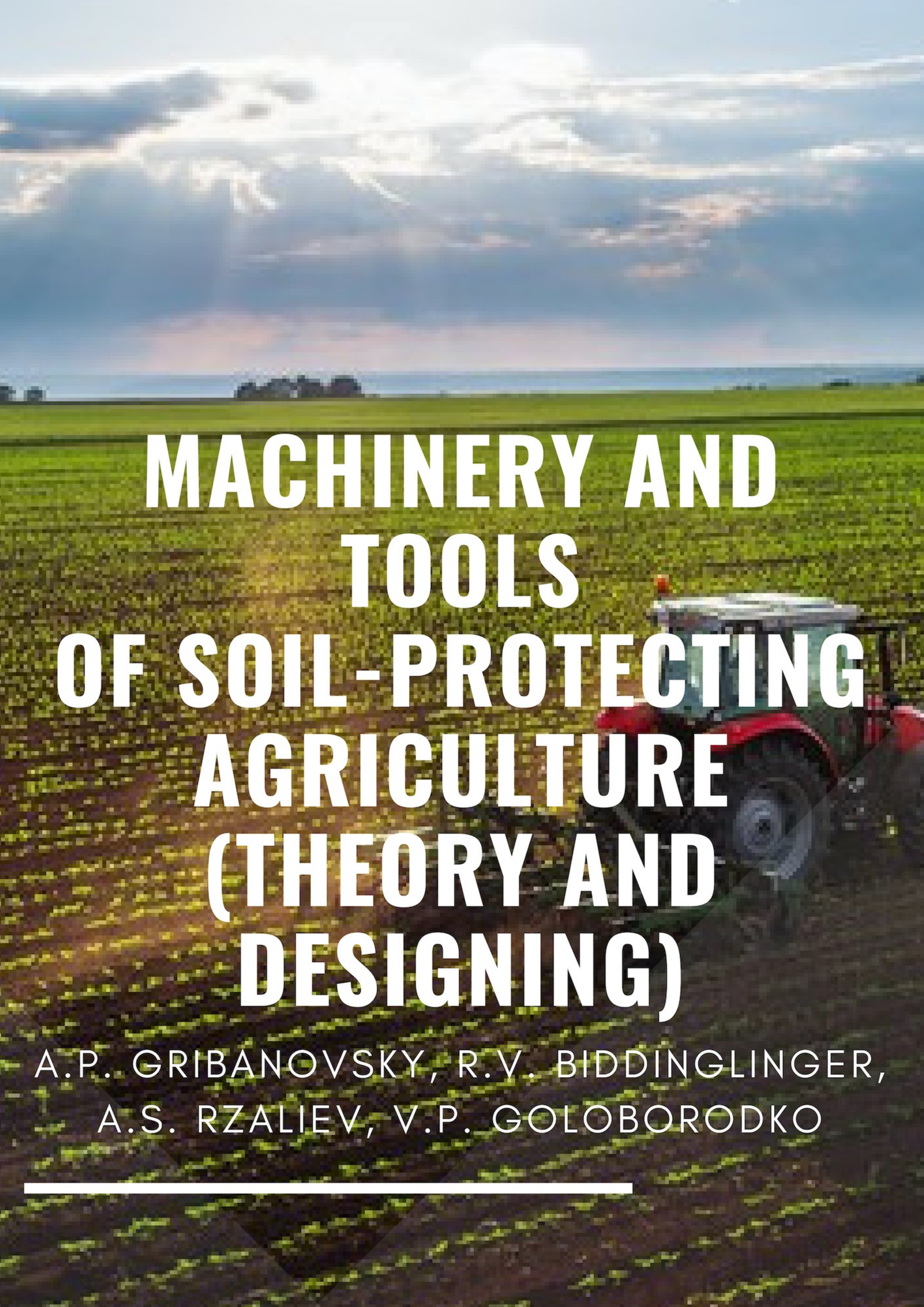

MACHINERY AND TOOLS OF SOIL-PROTECTING AGRICULTURE (THEORY AND DESIGNING)

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Machinery and Tools of Soil-Protecting Agriculture: Theory and Designing

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ISBN: 9781792636585

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Synopsis

Agricultural production is associated with land degradation and depletion, deforestation, environmental pollution, deterioration of water and air quality, etc.

According to soil scientists of the Moscow State University during the entire period of agricultural activity (about 10 thousand years), the world lost 2 billion hectares of arable land, which is much higher than the presently arable land area of - about 1,5 billion hectares. And the intensity of losses increases: 700 million hectares are lost over the past 300 years, with an average annual rate of 2,3 million hectares, and 300 million hectares are lost over the past 50 years with an average annual rate of 6 million hectares. According to modern estimates, approximately 1.2 billion hectares, out of 1,5 billion hectares in the turnover of farmland, are now in a state of degradation.

Such consequences of human agricultural activity are the result of a mismatch between his interests and the "strategy of nature". With the aim of concurrence of human interests with the "strategy of nature" and preserving the invaluable gift of the nature of soil fertility, mankind develops soil protection technologies for cultivating agricultural crops and technical means for the implementation of these technologies in relation to specific soil and climatic conditions. At the same time, the following basic principles are laid down in the basis of the performance of any technological operation in the cultivation of crops in soil conservation agriculture: minimal spraying of the treated soil layer; reduction of airflow velocity in the surface layer; maximum accumulation and rational use of moisture in the soil.

The principles of the soil protection system of agriculture and the specific working conditions of agricultural machinery exclude the possibility of using machinery designed for the classical (dump) farming system. This led to the emergence of a new type of machinery and tools to protect the soil from wind erosion and to combat drought.

The need for intensification of agriculture in hard arid regions, the formation of a system of anti-erosion measures taking into account zonal features, changed energy resources and forms of labor organization predetermine the expediency of developing new or improving existing machinery and tools for soil-protecting agriculture. The construction features of such machinery and tools, their adjustment and operating features are widely covered in the literature. However, to this day there is no generalized material in which the results of the main theoretical and experimental studies that form the basis for the creation of machinery and tools for soil conservation agriculture would be concentrated. Studies are scattered and published in scientific works of many different countries. This complicates work of researchers and designers aimed to improve and create new machinery for soil protection agriculture, hampers training of specialists in higher and secondary educational institutions.

In this book, an attempt is made to systematize the main results of numerous studies that are the basis for creation of machinery and tools for soil-protecting agriculture. In it the following circle of questions is considered: operating conditions of anti-erosion machinery and agrotechnological (initial) requirements to them; the main theoretical and experimental provisions for the selection of rational values for the parameters of operative parts and structural and technological circuits for erosion control machinery and tools for soil cultivation, accumulation and rational use of moisture in soil, intrusion of mineral fertilizers and herbicides,seeding of crops, accumulation of moisture in the soil at the expense of winter precipitation.

The research results presented in the book are used as the basis for the development in the former USSR of a complex of anti-erosion machinery and tools, which were produced in large volumes by the industry. This made possible to apply soil-protective agriculture in the USSR on an area of more than 60 million hectares, incl. in the Republic of Kazakhstan - on 22 million hectares.

Currently, the results of these studies are used in the Republic of Kazakhstan in developing of new and improving of existing machinery and tools for soil-protecting agriculture, which predetermines the necessity and expediency of writing of this book.The List of literature at the end of the book does not pretend to give the complete coverage of relevant publications profile and contains only works directly used in the creation of new and improving existing machinery and tools for soil conservation agriculture.

1.About Soil-Protecting Agriculture

Soil protection system of agriculture is one of the varieties of soil-protective, intensive and resource-saving agriculture. World experience in their development has shown that the provisions and principles of soil conservation do not depend on economic and social conditions but are determined by the natural and climatic conditions and the biological possibilities of agricultural crops cultivated under such conditions. At the same time, the basic requirements to the technologies of crop cultivation were clearly formed:
- soil protection from wind and water erosion;
- preservation and improvement of soil fertility;
- obtaining of the maximum yield with minimum energy, labor and financial costs;
- obtaining of high-quality products;
- environmental protection in order to ensure safety of human life.

Therefore, all technological operations of cultivation of crops with soil-protecting agriculture should solve the following main tasks.

1) Optimization of soil density and its structural state

On soils, the equilibrium density of which is close to optimal for cultivation of certain crops, the task of its loosening is shortened. In such a case, zero processing is possible if other tasks of tillage are replaced by other means (for example, the use of herbicides). The criterion for assessing the density and structural state of soils is the presence of water-resistant aggregates of more than 0.25 mm. For different soil types this scale has not yet been developed. But roughly, in loamy soils such aggregates should be more than 40%.

2) Regulation of water balance of soils

It is necessary to ensure the transfer of atmospheric precipitation into the soil thickness, to reduce their surface runoff, and to reduce the physical evaporation of moisture from the soil surface.

3) Prevention of water and wind erosion of soil

It is achieved by regulating the surface runoff of water and by reducing the wind speed in the surface layer of the soil by creating on its surface of various roughness's (small crests, stubble residues, etc.).

4) Regulation of the regime of organic and biogenic elements, placement of fertilizers and meliorants in the plow layer

It is established that the intensity of mineralization of organic matters depends on the nature and frequency of mechanical tillage. This process is most active during plowing, less intensively - with no-till processing, i.e. the biological activity of the soil decreases, since the entire microflora is in a small layer of soil.

It is clear that fertilizers and meliorants must be introduced into the root layer of the soil, and not scattered, in the truest sense of the word, on the surface of the field. Then, from the surface of the field, they are washed into water bodies, rivers, lakes. The consequences of it are well known.

5) Regulation of phytosanitary conditions

In the technology of cultivating crops before the appearance of pesticides, the mechanical treatment of the soil also carried out the task of combating weeds, diseases and pests. Today, with the increase in environmental requirements for crop production technologies, the role of mechanical tillage to maintain the proper and phytosanitary condition is increasing.

