing the speed of the shaft of selection of power of a tractor from 250 to 500 min^{-1} (figure 5). The width of the roll increases with decreasing height of the conveyor above the ground and increasing the speed of the shaft of the tractor power take-off. Taking into account the current agronomic requirements (the width of the roll should not exceed 1.5 m), the recommended speed of the conveyor belt should be 2.0-3.0 m/s. This corresponds to the speed of rotation of the shaft of selection of power of a tractor with a frequency of 250-375 min^{-1}.

The required rate of application of the substrate is provided by the selection of the optimal operating speed of the unit and the productivity of the feed conveyor and can be 0.03-0.25 m^3/m^2, and changing the same parameters allows depending on the needs to change the height of the roll within 0.05-0.15 m.

Discussion

Production tests were carried out in the garden plantations of the National University of Life and Environmental Sciences of Ukraine with a row spacing of 5.0 m on an area of 0.5 ha. As a substrate used three different physical properties of the substance: fresh sawdust with moisture and bulk density of 30% and 250 kg/m^3, respectively [28]; compacted sawdust with the corresponding indicators of 48% and 470 kg/m^3; used mushroom substrate (65% and 590 kg/m^3, respectively [29]).

The unit operated in two modes with substrate application rates of 0.05 and 0.10 m^3/m^2. They were provided with the speed of the unit, respectively 1.2 and 0.5 m/s and the productivity of the feed conveyor of the base machine is about 2.5 m^3/min [30]. The latter was achieved by installing a ratchet regulator of the specified conveyor in the area of the fifth mark. According to the tests, the deviation of the actual application rate from the set did not exceed 12%, the average width and height of the cover roll were 1.30 and 0.048 m, respectively, in the first and 1.10 and 0.095 m in the second mode. The machine worked stably with all the mentioned types of substrates. There was no damage to the trees.

Productivity of the machines for an hour of the basic time made about 2.0, variable – 0.5 hectares. The coefficient of reliability of the technological process was 0.99, and the use of variable time – 0.40. Operational and technological indicators were determined based on the operating conditions of the machine in the garden with a row spacing of 5.0 m, operating speed of 4.2 km/h at a substrate application rate of 160 m^3/ha. Substrate losses in any mode of operation of the unit did not exceed 0.5%. Studies have shown that compliance with certain operating ranges and the optimal mode of the working bodies of the machine and the unit as a whole ensures the implementation of the technological operation in accordance with current agricultural requirements with high quality indicators. Thus, the deviation of the actual rates of application of the substrate, as well as the width and height of the cover roll from the specified does not exceed 12, 10 and 15%, respectively.

Conclusions to Chapter 16

The capacity of the feed and cross conveyors can vary from 1.2 to 9.0 and from 2.0 to 5.3 m^3/min. To prevent unwanted excessive accumulation of substrate in the receiving hopper of the machine, the capacity of the feed conveyor should not exceed 5 m^3/min.

The distance of laying mulch in the roll relative to the transverse conveyor varies from 0.40 to 1.30 m depending on the height of the latter above the soil surface and the speed of rotation of the shaft of selection of power of a tractor. The dependence is recommended to be used to compensate for possible deviations of the trajectory of the unit relative to the centerline of a number of trees during the operation in order to ensure the required accuracy of laying the roll.

The productivity of the machine was 2.0 per hour of main time, variable time –

variable time – 0.40. Substrate losses in any mode of operation of the machine did not

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CHAPTER 17. RESEARCH OF GARDEN SPRAYER MACHINES OF NEAR-STEM AND INTER-STEM STRIPS OF ORCHARDS

Introduction

Weed control in the area of the root system of trees is one of the main prerequisites for obtaining products in a competitive volume [1] and quality [2]. The negative impact of weeds on cultivated plants is explained by the fact that they compete with them in the fight for moisture [3] and nutrients [4], promote the development of diseases and pests [5], create favorable conditions for the spread of rodents that damage plants in winter [6].