6) Creation of optimal conditions for seeding and obtaining of amicable shoots

When pre-seeding cultivation of soil, it is necessary to ensure conditions under which the seeds would lie on a densified wet bed, and from above would be covered with a loosened layer of soil. While seeding, the required number of seeds should be distributed evenly and strictly at a given depth.

7) Energy saving and economy

Along with the soil protection orientation and the desire for the biologization of agriculture, soil cultivation must ensure a minimum of inputs of energy, labor and financial resources.

The above tasks, correlated with different natural conditions and with the biology of growth and development of cultures, serve as a guide for selecting possible options for mechanical tillage and performing a particular technological operation in the cultivation of crops, and a set of certain mechanical treatments forms a system for cultivating the soil for a particular crop in the rotation.

Based on the analysis of domestic and world practice in use of mechanical types of soil treatment in the world today, the following classification of soil treatment systems is applied.

a) Dumping system of agriculture - full or partial wrapping of soil layers. Two types of it are used: different in depth and minimal.

The different in depth: deep - more than 24 sm, ordinary - 18-24 cm, small - 8-16 cm and superficial - up to 8 sm.

The minimal: surface - up to 8 sm and shallow - 8-16 sm.

Currently, the most widespread is the different in depthwith a tendency to minimization.

b) Mulching system - processing of soil without a turn of a layer with preservation of crop residues on the surface of the field.

Three types of it are used: deep, shallow and minimal.

The deep - systematic tillage of the soil to a depth of more than 24 cm without recirculation (on solonetzes, solonetsous soils and heavily compacted soils, on erosive soils).

The shallow depth - the alternation of shallow and deep tillage, depending on the crop cultivated in the crop rotation and soil conditions, for example, soil protection technology for the cultivation of cereals, developed under the guidance of Academician A.I.Baraev [44].

The mulching - shallow tillage of the soil without turnover of the formation (on light soils in texture).

c) Combined system - various combinations of dump soil treatments with no-tillage to various depths (usually in crop rotations with corn and in forest-steppe conditions). Can be used in combination of deep treatments, shallow or minimal.

d) Zero system - the soil is not subject to any kind of machining, except for the impact on it of the working organs of the sowing units. This system is called the system of the future. The condition, problems and prospects of this treatment system will be described below [4].

e) Ridge system - for cultivating only a certain type of crops on sliced ridges or ridges. It is commonly used in cold and humid climate.

f) Band-cultivation of soil with sowing of seeds into it. This system includes positive elements of minimal and zero soil protection technologies and is used mainly for cultivating tilled crops.

Intensification of agricultural production, regardless of the soil treatment systems, is usually accompanied by an increase in energy costs and productivity of agricultural units. An increase in the depth of soil cultivation, an increase in the number of operations in the soil treatment system intensify such negative phenomena as spraying of soil aggregates, excessive loosening of the treated soil layer and compaction of its underlying layers, loss of moisture, water and wind erosion [1].

World experience of agriculture shows that intensification of cultivation of crops leads to an increase in yields. But at the same time, energy costs per unit area of their cultivation are sharply increased, for example, at 2 - 3 times the increase in yield, energy consumption grows 5 - 20 times [2,3]. That is why the last 40 - 50 years in all developed agrarian countries in the world is searching for the most effective soil treatment system. The impetus for the search for a new soil treatment system was the following two factors.

The first factor is not the indisputable statement of scientists in the first half of the last century that the most important requirement,for any soil preparation system is effective control of weeds, and if the soil is practically free from weeds, then the requirements of the crops for processing quality are not very high. But the regulation of the phytosanitary state of the fields is only one of the six tasks mentioned above. In what way will the other five functional tasks be solved? This was the whole point and purpose of carrying out complex studies in different soil and climatic conditions of the countries of the world. An analysis of the results of these studies shows the following.

The abandonment of a large number of postharvest remnants on the surface of the field with zero tillage system, the absence of mixing of its layers and the spread of fertilizers on the field surface lead to large differences in the physical, biological and agrochemical properties of the soil along the profile (horizontally).

Ten-year research (Canada) on the effectiveness of zero processing on dark chestnut heavy loam soil showed that in the grain-steaming crop rotation the greatest changes were observed in the 0 - 6 sm layer - water permeability and useful water retention capacity decreased, the bulk density increased. But these indices in the 6 - 12 sm layer did not reach the limits that prevent the normal development of the root system of plants. However, most soils do not possess such favorable physical properties, and they can deteriorate, for example, compaction is possible, and the number of air-filled pores is reduced. It is for this reason, as established by research in Australia, that the length of the roots of wheat per unit volume of soil in zero processing compared to the dump was less by an average of 21%.

With zero processing, the accumulation of organic matter occurs predominantly in the surface layer of the soil.After 13 years of zero processing (Australia), the organic matter content in the 0 - 10 sm layer was 3.13%, with the dumping treatment – 2,87%. Although it is generally believed that when planted in the soil, the mineralization of plant residues, due to good aeration and high microbiological activity, proceeds faster than with zero treatment.

For this reason, there is a differentiation in the profile of the biological properties of the soil, which has a strong effect on the nutritional regime of the soil. The presence of an abundance of carbon sources and energy in the form of post-harvest residues in the surface layer leads to activation of microbiological activity, to a sharp increase in microbiomass in comparison with other methods of soil cultivation. Studies (USA, New Zealand) found that in the 0 - 5 sm layer with zero processing, the biomass for carbon was 1,6 times greater, and for nitrogen, 1,5 times. But already at a depth of 10 sm, the nitrogen content in the microbiomass becomes equal or even less at zero processing than in the dump. This trend was observed both in medium loamy and heavy loam soils.

In general, it has been experimentally established that the content of humus, total nitrogen, mobile forms of nitrogen, potassium and some trace elements is much higher in the surface layer of the soil, higher microbiological activity. However, due to stronger soil compaction, weakening of aeration, and often lower temperatures, the availability of nutrients is lower. Therefore, it is believed that with zero processing, the demand for mineral fertilizers may be higher.

With zero processing, the use of organic fertilizers causes great difficulties, since they are supposed to be plowed into the soil, and not left on the surface of the field.

The second factor is the introduction of herbicides into production, which make it possible to ensure the purity of fields from weeds.

In modern agriculture, with its intensification, the means of chemicalization play an important role in increasing the production of agricultural output. However, they have a negative impact on the environment. For example, world experience has shown that with an increase in the yield of grain by 3,5 times, the introduction of mineral fertilizers is increased 25 times, and chemical protection equipment 14 times. When the yield of grain is increased by 6,5 times, the application of mineral fertilizers is increased 125 times, and of protective equipment 100 times [2]. How can the environment sustain this load at all? As a result, gradually, but inevitably and continuously, the natural composition of the atmosphere, water, soil, etc. changes.

With an increase in the volume of application of chemical plant protection products, they can be washed into groundwater. Moreover,this process depends not only on the properties of chemical preparations, but also on the physico-mechanical properties of soils. Thus, 33% of the US area, where intensive agricultural production is conducted, coincide with zones where soil, climatic and hydrological conditions promote the washing out of chemical plant protection products in groundwater. According to the results of research in the USA, 10% of public and 4% of rural private wells contained chemical plant protection products, and in 13% of them the content exceeded the permissible levels. The close location of groundwater to the surface, irrigation, torrential rains, mistakes with the use of chemical protection agents accelerate their penetration into ground and drinking water. There have been no such studies in the Republic of Kazakhstan. Therefore, from the point of view of ecology, one should be cautious about the use of chemical means of plant protection. Where possible, you need to do without them or severely restrict their use, especially as according to research conducted in Canada [2, 3], the total costs of caring for "chemical vapor" depending on the degree of contamination are quite high and amount to 52 - 100 US dollars per 1 hectare, and for an ordinary ferry - 37 - 60 dollars per 1 ha. According to the Kostanai Agricultural Research Institute, the application of mechanical or chemical treatments on steam fields is practically the same, amounting to approximately $ 42 per hectare.

It was suggested that these factors make it possible to minimize mechanical tillage of the soil, which will allow [1]:

- to receive an economic effect (economy of a labor, combustive-lubricating materials, material resources and money resources);

- minimize loss of moisture under conditions when its content in the surface layer of the soil is critical;
- to preserve organic matter in the upper 5-sm layer of soil;
- reduce the risk of water and wind erosion;
- maintain the main advantages of the undisturbed structure of the treated soil layer.

Thus, the results of long-term foreign studies in general have confirmed the expected benefits of zero tillage. But at the same time, the following main shortcomings were revealed [4]:

- higher costs for chemical plant protection products from weeds, pests and diseases;
- additional costs for the maintenance of special equipment while maintaining the traditional, as usually
not all soils in one farm are suitable for zero processing;
- not all crops produce high yields with zero processing;
- higher requirements for the qualification of farmers, especially with regard to the use of chemical plant
protection products, mineral fertilizers and soil ameliorants;
- difficulties with the use of organic fertilizers, since their effectiveness is not low in soil.

For these reasons, despite very optimistic forecasts for the introduction of zero processing, it is still used on a small area, and in the past 15 years, the growth in areas has slowed or stabilized (in the US, the share of use is just over 6% of traditional tillage systems, and in Canada by 0,2% of arable land) [4]. This indicates that in the production environment, zero processing has already found its niche. But intensive research is continuing on its improvement and possibilities for expanding its use. An extensive experimental material has been accumulated on the feasibility of combining zero tillage with other systems, especially where soil properties deteriorate with prolonged zero processing. This makes it possible to realize the advantages of zero processing on soils that are not resistant to compaction.

In connection with the above research on the development of technologies and technical means of soil conservation in Kazakhstan, which has huge arable land in various soil and climatic conditions, should be continued, which predetermines the usefulness and usefulness of the material in this book.

2. Machinery and tools of soil-protecting agriculture for soil treatment

2.1. Working conditions and agrotechnological (initial) requirements for machinery and tools

As noted in the previous section, for solving technological problems of soil conservation agriculture, optimization of soil density and its structural state; regulation of water balance of soil; prevention of wind and water erosion; regulating the regime of organic matter and nutrient elements and placing fertilizers in the plow layer; regulation of phytosanitary conditions; creating optimal conditions for sowing; maintenance of energy saving and profitability are applied dump, mulch, combined, zero, comb-bedding and strip systems of processing of soils or their certain combinations.

In soil conservation agriculture for the cultivation of some tilled crops is used dump soil treatment system. The complex of machines of this system is well known, and the conditions of their work and agrotechnological requirements (initial) to them are taken into account when creating them. Numerous and long-term research results in the development of these machines and tools and their improvement are widely covered in numerous literature and therefore are not included in this book.

Technology of cultivation of crops on soils subject to wind erosion, provides for the following types of its processing without turnover of the formation with the maximum amount of crop residues remaining on the surface of the field with minimal spraying of the treated layer:

- processing to a depth of 8 - 16 sm (autumn plowing, first and intermediate steam treatments, pre-sowing treatment, treatment of perennial grasses);
- loosening to a depth of 20 - 27 sm (autumn plowing, first or last steam treatment, treatment of perennial grasses);
- cultivation of pure steam to a depth of 6 - 8 sm;

- surface treatment of soils to a depth of 4 - 6 sm (early spring loosening of the soil, stubble harrowing, care of crops of perennial grasses);
- stubble disc of soil after harvesting of crops; - cracking;
- chiseling.

For each of these types of soil cultivation in soil conservation agriculture, special machines have been developed.

The working conditions of soil-cultivating machines in soil-protecting agriculture are extremely difficult. They should be efficient with a considerable amount of crop residues on the field surface and operate within a wide range of moisture and hardness of soils of different mechanical composition; The blades of their working organs move in the soil layer, in which the main mass of plant roots is located. These working conditions of soil-cultivating machines are combined with rigid agrotechnological (initial) requirements to them [5, 6].

Thus, for shallow cultivation of soil to a depth of 8 - 16 sm, the deviation from the specified shall not exceed ± 1 sm. At least 85% of the crop residues should be left on the field surface with complete cropping of the weeds. The height of the ridges and the depth of the furrows should not be more than 6 sm. The best quality of soil treatment is obtained with its humidity (16 -21) %, which is approximately (55 -65) % of the total field moisture capacity.

When carrying out small-scale cultivators, it is allowed to keep crop residues on the field surface up to 55%, but the height of ridges and furrows should not be more than 4 sm.

With a deep (20 - 27 sm) loosening of the soil, deviation from a given depth is allowed no more than ± 2 sm, and the weeds must be cut completely. Stubble residues on the surface of the field should remain at least 75%. On the leveled surface of the field, only grooves behind the struts of working organs with a depth of not more than 8 sm are allowed. The soil should loosen mainly on fractions from 2 to 10 sm, and the number of blocks of more than 1 sm does not exceed 20%.

After treatment with rod or rod-paw cultivators, the surface of the field should be level - ridges and furrows not more than 4 sm, weeds are completely cut, the depth and width of the furrow behind the ridges are not more than 4 and 7 sm, respectively.

When surface treatment of the soil with needle discs and other tools to a depth of 4 - 6 sm, the value of individual lumps and ridges should not exceed 5 sm. To protect the soil from erosion it is important that stubble on the field surface remains at least 90% and the number of unstable wind fractions of soil size less than 1 mm. When performing this operation, the height of the comb ridges and the depth of the furrow grooves should not exceed 5 sm, and at least 80% of the weed seeds should be embedded in the soil.

Working in such difficult conditions, tillage machines and soil conservation tools should ensure not only the fulfillment of the specific agrotechnological (initial) requirements specified above, but also reduce energy, material and financial costs. This predetermines the need for a careful choice of the type of working bodies, justification of their geometric parameters and operating modes; arrangement of working elements on the frame of the machine; basic schemes of machines and the justification of their design and kinematic parameters. For the same reasons, the use of existing results of theoretical and experimental research, which is the basis for the creation of conventional tillage machines, is limited in some cases, in others it is unacceptable.

2.2. Types and parameters of operative parts of soil-cultivating machinery and tools of soil-protecting agriculture

2.2.1. Flat-cutting paws

On re-compacted or waterlogged soils with a large number of root-off weeds with fine planar cutting, it is advisable to use lance paws of small width of capture (Fig. 1). Such paws with spring struts are installed on heavy cultivators, for example, KPE-3,8A, KTS-10-1 and KTS-10-2, etc. The choice of the values of their optimal geometric parameters does not cause difficulties, since heavy cultivators are subject to less stringent requirements for spraying the treated soil layer and preserving crop residues on the field surface. Rational values of the geometric parameters of such paws and spring bars are sufficiently substantiated in [13,14,19,25,45] and are given in GOST 1343-82.

It is not recommended to use heavy cultivators with such lancet paws on soils highly susceptible to wind erosion, as they seal up residues in the soil and spray the treated soil layer [25, 27] (Fig. 2).

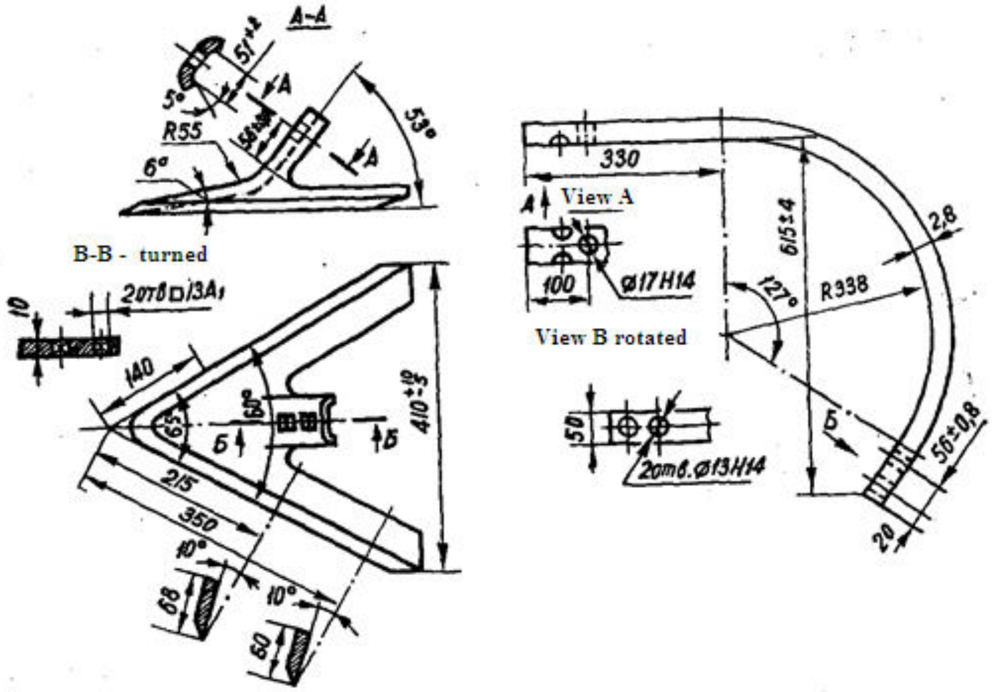


Fig. 1. Working body of heavy cultivators

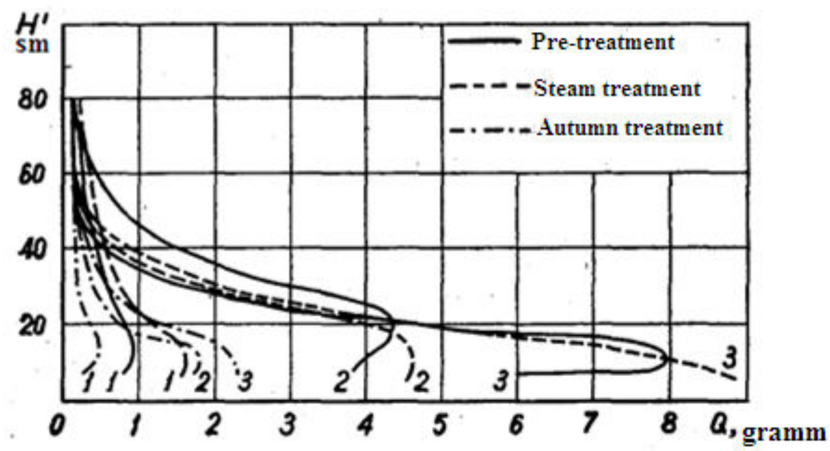


Fig. 2. Removal of erosion soil particles (<1 mm) from the surface of fields,
treated with a heavy cultivator and a flat cutter (sandy loam,t = 5 minutes, air speed 12 m/s);
1-raw stubble; 2 - flat-cutting; 3 - treatment with a heavy cultivator

In connection with this, special cultivators for small (6-8 sm) tillage - tools for preseeding processing OP-6 were developed for tractors of different classes; OP-8; ОП-10; OP-12. They were equipped with a lancet foot with geometric parameters of the working organs of seeder-cultivators, but with a width up to 330 mm. The justification of its parameters and the type of suspension will be given in the section of sowing machines.