The use of traditional means of mechanical tillage of interstitial strips in intensive gardens is ineffective [7]. First, it is difficult to ensure the mechanical treatment of interstitial strips when the trees in a row at a distance of 2 meters or less [8], and secondly, such treatment is complicated by the possibility of mechanical damage to the root system of trees obtained using vegetatively propagated rootstocks [9]. Therefore, the use of herbicides as the main means of weed control in these cases is the most rational solution to this problem [10], as evidenced by world experience [11]. Analysis of the state of the means of performing this technological operation showed that most of them are hung on the frame of a tractor or sprayer [12] and have a rod structure at the end of which is a spray section [13]. The bar has the device for change of length that allows to carry out technological operation in gardens with various width of interrows [14].

The quality of such devices [15], which are mostly made in the workshops by the efforts of machine operators themselves in most cases does not meet agricultural requirements due to [16], as a rule, exceeding the rate of consumption of the chemical [17] or uneven distribution on the surface of the strip being treated [18]. Therefore, their common disadvantage is the insufficient quality and low productivity of this technological operation [19], and hence low efficiency [20]. In order to determine the influence of design and technical parameters of the spraying section of the device on the quality of its work, an experimental sample of a device for applying herbicides to the stem-interstitial strips of the garden with two spraying sections was attached and mounted on the front of the tractor frame.

Materials and methods

The spray section (figure 17.1) consists of a rear *1* and front *2* protective covers connected by a frame *3*, which in turn is attached to the mechanism for holding the section in the desired position. To increase the width of cultivation and ensure the necessary uniformity of distribution of the chemical, two sprayers *4* (figure 17.1) are installed in the upper part of the section at a height *h1* above the soil surface and the angle of the saw α relative to the horizontal plane. To process the interstitial strip, the section is equipped with an additional sprayer *5* (figure 17.1).

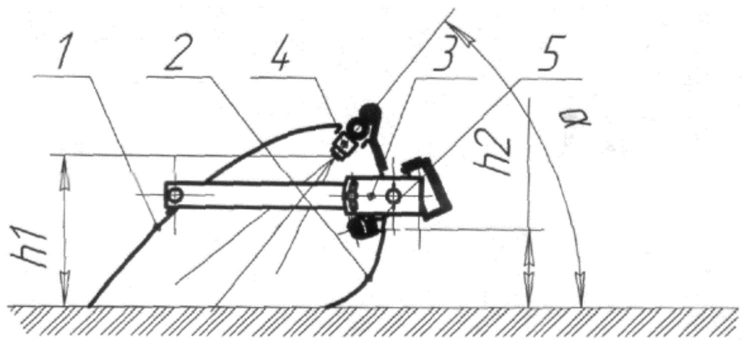

Fig. 17.1 Scheme of the spraying section (side view) and placement of sprayers relative to the horizontal plane (surface of treatment): 1, 2 – protective casings; 3 – connecting frame; 4 – section sprays; 5 – additional spray

The quality of the device was determined by the parameters of density and uneven coverage of the surface to be treated. The study was performed in the field using indicator paper Spray test paper from the Dutch company Hardy. The coating density (pcs/cm^2) was determined by counting 200–300 μm droplets on indicator paper using a microscope.

The dispersion of the spray was determined visually by comparing the droplet size with their sample of known size. The degree of non-uniformity of the coating was evaluated by the coefficient of variation. During the research, the indicator strips were placed on a straight perpendicular axis line of a number of plants at a distance of

0.10 m from each other, and the distance between the extremes was 1.90 m.

As a result, it was found that the density and unevenness, and therefore the quality of treatment of liquid droplets of zone "a" (figure 17.2) depends on the type of sprayers, their performance, location on the working section, direction of spraying and pressure of working fluid.