The most fully satisfies the conditions of work and specific requirements for the planning of soils, a wide-cut planar paw, which consists of two dihedral wedges placed at an angle to the direction of movement in the plane, having a common edge obtained by crossing two planes (knives) (Figure 3). The transverse profile of the working surfaces of the knife paws can be flat or curved. The main geometric parameters of such paws are: the angle between the installed knife and the bottom of the furrow α, lying in a plane perpendicular to its blade; height of soil formation - h; width of the knife - $l(\widehat{l})$, angle of the blade solution - 2γ; the radius of curvature of the working surface of the knife is *r* and the width of the grip is *2b* *. Geometrical parameters of the paw are interconnected by dependencies

$$\sin\alpha = \frac{h}{l}; \tag{1}$$

$$\operatorname{tg}\alpha_2 = \operatorname{tg}\alpha\sin\gamma, \tag{2}$$

and in the case when the transverse profile of the section of the working surface of the knife has a curved (convex) shape [7, 8]

$$h = r\left[\cos\left(\alpha' - \frac{180\cdot l_1}{\pi r}\right) - \cos\alpha'\right]. \tag{3}$$

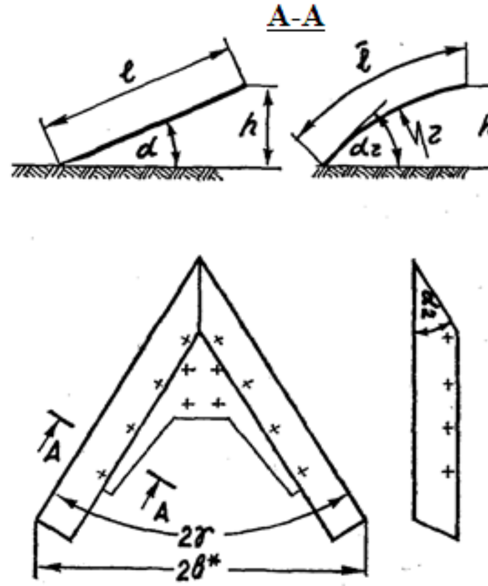


Fig. 3. Basic geometric parameters of the flat-paw

Based on the correlation-regressive analysis of the experimental data obtained in the most characteristic conditions of the operation of plane cutters, the influence of each geometric parameter and the speed of the flat-topped paws on the quality indicators of their work and traction resistance is given. It was found that the stubble conservation on the surface of the field and the spraying of the treated soil layer depend more on the speed of movement (the coefficients of pair correlation are respectively equal to (-0,623 and 0,446), the height of the formation (-0,357 and 0,252), the angle of inclination of the knife to the bottom furrows (-0,357 and 0,421), the angle of the paw solution (-0,159 and 0,154), and to a lesser extent depend on the radius of curvature (-0,0118 and -0,032) and the blade width (0,082 and 0,0209). The largest positive correlation is observed between the traction resistance of the flat-topped paw and the speed of its movement (the coefficient of pair correlation is 0,541), the height of the rise of the formation (0,211) and the angle of the knife setting relative to the furrow bottom (0,531). An inverse correlation is observed between the drag force of the paw and the radius of curvature of the cross section of the profile of the working surface of the knife and its width (-0,137-0,248).

The rationale for the optimal values of these geometric parameters and modes of operation of wide-cut planar paws has not received due attention until recently. Their choice was carried out on the basis of the results of theoretical and experimental studies carried out in substantiating the parameters of passive working elements based on the trihedral wedge used on machines for dumping soil. Comparing the goals of these studies, the methodology and the conditions of their carrying out with specific agrotechnological (initial) requirements and operating conditions of flat-bottom cultivators, it can be stated that the geometric parameters of the paw are optimized mainly from the condition of ensuring minimum traction resistance.

Observations of the operation of flat-topped paws in the field showed that the maximum amount of crop residues on the field surface can be kept and the soil can only be minimized by spraying the parameters and regimes of the soil with the least soil unloading (loading is the ratio of the actual mass of the soil to the knife to the theoretical one). In the future, this was confirmed experimentally - the correlation coefficient between soil piling and stubble conservation is -0,89, and between loading and spraying of the treated soil layer +0,85 [7,8]. The difference in the amount of actual soil mass on the knife and the theoretical is explained by the fact that as the angle of inclination to the bottom of the groove of the working surface of the paw, its length and the height of the formation increases, the relative velocity of soil movement along the working surface (v_r) decreases in comparison with the translational velocity of the paw (v_n) and reaches a value at which the soil begins to accumulate on the knife (Fig. 4).It has been proved [7,8] that theoretically and experimentally the soil loading can be quantitatively estimated by a coefficient equal to the ratio v_n/v_r.

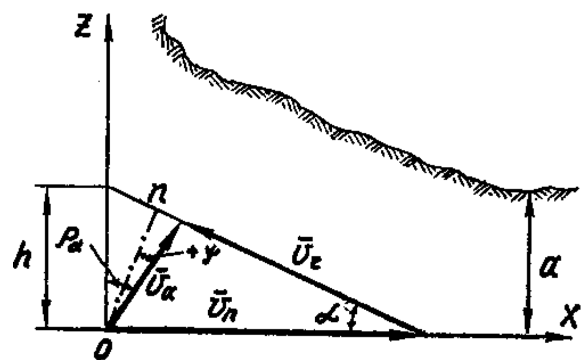


Fig. 4. Soil immersion by labour body

In order for the paw to not unload the soil, the cut under it should create the necessary support (Q), ensuring the movement of the soil along the working surface. The direction of the support was taken parallel to the working surface of the paw, and its magnitude was determined by the formula

$$Q = a \cdot b \cdot \sigma_{\text{сж}}, \tag{4}$$

where: a – depth of loosening;
b - the width of the claw;
$\sigma_{\text{сж}}$ - ultimate resistance of the soil to compression.

However, studies [9, 29] have established that the stress on soil compression in the cross-section of the undercut is distributed unevenly: the smallest stress will be at the boundary with the working surface of the paw and on the surface, and the maximum on 0,25a of the working surface. Since the movement of the element of the soil layer takes place in the direction of absolute velocity, the ($\bar{v}_a$)cross-sectional area on which the element is supported will increase by $1/\cos\rho_a$. The equal force of the backwater (the reaction of the undeformed soil) is rejected (Fig. 5) from the normal to $\bar{v}_a$ на ythe inner friction head φ_1 and for the paw with $2\gamma = 180°$ is equal to

$$Q_{\alpha_0} = 0{,}5 \cdot a \cdot b \cdot \sigma_{\text{сж}} / \cos\rho_a \cos\varphi_1, \tag{5}$$

and for the paw with $2\gamma \neq 180°$

$$Q_{\alpha'} = 0{,}5 \cdot a \cdot b^* \cdot \sigma_{\text{сж}} / \cos\rho_{\alpha'} \cos\varphi_1 \sin\gamma, \tag{6}$$

where: α'-is the actual cutting angle of the flat-paw, the formula for which will be given below.

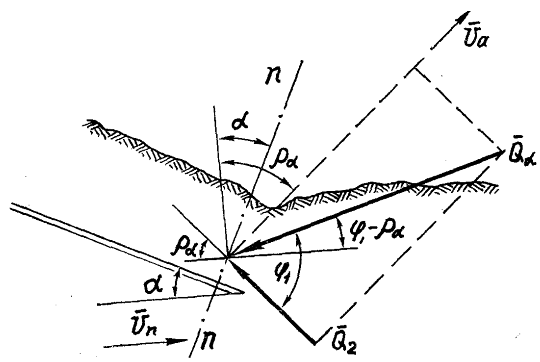


Fig. 5. The support created by the undeformed soil lying ahead of the labour body

Increase the amount of backwater, depending mainly on the physical and mechanical properties of the soil and the specified depth of loosening, is not possible. Therefore, to ensure the movement of the soil along the working surface without reloading, one should strive either to reduce the angle of inclination of the knife to the bottom of the furrow or the height of the rise of the formation, or to reduce the mass of soil on the knife by reducing its length (width). The phenomenon of unloading was not taken into account when justifying the optimal values of the geometric parameters of the working organs of tillers, since in some cases this is not a determining condition (for example, when the soil is treated to a depth of more than 20 sm), in others - they were not subjected to stringent requirements for spraying the treated layer soil and the conservation of crop residues on the surface of the field. Consequently, the optimal values of the geometric parameters of the flat-topped paws and their operating modes should be selected from the condition that they ensure a minimum soil unloading.

The working conditions of tillage machines and soil farming tools are characterized by the fact that they must work both at low (pre-sowing treatment, cultivation of stubble fallow land), and with increased hardness of soils (treatment of plowland and a layer of perennial grasses). Therefore, the geometric parameters of the planar paws and the weight of the machine must ensure that it is well recessed. The latter can be increased with ballast if the mass of the vehicle is insufficient. But this is an extreme measure, which leads to unnecessary energy costs and complicates the operation of the machine. Therefore, the optimal values of the geometric parameters of the flat-topped paws, found by the criterion of minimum soil unloading, must be linked with the condition of ensuring the best reclamation.

The deepening capacity of labour organs (n^0) in agricultural mechanics is usually characterized by the ratio of vertical R_z to horizontal (R_x) the components of the total resistance acting on the part of the soil when moving them.

The methodological foundations of the definition $R_x$and R_z are sufficiently reflected in the existing literature. However, the special conditions for the operation of flat-topped paws and the agro-technological (initial) requirements to them predetermine the need for some refinement. The dependences proposed by the researchers for determining $R_x$and R_z sharply differing from each other, which is explained not only by the different views of the authors on the prevailing form of soil deformation under the influence of the labour organ and the complexity of the processes occurring in this process, but also by the fact that some of them take the cutting angle of the trihedral wedge equal to the setting angle its working surface to the bottom of the furrow -α, and the others to the angle $\alpha_2$lying in the longitudinal-vertical plane. Actually, the actual cutting angle (α') only lies within these limits. Its magnitude is determined by the direction of the relative velocity of soil movement along the working surface of the knife. Practice confirms that the direction of movement of soil along the knife does not coincide with the direction of movement of the paw, nor with the direction perpendicular to its blade.

When the component of the soil resistance is found by the inertia of the formation, the value of the absolute velocity of the formation movement was determined on the basis of the condition of equality of its translational and relative rates of soil movement along the working surface, that is, under the condition of motion of an absolutely incompressible elastic bed. In the real condi-

tions$|v_n| \neq |v_r|$. In this connection, it is proposed to find the formula for determining the component of the traction resistance of the working member due to the inertia force from the following considerations [7, 8].
To determine the magnitude and direction of the force $F_{\alpha_0}^u$ acting on the flat-cut foot with 2γ=180°at the time of changing the direction of movement of the soil layer, depending on the angle of inclination of the knife to the bottom of the furrow, we use the Euler theorem. Let the paw move with speed $v_n$and cuts a layer of soil with thickness $a$and widths $b(a \text{ x } b)$ (Fig. 6). Then the paw can be regarded as fixed, on which the bed of the soil runs, changing its direction and, possibly, the speed of movement from $|v_n|$ до $|v_r|$. Moreover $|v_r| < |v_n|$, the thickness of the layer increases. The reasons for this are forces opposing the movement of the soil layer up the working surface of the knife (the forces of external and internal friction, the component of gravity). Suppose that in section 1-1 the thickness of the soil layer is equal$a$, and in the cross section 2-2 is a_1. With the steady movement of the soil along the working surface of the paw, the soil masses traveled per unit time through sections 1-1 and 2-2 must be equal to the soil masses, otherwise there will be continuous accumulation of the soil. Therefore, $m_1 v_n = m_2 v_r = M_c$ or $ab\rho v_1 = a_1 b\rho v_2 = M_c$, whereM_c – the secund soil mass; ρ – the soil density.

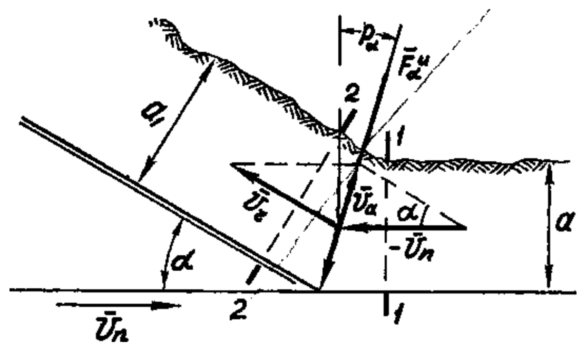


Fig. 6. The magnitude and direction of the force Pacting on the paw from the side of the soil layer when the direction of its movement changes.

By Euler's theorem, the change in the second momentum is equal to the sum of the external (volume and surface) forces acting on the flow

$$\bar{F}_{\text{об}} + \bar{F}_{\text{пов}} = M_c \bar{v}_r - M_c \bar{v}_n. \tag{7}$$

Let us assume that the length of the arc 1-2, on which the moving soil layer changes its momentum, is small. Consequently, the mass of this volume of soil is small and $\bar{F}_{\text{об}} \approx 0$. Thus $\bar{F}_{\text{пов}} \approx M_{\text{c}}(\bar{v}_r - \bar{v}_n)$. Since $\bar{v}_r = \bar{v}_n + \bar{v}_a$, thus $\bar{F}_{\text{пов}} = M_{\text{c}} \cdot \bar{v}_a$, a $|v_a|$ (see Fig. 4) is determined from formula

$$|v_a| = |v_n| \sin\alpha_0 / \cos(\rho_{\alpha_0} - \alpha_0) \, . \tag{8}$$

With some assumption it can be assumed that $|F_a^u| \approx |F_{\text{пов}}|$, then

$$|F_a^u| = a \cdot b \cdot \rho \cdot \frac{v_n^2 \sin\alpha_0}{\cos(\rho_{\alpha_0} - \alpha_0)} \, . \tag{9}$$

In the special case, when $|v_n| = |v_r|$ and $\rho_{\alpha_0} = \alpha_0/2$, i.e. when moving of an incompressible elastic bed of soil, we obtain the well-known formula $|F_a^u| = 2 \cdot a \cdot b \cdot \rho \cdot v_n^2 \sin\alpha_0 / 2$.
For a flat-paw with $2\gamma \neq 180°$ and the actual cutting angle α', which lies within $\alpha_2 \leq \alpha' \leq \alpha_0$, formula 9 takes the form of

$$|F_{\alpha'}^u| = \frac{ab^* \rho (v_n^*)^2 \sin\alpha \;\; ' \sin\gamma}{\cos(\rho_{\alpha'} - \alpha') \sin^2(\varepsilon_x + \gamma),} , \tag{10}$$

where:ε_x - is the projection of the relative travel speed deviation of angle of soil along the knife of the paw from the direction of its movement ε to the horizontal plane (Fig. 7).

The value of the angle ε_x is found from

$$\varepsilon_x = [arc\,\mathrm{tg}(ctg\delta \cos\alpha_0)] - \gamma\ , \tag{11}$$

where: δ - is the angle characterizing the deviation of the relative velocity of soil movement along the working surface of the paw knife from the perpendicular to its blade.

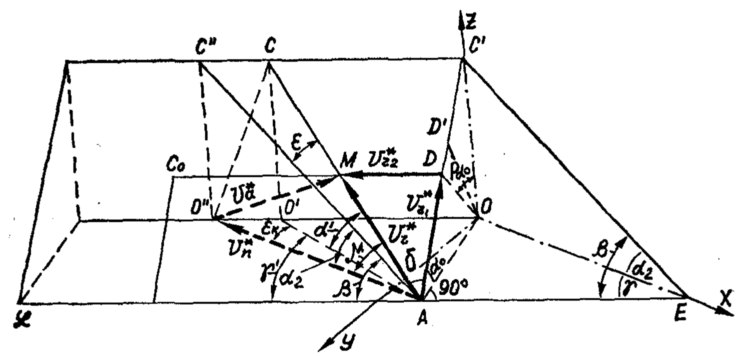


Fig. 7. To the analysis of the work of the flat-paw ($2\gamma \neq 180°$)

To determine the calculated values of the angle δ, as well as the angle ρ_{α_0}, characterizing the direction of the absolute velocity of soil movement under the influence of the knife of the paw (see Fig. 7), the following relationships are proposed:

$$\mathrm{tg}\delta = \frac{f}{f_1+f}\mathrm{ctg}\gamma\cos\alpha_0 + \frac{f}{f_1+f}\sin\alpha_0\,\mathrm{ctg}\gamma \times$$

$$\times\left\{\frac{E_{\alpha_0}\mathrm{tg}\alpha_0 + K_{\alpha_0} - D_{\alpha_0}}{2K_{\alpha_0}tg\alpha_0} \pm \left[\left(\frac{E_{\alpha_0}\mathrm{tg}\alpha_0 + K_{\alpha_0} - D_{\alpha_0}}{2K_{\alpha_0}\mathrm{tg}\alpha_0}\right)^2 - \frac{E_{\alpha_0} + D_{\alpha_0}\mathrm{tg}\alpha_0}{K_{\alpha_0}\mathrm{tg}\alpha_0}\right]^{\frac{1}{2}}\right\}; \tag{12}$$

$$\rho_{\alpha_0}(1{,}2) = arctg\left\{-\frac{E_{\alpha_0}\mathrm{tg}\alpha_0 + K_{\alpha_0} - D_{\alpha_0}}{2K_{\alpha_0}tg\alpha_0} \pm \left[\left(\frac{E_{\alpha_0}\mathrm{tg}\alpha_0 + K_{\alpha_0} - D_{\alpha_0}}{2K_{\alpha_0}\mathrm{tg}\alpha_0}\right)^2 - \right.\right.$$

$$\left.\left. - \frac{E_{\alpha_0} + D_{\alpha_0}\mathrm{tg}\alpha_0}{K_{\alpha_0}\mathrm{tg}\alpha_0}\right]^{\frac{1}{2}}\right\}, \tag{13}$$

where:

$$\mathrm{E}_{\alpha_0} = \mathrm{A}_{\alpha_0} + \mathrm{B}_{\alpha_0}\sin(\alpha_0+\varphi_1) - C_{\alpha_0}cos\,(\alpha_0+\varphi_1);$$

$$K_{\alpha_0} = -\mathrm{B}_{\alpha_0}\cos\,(\alpha_0+\varphi_1) - C_{\alpha_0}sin\,(\alpha_0+\varphi_1)$$

$$B_{\alpha_0} = \frac{0{,}5\cdot a\cdot b^*\sigma_{\text{сж}}(f_1+f)}{\sin\gamma\cos\varphi_1};$$

$$C_{\alpha_0} = \frac{0{,}5\cdot a\cdot b^*\sigma_{\text{сж}}}{\sin\gamma\cos\varphi_1};$$

$$D_{\alpha_0} = [ab^*\rho\,(\vartheta_n^*)^2\sin\alpha_0]\cdot\sin\gamma;$$

$$A_{\alpha_0} = \frac{a\, b^* \sigma_{сд}}{\sin\gamma \sin\theta_{\alpha_0}}\left[(f_1+f)\sin\left(\alpha_0+\theta_{\alpha_0}\right)-\cos\left(\alpha_0+\theta_{\alpha_0}\right)\right]+$$

$$+al\rho g\left(\frac{b^*}{\sin\gamma}-0{,}51\, l ctg\ \beta\right)\left[(f_1+f)\cos\alpha_0+\sin\alpha_0\right]+$$

$$+a\rho\ (v_a^*)^2(f_1+f)b^*\sin\alpha_0\sin\gamma;$$

where: $f, f_1, \varphi,\ \varphi_1$- respectively, the coefficients and angles of external and internal friction of the soil;

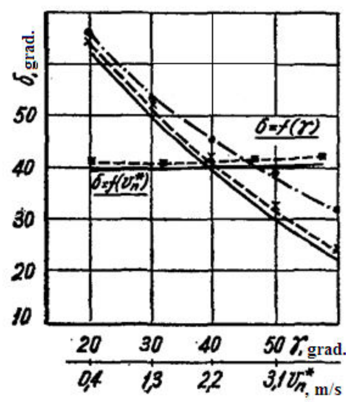
$\sigma_{сж}, \sigma_{сд}$ - soil resistance to compression and shear;

g - acceleration of gravity;

ρ - density of soil;

θ_{α_0}- the shear angle of the formation element, which is found from the known relationship $\theta_{\alpha_0} = \frac{\pi}{2} - \frac{\alpha_0+\varphi+\varphi_1}{2}$;

β - the angle of the cutter's knife and β = arctg (tgγ/ cos α_0).