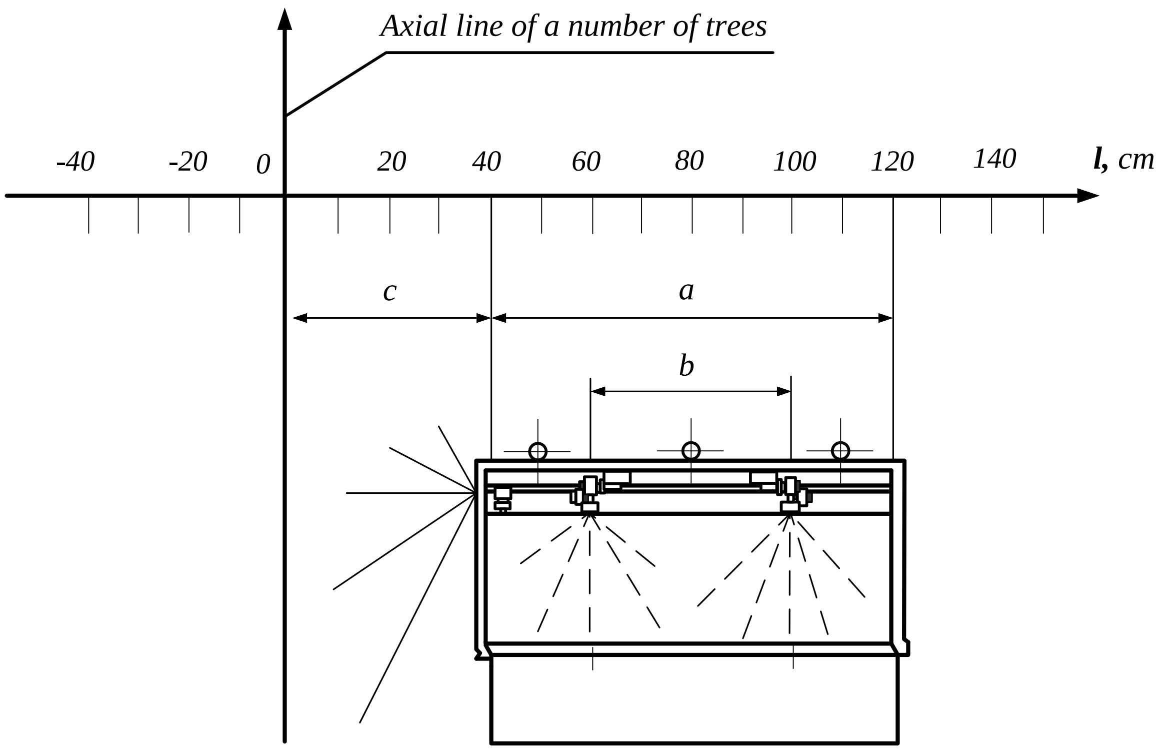

Fig. 17.2 Scheme of the spray section (top view) and its location relative to the centerline of a number of trees.

The required quality of processing, namely the effective density of the coating of the treatment plane with drops of test size (200–300 μm) provide the following components: type of spray – flat jet; cut angle – 110°; productivity at a pressure of working liquid of 0.2 MPas – about 1 l/min; speed of movement of a tractor (working section) – 1.0–1.3 m/s; height of placement of sprays above the plane of processing

(*h1*) – 0,25 m; the angle of the sawing direction relative to the horizontal plane (α) – 40°; the distance between the sprayers (*b*) – 0.35 m; the width of the working section (*a*) – 0.8 m.

Results

As can be seen from the presented material (figure 17.3, curve 1), the density of coating drops of the test size of the processing plane, which is within the zone "a", in compliance with the above parameters, is in the range of 80-120 pcs/cm^2 (minimum agricultural requirements), coating density – 30 pcs/cm^2, which is sufficient to ensure the required quality of the technological operation.

To ensure the treatment of the entire stem-interstitial strip, the spray section must be equipped with an additional spray directed in the direction of the location of the protective and interstore zone "c".

Its optimal parameters are:

- type of spray – flat jet;
- cutting angle – 110°;
- productivity at a pressure of working liquid of 0.2 MPas – about 0.5 l/min;
- height of placement above the plane of processing – 0.13 m;
- angle of the sawing direction relative to the horizontal plane – 25–35°;
- the angle of inclination of the plane of the spray torch relative to the horizontal plane – 10°.

Curve 2 (figure 17.3) characterizes the distribution of the density of the droplet coating of the surface of the interstitial strip by an additional spray.

Accordingly, curves 3 and 4 (figure 17.3) correspond to the distribution of the drop density of the surface of the other half of the stem-interstitial strip, which occurs when the unit moves along the adjacent row spacing in the opposite direction, respectively, additional and main sprays section.