—— - design;
– – – - experimental, obtained in soil channel;
– ·–·–experimental, obtained in the field

Fig. 8. The dependence of the angle δ on the angle of the paw solution and its velocity translational motion - for δ = f(γ): α = 6 sm, v = 2,1 m/s, α_0=25°; for δ=$f(v_n^*)$: 2γ= 180°, α_0=25°, h=35 mm.

The reliability of the proposed dependence (12) was confirmed experimentally (Fig. 8): $\delta = f(\gamma)$ and $\delta = f(v_n^*)$. In the field, the experimental values $\delta = f(\gamma)$ differ slightly from the calculated values, which is apparently due to the presence of plant roots in the treated soil layer and the worst conditions for their descent from the knife-blade blades.

The experimental verification of the dependence for the determination of the angle ρ_{α_0} was made indirectly through an angle ψ, since there is a relationship between them (see Fig. 4)

$$\psi = \rho_{\alpha_0} - \alpha_0;\ \ -\psi = \alpha_0 -\ \rho_{\alpha_0}.$$

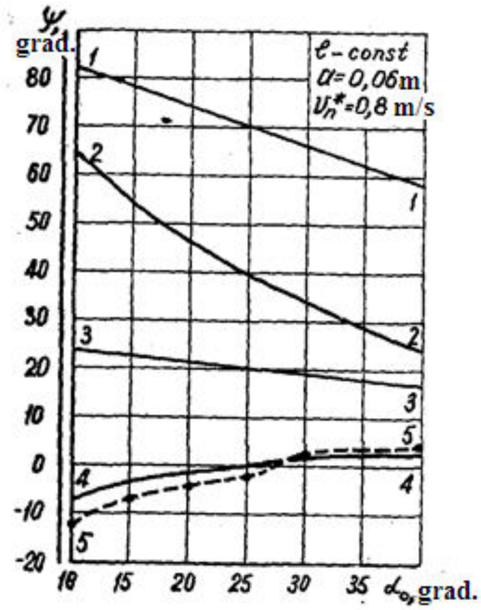


Fig. 9. The change in the angle ψ characterizing the direction of the absolute displacement velocity of the soil as a function of the angle α_0:
1-1 - according to N.V. Grischenko;
2-2 - according to M.M. Severnev;
3-3 - according to M.E. Demidko;
4-4 - according to the proposed formula (13);
5-5-experimental

From Fig. 9 is seen that the dependence (13) in comparison with those proposed by other researchers is more in line with the experimental data.
Formulas (12) and (13) allow us to derive a relationship for determining the relative rate of soil movement along the working surface of the knife of the paw

$$v_r^* = \frac{v_n^* \cos\gamma}{(f_1+f)(\cos\alpha_0+\operatorname{tg}\rho_{\alpha_0}\sin\alpha_0)}\left[(f_1+f)^2\operatorname{tg}^2\gamma + f_1^2\left(\cos\alpha_0+\operatorname{tg}\rho_{\alpha_0}\sin\alpha_0\right)^2\right]^{\frac{1}{2}},$$

and, consequently, a formula for finding the coefficient of soil unloading by a planar paw

$$k_\gamma = \frac{(f_1+f)\left(\cos\alpha_0+\operatorname{tg}\rho_{\alpha_0}\sin\alpha_0\right)}{\left[\sin^2\gamma(f_1+f)^2 + f_1^2\cos^2\gamma\left(\cos\alpha_0+\operatorname{tg}\rho_{\alpha_0}\sin a_0\right)^2\right]^{\frac{1}{2}}}. \quad (14)$$

For a paw with an angle 2γ = 180° this formula takes the form of

$$k_{\alpha_0} = \cos\alpha_0 + \sin\alpha_0 \operatorname{tg}\rho_{\alpha_0} \quad (15)$$

and it will be possible to determine the effect of тonly the angle of inclination of the working surface of the knife relative to the bottom of the furrow on soil unloading.
The results of calculated and experimental values of the coefficients of soil unloading as a function of the angle of inclination of the blade knife $k_{\alpha_0} = f(\alpha_0)$, the angle of the solution$k_\gamma = f(\gamma)$ and the speed of its movement $k_\upsilon = f(v_n^*)$are shown in Figures 10, 11 and 12, respectively.

The actual cutting angle of the plane-cut paws is determined by the ratio

$$\operatorname{tg}\alpha' = \sin\alpha_0\left[\left(\frac{f}{f_{1+f}}\right)\operatorname{ctg}^2\gamma\left(\cos\alpha_0+\operatorname{tg}\rho_{\alpha_0}\sin\alpha_0\right)^2 + \cos^2\cdot\alpha_0\right]^{-\frac{1}{2}}. \quad (16)$$

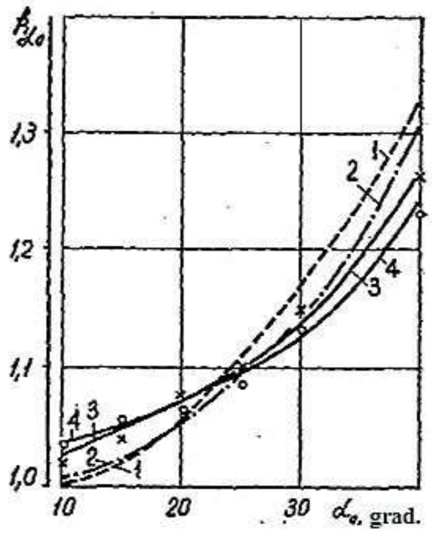


Fig. 10. Soil loading depending on the angle of inclination of the paw knife relative to the bottom of the furrow (a=6 sm, $v_n^* = 0{,}8$ m/s)

1– – –1 –the estimated;
3–x–x-3 – experimental;
***l* - const**

2-•-•-2 – the estimated;
4 - - -4 – experimental.
***h* - const**

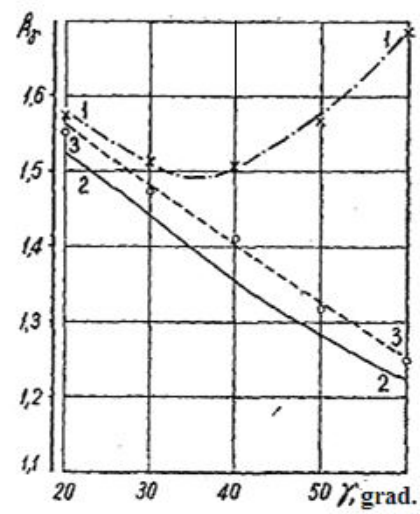


Fig. 11. Soil pumping with the paw, depending on the angle of its solution (a= 6 sm; v_n^* =2,1 m/s).

1 – ·x –·x-1 – experimental (soil channel);
2 —— 2 – the estimated;
3 – -o- -o-3 –experimental (field conditions).

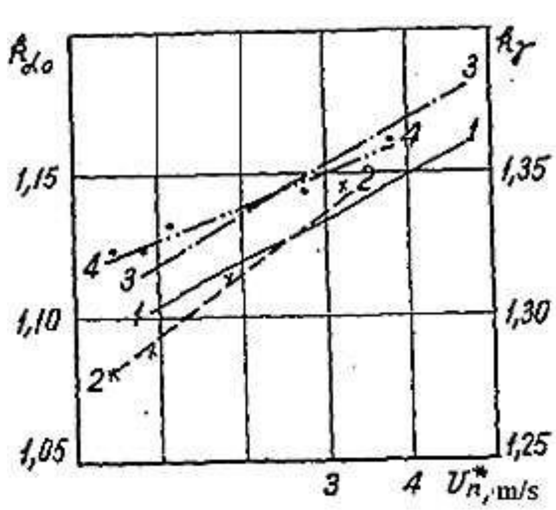


Fig. 12. Loading of soil by a paw
(1,2)– $k_{\alpha_0} - 2\gamma = 180^0$;
(3,4) $- 2\gamma \neq 180^0$ depending on speed of movement - for $2\gamma = 180^0$:α_0=25°, h=35 mm, l =100 mm; for $2\gamma \neq 180^0$: $2\gamma = 80^0$, h=35 mm, α_0=25^0, l=80 mm.
1——1- the estimatedwedge;
2– –x– –x–2 experimental wedge;
3–·–·–·–3–the estimatedpaws;
4 –··x–··x–4 experimentalpaws.