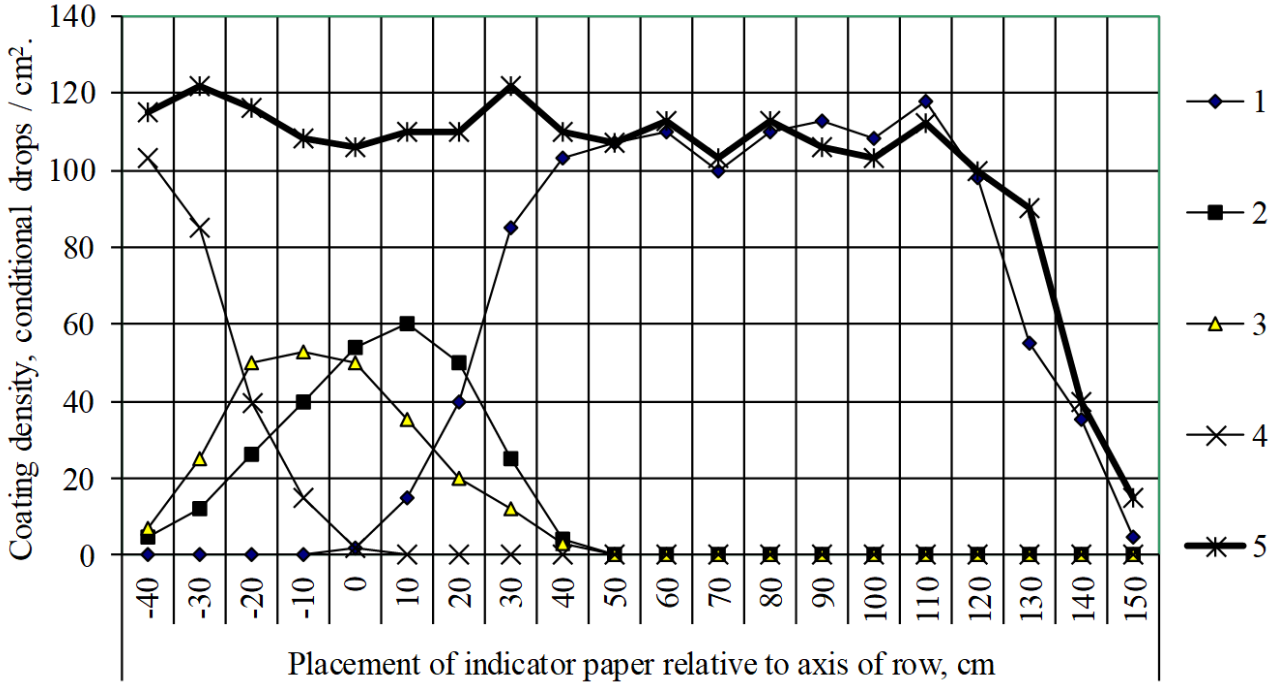

Fig. 17.3 Density of coating of drops of a surface of processing depending on distance concerning an axial line of a number of trees at a full cycle of processing of a trunk and inter trunk strip: 1, 4 – operation of sprayers of section (zone a); 2, 3 – operation of an additional spray (zone c); 5 – operation of all sprayers of section.

Curve 5 (figure 17.3) is the result of the distribution of the density of the coating of the drug stem-interstitial strip (that part of which was observed) when performing the spray section of the full cycle of treatment.

Discussion

According to the results of experimental studies, it is confirmed that the developed device in compliance with the optimal parameters [21] and modes of operation provides the required quality of the technological operation [22]. The density of the coating with test drops of the treatment plane is in the range of 100–120 pcs/cm^2, and the unevenness of the coating (coefficient of variation) does not exceed 6% [23, 24].

The authors confirmed that the dispersion of the spray is also satisfactory [25]. About 90% of the total liquid covered by the treatment surface settled in the form of droplets [26], the size of which is in the range of 100–300 μm, which is optimal for this process [27]. To ensure the required efficiency of use of the device in gardens of both intensive and traditional types [28], the section sprayers are connected to the pressure line individually and can be, if necessary, disconnected from it [29].