Taking into account the coefficient of soil unloading by the planar paw, its actual cutting angle at $2\gamma \neq 180^{\circ}$, and also assuming that the resistance of the rack can be neglected, since its thickness is 1 - 3% of the width of the claw of the paw and it works in the loosened layer of the soil, the horizontal and vertical components of the total soil resistance of the paw it is proposed to determine by the formulas:

$$R_{\gamma x} = ib^{*}q(1 + \operatorname{ctg}\gamma \operatorname{tg}\varphi) + \frac{ab^{*}\sin(\alpha'+\varphi)\sin(\varepsilon_x+\gamma)}{\cos\varphi \sin\gamma} \times \left\{\frac{0{,}5\, Q_{\text{сж}} \sin(\alpha'+\varphi_1-\rho_{\alpha_0})}{\cos\rho_{\alpha_0}\cos\varphi_1 \sin\gamma} + \right.$$

$$\left. + \frac{\sigma_{\text{сд}}\sin(\alpha'+\theta_{\alpha'})}{\sin\theta_{\alpha'}\sin\gamma} + l\rho q k_{\gamma} \times \left(\frac{1}{\sin\gamma} - \frac{0{,}5l \operatorname{ctg}\beta}{b^{*}}\right)\cos\alpha' + \frac{\rho(v_n^{*})^2 \sin\alpha' \sin\gamma}{\sin^2(\varepsilon_x+\gamma)}\right\}; \quad (17)$$

$$R_{\gamma z} = \frac{ab^{*}\sin(\alpha'+\varphi)}{\cos\varphi}\left\{\frac{0{,}5\,\sigma_{\text{сж}}\sin(\alpha'+\varphi'-\rho_{\alpha_0})}{\cos\rho_{\alpha_0}\cos\varphi_1\sin\gamma} + \frac{\sigma_{\text{сд}}\sin(\alpha'+\theta_{\alpha'})}{\sin\theta_{\alpha'}\sin\gamma} + l\rho q k_{\gamma} \times \right.$$

$$\left. \times \left(\frac{1}{\sin\gamma} - \frac{0{,}5l \operatorname{ctg}\beta}{b^{*}}\right) \times \cos\alpha' + \frac{\rho(v_n^{*})^2 \sin\alpha' \sin\gamma}{\sin^2(\varepsilon_x+\gamma)}\right\}, \quad (18)$$

where: i – knife blade thickness;

$$\theta_{\alpha'} = \frac{\pi-(\alpha'+\varphi+\varphi_{\prime})}{2};$$

q - specific soil resistance to the introduction of the paw blade;
l - the length of the knife blade.
At $2\gamma = 180°$, formulas 17 and 18 are transformed into dependencies that allow one to analyze the influence of the angle of inclination of the knife relative to the furrow bottom on R_{xa_0} and R_{za_0}, therefore if $\gamma = 90°$, then $\alpha' = \alpha_0$; $\operatorname{ctg}\gamma = 0$; $\varepsilon_x = 0$; $\sin\gamma = 1$; $k_\gamma = k_{\alpha_0}$.
For the operation of flat-cutting tools at speeds up to 12 km / h, the values of the geometric parameters of the feet should lie within the following limits: the angle between the knife and the bottom of the groove is 22 - 27 °, the angle of the groove is 60 - 80^0, the height of the formation is 25 - 35 mm, the width of the knife 75 - 80 mm, the shape of the working surface is rectilinear (the radius of the transverse profile of the knife is equal to infinity).
Increasing the width of the claws helps improve the quality of their work, since with a smaller number of racks per 1 m of gripper, more remains on the surface of the field of stubble residues and less sprayed the treated soil layer. However, with an increase in the width of the claw, its deeper ability decreases, which follows from an analysis of the relationship between $R_{\gamma z}/R_{\gamma x}$ the results of the experiment (Table 1).
Analysis of the data of Table 1 shows that the paws, having a working width of 700 - 1100 mm, satisfy the agrotechnological (initial) requirements for stubble conservation on the field surface and have sufficient deepening ability. When carrying out certain technological operations, for example, during pre-sowing treatment, when the soil moisture content is high and the field surface is rapidly covered by shoots, the width of the claw grip can be reduced to 300-500 mm.

Table 1 - Influence of the width of the capture of flat-topped paws on stubble conservation and strength characteristics

Width of capture of paws, 2 b*, mm	The amount of stubble on the surface of the field, pcs / sq. m		Percent of preservation of stubbles on the surface of the field	Indicators of power characteristics		
	before passage	after passage		$R_{\gamma x}$, кН	$R_{\gamma z}$, кН	$R_{\gamma z}/R_{\gamma x}$
500	154 ±5	100 ± 3	65,0	0,76	0,44	0,58
760	254 ±7	184±5	72,5	1,0	0,54	0,54

1100	214 ± 6	167±4	79,3	1,14	0,58	0,51
1400	210±7	181 ±5	86,2	1,43	0,64	0,45

The cut-off paw (Fig. 13), having the values of the above-mentioned geometric parameters, was installed on: KPN-4, KPSh-5, KPSh-9, KPSh-11 flat-top cultivators; tools for the non-perforated treatment of a layer of perennial grasses -OPT-3-5; flat-cutting cutters PSh-3 and PSh-5; universal soil-cultivating tools YPO-4.

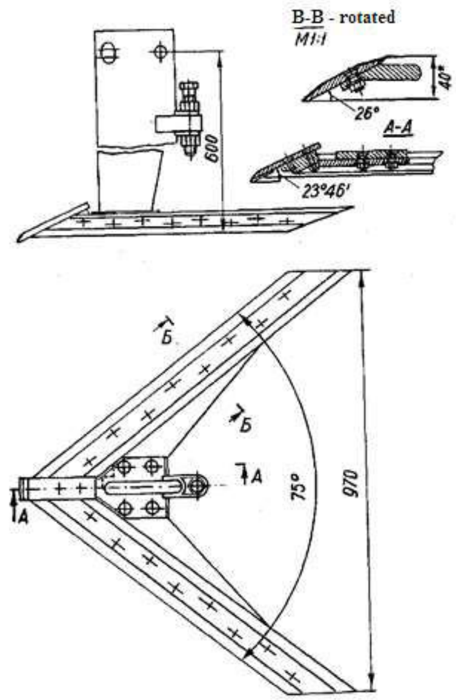


Fig. 13. The working body of KPN-4, KPSH-5, KPSh-9, KPSh-11, OPT-3-5flat-top cultivators, PSh-3, PSH-5and YPO-4 slitting cutters

The working conditions of machines when processing soils without a turn of the formation to a depth of 20 - 27 sm are slightly different from the working conditions of flat-top cutters intended for small (8 - 16 sm) loosening of soils. Thus, a large depth of loosening provides sufficient soil support, necessary to move the paw of the cut formation without reloading and completely cutting the roots of weed plants with knife blades. This allows to increase the angle of the paw solution without any detriment to the quality of the work. To the deep loosers, there is an increased requirement to ensure a sufficient degree of crumbling of the crop undercut by a considerable thickness of soil, which determines the need to increase or increase the height of the bed of the paw or the angle of inclination of the knife. In addition, the large energy intensity of the process of deep loosening of the soil without turnover of the bed predetermines the need to create a sufficiently robust construction of the feet of the deep loosener, which is most easily achieved with a large angle of the paw solution and the height of the formation. Therefore, when choosing the optimal values of the basic geometric parameters of the paws for the deep loosener, the loading phenomenon is not taken into account, and they are selected from the condition of ensuring a sufficient degree of crumbling of the soil, minimum traction resistance and best burying ability.

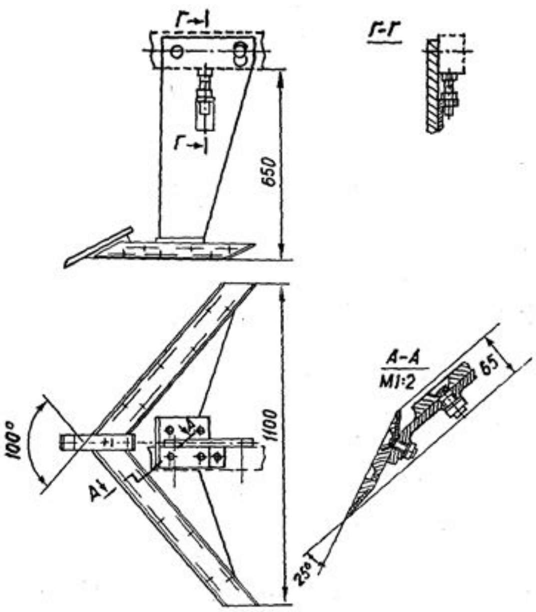


Fig. 14. Labour body of flat-cutting planers KPG-250, KPG-250A, PG-3-5, PG-3-100 and universal tillage tool YPO-4

Proceeding from this, the following values of their geometric parameters must be taken for the paws of the deep loosener: the angle of the paw solution is 100°, the pitch of the ploughshare is 25° relative to the bottom of the furrow, the height of the formation is 65 mm, and the blade width is 155 mm [33]. The construction of such a paw is shown in Fig. 14 and is used on flat plows-deep loosersPG-3-5, KPG-250A and PG-3-100, as well as on flat-top fertilizer plows KPG-2,2 and GYN-4 and on universal soil-cultivating toolsYPO-4.
To calculate the traction resistance of the flat-groove paw of the deep plow, the relation [30,33] is derived:

$$R_{\gamma x} = R_x^{\text{ст}} + R_x^{\text{деф}} + R_x^{\text{тр}} + R_x^{\text{лезв}} + nR_x^{\text{б}}, \tag{19}$$

where: $R_x^{\text{ст}}, R_x^{\text{деф}}, R_x^{\text{тр}}, R_x^{\text{лезв}}$ - horizontal components of the forces of resistance, due to the stable, deformation of the formation, separation, transportation of the formation along the working surface of the knife, introduction of blades into the soil;

$R_x^{\text{б}}$- horizontal component of the resistance force of destruction of the soil in the lateral limits of the furrow (capture of the paw);
n – an indicator that takes into account working conditions (when $n = 2$ unfree working conditions, $n = 1$ semi free and $n = 0$ free working conditions).
Analytic determination of the magnitude of the components of the soil resistance forces to a flat three-edged wedge with sufficient support and without taking into account soil unloading does not cause difficulties. The dependencies for their determination have already been derived [30].
To determine the $R_x^{\text{б}}$ the dependence is proposed,

$$R_x^{\text{б}} = 0{,}16\,\pi\cdot a^3\rho\left[\text{ctg}^2\,\Psi_1 + \frac{3\rho_{\text{отр}}\cdot\lambda}{2\cdot\alpha\rho}\right] tg(\alpha_0+\varphi), \tag{20}$$

where: Ψ_1– angle of formation separation in the lateral plane;
$\sigma_{\text{отр}}$ - temporary resistance of soil to tearing;
a – is the depth of loosening of the soil;

λ – coefficient of proportionality, depending on the angle of separation of the formation in the lateral plane, and is found by the formula:

$$\lambda = \cos\ \Psi_1 \operatorname{ctg}\ \Psi_1 \quad \left(0{,}71\ \operatorname{ctg}\ \Psi_1 + \frac{arc\sin\sqrt{1-0{,}5\operatorname{ctg}^2\Psi_1}}{\sqrt{1-0{,}5\operatorname{ctg}^2\Psi_1}}\right). \tag{21}$$

Values λ, calculated by this formula for different values Ψ_1(35°, 40°, 45°, 50°, 55°и 60°), are equal, respectively: 2,34; 1,73; 1,25; 0,93; 0,68; 0,49.