Therefore, the section can operate in three modes depending on the required processing width of the stem-interstitial strip [30]. The width of the treated strip during the operation of the section with one sprayer is 0.5–1.0, with two 1.0–1.5, with three – 1.5–2.0 m. Production tests were carried out in the experimental plantations of the orchards of National University of Life and Environmental Sciences of Ukraine. The total area of the cultivated garden was about 10 hectares. Age of trees is 5–12 years, width between rows is 4–5 m. Keeping the soil between rows is a natural turfing.

The use of the developed device in comparison with the existing means of chemical treatment of the stem and inter-stem bands allows to reduce the cost of the active chemical substance by 30–40% [31]. Application of Roundup with a dose of 5 l/ha of the treated area gave the treatment efficiency of 96%.

Conclusions to Chapter 17

Ensuring the treatment of the entire stem and inter-stem strip of orchards is carried out by a chemical treatment machine with spray sections, which are equipped with additional sprayers directed in the direction of the location of the protective and stem stem. The optimal parameters of such additional sprays are: type of spray – flat jet; cutting angle – 110°; productivity at a pressure of working liquid of 0.2 MPas – about 0.5 l/min; height of placement above the plane of processing – 0.13 m; angle of the sawing direction relative to the horizontal plane – 25–35°; the angle of inclination of the plane of the spray torch relative to the horizontal plane – 10°.

The density of the coating with test drops of the plane of treatment by a chemical treatment machine with spray sections are in the range of 100-120 pcs/cm^2, and the non-uniformity of the coating does not exceed 6%. The dispersion of the spray is satisfactory. About 90% of all protective chemical liquid, which covers the surface of the treatment, settled in the form of droplets, the size of which is in the range of 100-300 μm.

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CHAPTER 18. MODELLING THE THROUGHPUT CAPACITY OF THRESHING-SEPARATING APPARATUS OF GRAIN HARVESTER'S COMBINES

Introduction

The generalizing technical [1] and operational indicator of the threshing device is considered as throughput [2]. Implementation of harvesting technology is a condition of stability and quality of threshing bread mass [3]. Harvesting is characterized by certain quality indicators: technological standards (capacity) [4]; and permissible deviations from it (technological admission to standards) [5]; the accuracy of the developed requirements [6] or the level of coincidence of the quality indicators obtained in real production with the permissible ones [7].

Qualitative indicators of work of combine harvesters Slavutich KZC-9F and Slavutich KZC-9M at correct regulation of working bodies [8] and conditions are defined by ISO 22611. Indicators and characteristics for tolerances are given in table 18.1. Uniformity of supply of bread mass in the threshing machine depends on presence [9] and influence of a considerable number of factors and factors: unevenness of density of plants [10], height and humidity of bread [11], use of width of capture of a reaper [12], unevenness of height of mowing [13], unevenness of giving by an auger of a reaper [14] and floating inclined transport degree of weeding [15].

In most literature sources [16-18] the capacity of the threshing machine is given as a constant value and depends on design and operational factors $g_n = 0.485 \cdot (N_e \cdot 32^{-1} + F_n \cdot 0.26^{-1} + F_c \cdot 1.5^{-1} + F_o \cdot 0.8^{-1}) - 0.83$. In real production, as practice shows, bandwidth is a variable that varies from objective subjective factors [19]. Objective factors are: soil and climatic conditions [20], relief and contours of fields [21], physical and mechanical characteristics of the harvested crop [22], design and operational indicators [23] and characteristics of combines [24]. Subjective factors – weeds [25], straw [26], agricultural crops [27], moisture content of the bread mass

[28], the qualification of the combine [29], the choice of working speed [8] and cutting height [12], the width of the header [6].

Table 18.1

Qualitative indicators of work of combine harvesters.

Qualitative indicators of work	Slavutich KZC-9F	Slavutich KZC-9M
Grain productivity is the term of the main time (t/hour)	14.0	11.3
Loss of grain no more: for the reaper at the lodging of bread up to 20% (%)	0.5	
Loss of grain is not more when the bread falls on the picker (%)	1.5	
Loss of grain no more (unthreshing, cleaning) for the thresher (%)	1.5	
Grain crushing no more – ear (%)	2.0	
Grain crushing no more – corn (%)	3.0	
Grain crushing no more – sunflower (%)	3.0	
The content of contamination in the grain mass of the hopper (%)	2.0	

Materials and methods

The calculation of the actual capacity of the threshing machine of combines must take into account the agrobiological state of the bread mass during combining: it is the straw mass of the bread mass, grain moisture, straw moisture and weeds. Normative indicators are: straw $\delta_s = 1.5$, grain moisture $V_{g.m.} = 15\%$, straw moisture $V_{s.m.} =$ $= 17\%$ and weed $V_{w.} = 5 - 50\%$. Taking into account the above factors, the actual bandwidth can be determined by the following dependence $q_f = a \cdot b \cdot \delta \cdot q_m$, where q_f – the actual capacity, kg/s; a – coefficient that takes into account the change in capacity of the ratio of grain and straw $V_{g.m.} = 15\%$; b – coefficient that takes into account the change in capacity depending on the moisture content of the bread mass

$V_{s.m.} = 17\%$; δ – a coefficient that takes into account the change in capacity from the contamination of the bread mass $V_w = 5\%$.

To take into account the influence of straw, the author [8] proposes to use the value of the coefficient a, determining from the expression: $a = (1 - C_o) \cdot (1 - C)^{-1} = (1 - 0.66) \cdot (1 - 0.64)^{-1} = 0.94$, $(1 - 0.94) \cdot 100\% = 6\%$, where C_o, C – estimated and actual grain content in straw, in fractions of a unit. After the transformations of this formula, machine test stations are used when testing combines $a = \{0.6 \cdot (1 + \delta_s^{-1})\}^{-1} = \{0.6 \cdot (1 + 1.5^{-1})\}^{-1} = 1$. The calculated values of the coefficient a depending on the straw are given in table 2. As can be seen from table 18.2, the change in straw in the bread weight from $\delta_s = 0.90$ to $\delta_s = 1.90$ changes the value of the coefficient of influence a from 0.79 to 1.09, i.e. by 30%.

Table 18.2

The calculated values of the coefficient a depending on the straw.

δ_s	0.90	1.00	1.10	1.20	1.30	1.40	1.50	1.60	1.70	1.80	1.90
a	0.79	0.83	0.87	0.91	0.94	0.97	1.00	1.02	1.05	1.07	1.09

The influence of the moisture content of the bread mass on the value of the coefficient b for unaltered grain stands of cereals under regulatory conditions can be determined from the dependence:

$$b = \left\{(100 - V_g) \cdot (100 - V_{gi})^{-1} + \delta_s \cdot (100 - V_s) \cdot (100 - V_{si})^{-1}\right\} \cdot \{1 + \delta_s\}^{-1},$$

where V_g and V_s – normative conditioned humidity of grain and straw; V_{gi} and V_{si} are the actual moisture content of grain and straw, %. For calculations we will accept change of humidity of grain V_g from 20% to 13%, straw V_s from 22% to 15%. The calculated data of the change of the coefficient b from the change of moisture content of grain and straw are given in table 18.3.

As shown in table 3, the decrease in grain moisture V_g from the value of 20% to 13% and straw V_g from 22% to 15% affects the value of the coefficient in the range from 1.063 to 0.976, i.e. 0.087=8.7%.

Table 18.3

The calculated data of the change of the coefficient b from the change of moisture content of grain and straw.

Indicators	Humidity of grain and straw, (%)							
Grain V_g	20	19	18	17	16	15	14	13
Straw V_s	22	21	20	19	18	17	16	15
Coefficient b	1.063	1.048	1.037	1.024	1.012	1	0.988	0.976

In the case of weediness of the bread mass, the value of the coefficient of influence b can be determined from the dependence: $\delta = \left\{C_o \cdot \left(100 - V_{gi}\right) \cdot \left(100 - V_{gi}\right)^{-1} + (1 - C_o - \varepsilon) \cdot (100 - -V_s) \cdot (100 - V_{si})^{-1}\right\} \cdot \{1 + \varepsilon\}$, where ε – the content of weeds in the bread mass in fractions of a unit.

The calculated values of the coefficient of influence b of weediness of the bread mass on the capacity of the threshing machine of the combine harvester are shown in table 18.4. Estimated values of the impact factor on the throughput the ability of the thresher under normal conditions $V_g = 15\%$, $V_s = 17\%$.

Table 18.4

The calculated values of the coefficient of influence b of weediness of the bread mass on the capacity of the threshing machine of the combine harvester.

Indicator	The relative weediness of the bread mass, (%)								
V_w	5	10	15	20	25	30	35	40	45	50
b	0.976	0.957	0.943	0.924	0.912	0.982	0.875	0.868	0.870	0.840

The threshing capacity is most significantly affected by weediness due to humidity, with a relative weediness of 0.05 bread weight, the capacity of the threshing machine is reduced by 2.5%, and with 50% weed – by 16%. The most significant factor influencing the capacity is the design of modern combine harvesters, which are equipped with combine harvesters. These are electronic means of controlling crop losses. All modern combine harvesters are equipped with electronic systems of control of losses of a crop behind a thresher. Modern electronic control systems consist of three elements: piezoelectric sensors, which are mounted on straw walkers (straw walkers) and sieve condition. Sensors that receive pulses from falling grains are connected to pulse shaping amplifier. From the sensors, signals are sent to the cab of the combine to the block of loss information – BLI (block of loss information), the indicators of which are guided by the combine when choosing the operating speed in the operational plan for loading the thresher. As shown in table 18.1, the permissible threshing losses should not exceed 1.5% of the organic crop grown in a particular field. The most significant disadvantage of modern electronic crop loss control systems is that they show visually relative losses, rather than quantitative ones, which we must take for granted. If you are guided in the management of the combine in the field, the relative signals of the BLI includes the subjectivity of the operator, which depends on its physiological characteristics, knowledge, skills, depending on experience and qualifications, because it is necessary to take into account uneven weight, yield, folds, fields within 50% of the average value.

Results and discussion

Subjective perception of the BLI sensor readings in the range from "from – to" leads to a decrease in the efficiency of bandwidth utilization and, accordingly, to a reduced use of engine power. To verify the relative readings of the loss sensor, a practical experiment of numerical values "from – to" was performed. The speed selection in the Slavutich KZC-9F and Slavutich KZC-9M are controlled by a lower-to-to sensor, which is shown in the 1, 2, 3, 4 format printed from the on-board computer under version №1, №2, №3, №4. Analysis of forms of versions №1, №3 according to the lower value "from" the time of passage of the combine in the bend was 10.2 minutes or 612 seconds, the path of the bend was 1080 meters, working speed 6.282 km/hour. Taking into account the collected area of 0.730 ha, the width of the harvester on the average value was 689 meters or 91.8% of the geometric width of the harvester 7.5 meters. From the variant of version №4 it is seen that the working speed was higher 9.833 km/hour (passage time was 432 seconds, the length of the path in the bend 1.180 km, the collected area – 0.733 ha.). The average width of the harvester in the fold was 6.21 meters, or 83% of the geometric width. The capacity of the thresher in version №3, the average value at straw size $\delta_s = 1.5$ was $q_1 = 8.9$ kg/s, in version №4 $q_1 = 14.4$ kg/s. The calculated value of throughput according to formula was 19.63 kg/s, that is, according to the first and fourth options; the bandwidth was used within 43.3% and 74.3%. Calculations on the indicators of average values are given below.

Increasing productivity by the average values of harvested hectares per hour: option №1 S_1=4.43 ha/hour; №4 S_4=6.28 ha/hour. $\Delta S = S_4/S_1 = 6.28/4.43 = 1.417$, $+41.7\%$.

Increase of productivity on average values of the collected tons for an hour: option №1 q_{h1}=21.38 t/h; №4 q_{h4}=34.05 t/h.

$\Delta q = q_{h4}/q_{h1} = 34.05/21.38 = 1.592$, +59.2%.

Increase in productivity on average values of the collected tons from hectare, t/h. №1 q_{n1}=4.83 t/ha; №4 q_{n4}=4.83 t/ha. $\Delta q_n = q_{n4}/q_{n1} = 5.42/4.83 = 1.122$, $+12.2\%$.

Relative values for fuel consumption per hour, l/h: №1 G_1= 54.0 l/year; №4

G_4= 77.143 l/year. $\Delta G = = G_4/G_1 = 77.143/54.0 = 1.428$, +42.8%.

Relative values for fuel consumption per harvested 1 ha area; №1 Q_1=12.529 l/t; №4 Q_4=12.280 l/t. $\Delta Q = Q_4/Q_1 = 12.280/12.529 = 0.980$, -2%. Relative values for fuel consumption per 1 ton collected; №1 Q_{t1}=2.525 l/t; №4 Q_{t4}=2.265 l/t. $\Delta Q_t = Q_{t4}/Q_{t1} = 2.265/2.525 = 0.897$, -10.3%.

The coefficient of the actual width of the header in the 1st option; 0.730 ha; path 1.058 km. 0.918, -0.61 meter. The coefficient of actual capture of width of a reaper in the 4th option; 0.730 ha; path 1.180 km. 0.813, -1.29 meter. Summary data on improving the efficiency of use Slavutich combines on the relative indicators of grain loss for the thresher with shifted icons to the indicator "to" are shown in table 18.5.

Table 18.5

Summary data on improving the efficiency of use Slavutich combines

The name of the meter readings	07.08.2020. Data of research			
	«from»	average values	average values	«to»
Work time, (seconds)	612	540	468	432
Shredder time, (hours)	0.17	0.15	0.13	0.12
Area, (ha)	0.739	0.750	0.745	0.733
Shredder area	0.739	0.750	0.745	0.733
Road section, (km)	1.058	1.082	1.013	1.180
Mass of harvest, (tons)	3.56	4.70	4.65	4.00
Humidity, (%)	13.9	14.5	14.2	13.3
Fuel consumption, (liters)	9.01	10.01	10.01	9.01
Fuel consumption on the road, average values, (ha/hours)	4.43	5.00	5.59	6.28
(tons/hours)	21.38	31.35	34.85	34.05
(tons/ha)	4.83	6.27	6.24	5.42
(liters/hour)	54.00	60.00	67.50	77.14
(liters/ha)	12.190	12.000	12.079	12.280
(liters/tons)	2.529	1.914	1.937	2.265
Working speed, (km/hours)	6.282	7.2113	7.792	9.833

Loading of the threshing machine determines the productivity of the combine and fuel efficiency due to fuel consumption for harvesting 1 hectare, as well as threshing 1 ton of grain.

The power of the combine engine is designed to perform work in extreme conditions, as a result, the average load is 2/3 of full.

During the operation of the combines, the time is only 45–50% of the total operating time of the engine. Underemployment of the engine leads to overconsumption of fuel.

So, if the SMD-21M diesel at full operational power of 103 kW consumes fuel of 0.24 kg on 1 kW for an hour, at loading on 50% – 51 (kW seconds), specific fuel consumption increases by 41 (g/kW hours), or 17%, which corresponds to fuel consumption 2.1 kg/year.

Forcing the engine due to the intensification of inflation parameters worsens the situation. Thus, the SMD-23M diesel, boosted to operating capacity 118 kW at a load of 51 kW will consume 25% per unit of power instead of 17%.

Conclusions to Chapter 18

The calculation of the actual capacity of the threshing machine – combine harvester and, accordingly, productivity must objectively take into account the agrobiological state of the grain mass, grain moisture, straw moisture, weediness of the bread mass due to correction factors.

It is also necessary to use the design factors inherent in the electronic device for monitoring crop losses for the accuracy of the use of the relative BLI in determining the mechanical losses of the thresher by the definition of "from – to". By increasing the allowable losses to the upper value "to" the capacity of the thresher can be increased up to 40% while saving fuel due to more efficient use of engine power. That is, a qualified manager of the relative readings of the loss sensor can increase the productivity of the combine to 40% and reduce fuel consumption by 10% due to the optimal engine load.

The specified method of calculation of throughput of threshing – the device taking into account an agrobiological condition of bread weight and taking into account use of losses of grain on indicators of BLI is resulted.

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