The strength of the lateral resistance in an explicit form depends on the depth of loosening in the cube (see formula 20) and does not depend on the width of the claw.

At a constant depth of loosening $R_x^б$remains constant throughout the entire range of variation of the width of the claw (400 - 1600 mm), and the specific weight of the lateral force in the total volume of its resistance decreases with increasing capture width, which is advantageous from the point of view of energy expenditure. However, an increase in the width of the clamping of the paw, as indicated above, leads to a decrease in the penetrating ability of the paw.

When designing planar cutter-deep plows to determine their traction resistance, it is expedient to use the rational formula of Academician Goryachkin V.P., having determined from the results of their testsin different regions of the country for the main types of work the averaged values of the coefficients included in k u ε.

As a result of long-term tests of flat-cutting plows with the same working organs, a large amount of experimental material has been accumulated on machine testing stations located in different soil and climatic conditions according to their specific traction resistance, depending on the type of work, soil type, physical condition, depth of loosening and speed. These data determine the coefficients k and ε. rational formula of Academician Goryachkin V.P.

Analysis of the averaged values of the indicators characterizing the working conditions of flat-cutting planers in the areas of activity of different MIS over a 10-year period shows that in the 0 - 30 sm layer soil moisture (carbonate chernozems) was significantly higher in steam treatment than in the fall (55,4 - 81,7)% and (13,9 - 72,0)% of PPV, and the hardness is accordingly lower (0,03 - 3,67) and (0,14 - 4,17) MPa.

The material consumption of flat-top cutters at 1 m of the width of the gripper ranges from 226,2 to 343,4 kg. Therefore, in order to calculate the traction resistance of these guns at 1 m of their capture width, the average value of their material capacity can be taken as 270 kg/m.

Specific tractive resistance of flat-top cutters with steam processing (curve 1) and autumn plowing (curve 2) at depths of 15 - 20 sm (average for calculation of 17,5 sm), 20 - 25 sm (average for calculating 22,5 sm) and 25 - 30sm (average for the calculation of 27,5sm) depending on the speed of movement is shown in Fig. 15.

These dependences are based on the averaged data of the specific traction resistances, the values of which were in the intervals of velocities from 1,4 - 1,5 m/s, 1,5 - 1,6 m/s, etc., up to 2,9 - 3, 0 m/s for each depth of loosening. In each interval of the depth of loosening and speed, the averaged point plotted on the graph was determined by not less than five values of the specific traction resistance. When processing steam and fallow at a depth of 17,5 sm, specific traction with increasing speed increases more intensively than at depths of 22,5 and 27,5 sm.

This is explained, in all probability, by the large expenditure of energy on the rejection of the small thickness of the undercut soil, rather than on its deformation. With an increase in the thickness of the crop undercut without a turn of the bed, the spread of the soil by the working organs decreases, but the energy expenditure on its deformation increases. In this case, the difference between the values of the specific traction resistances during the processing of steam and fallow at a depth of 17,5 sm reaches a significant value. When processing steam and fallow for a great depth, this difference decreases, and at a processing depth of 27,5 sm reaches the smallest value. This regularity is explained by the fact that according to the accepted technology of processing pure stubble steam in these zones, several of its first treatments are carried out to a depth of 8 - 14 sm and only the last one - to a depth of more than 20 sm. In this regard, the hardness of the soil in the 20 - 30 sm layer the steam and the fall-out become almost the same.

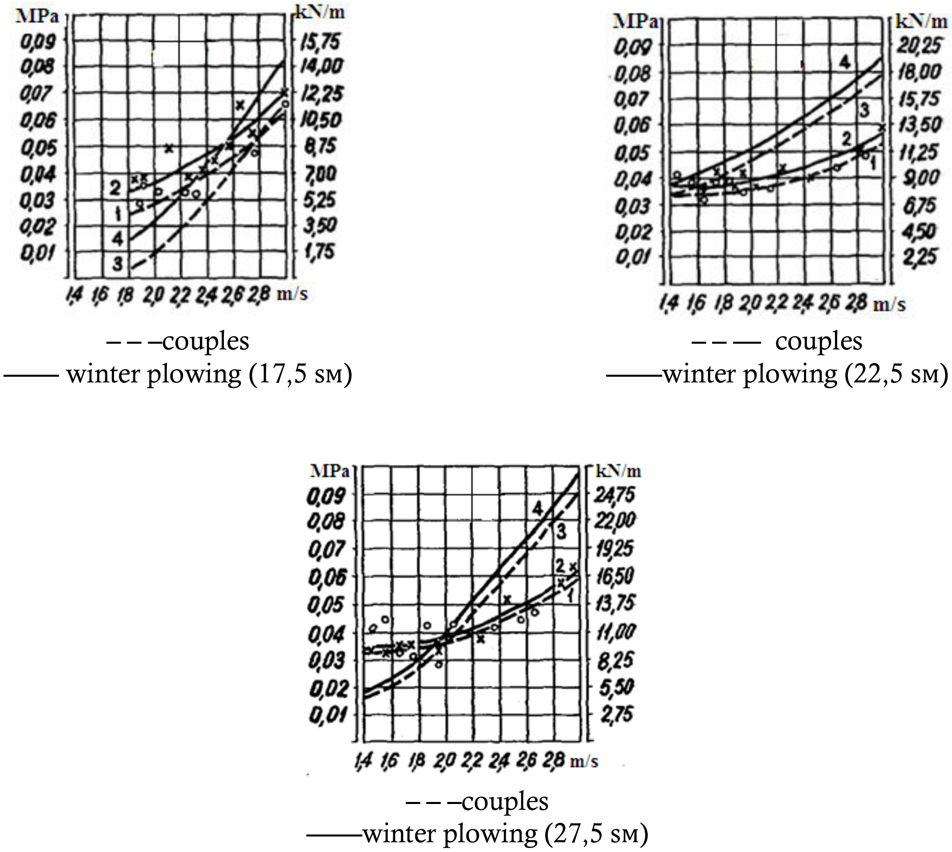


Fig. 15. Specific tractive resistance of flat-top cutters-deep-loosers when processing steam and autumn plow at a depth of 17,5; 22,5 and 27,5sm (1,2 - MPA, 3,4 - kN/m)

Knowing the average specific traction resistance in the processing of steam and autumn plowing at different mean depths of loosening by deep-ripper plows for operating speed values with an interval of 0,2 m/s, the values of specific traction resistances per 1 m of the tool grip width (see Fig. 15, curves 3, 4). These graphical dependencies of the specific traction resistance on 1 m of the width of the tool grip on the speed of motion were approximated by the rational formula of Academician Goryachkin V.P. [12].

The valuesof the coefficients k and ε for each of the average calculated depths of loosening and the type of soil cultivation were the least squares method (Table 2). The difference between the experimental values of the specific resistance of flat-ripper-deep loosers per 1 m of the capture width and the calculated values found by the rational formula of Academician Goryachkin V.P. with the values of the coefficients k and ε given in Table 2 does not exceed 5%.

Table 2 - Values of the coefficients k and ε for determining the traction resistance of the planter-deep ripper according to the rational formula of Academician Goryachkin V.P.

Depth of loosening, sm	Type of treatment of soil	k,H/m^2	ε, Hs^2/m^4
17,5	fallow	1240,46	6182,90
22,5	=//=	23849,90	2900,00
27,5	=//=	20536,10	4031,56
17,5	autumn-plowed	5198,30	6771,70
22,5	=//=	25918,70	2933,90
27,5	=//=	21175,60	4156,40

Analysis of the values of the coefficients k and ε shows that when processing steam and fallow at a depth of 17,5 sm, the value of the coefficient ε, taking into account the influence of the speed of movement on soil resistance is much larger than k due to deformation of the soil formation. When the soil is treated to a depth of 22,5 sm and 27,5 sm, on the contrary, k is greater εthan, as indicated above, a more intensive increase in the specific traction resistance with increasing speed of the unit at a shallow depth of loosening.

2.2.2. Chisel operative tools

In recent years, world-wide practice has been widely used in the cultivation of chisel type tillers. This is due to the fact that their use can reduce excessive soil compaction and increase its infiltration and erosion properties, as well as reduce fuel costs when replacing plowing plowing.
Agrotechnological (initial) requirements envisage the creation of chisel plows as universal machines operating in a wide range of loosening depths (20 - 45 sm), widths of traces (30, 40 and 50 sm) and which can be equipped with other types of working tools.
The working body of the chisel plow is a straight or curved stand with a removable tool, and sometimes with a pointed paw. The quality and energy intensity of the loosening of the soil by the chisel working organ mainly depend on the parameters of the nosepiece and the post [20,21].
The nosepiece acts on the soil layer similarly to the dihedral wedge. At the same time, the front working elements operate in conditions of blocked cutting. The profile of the loosening zone of the soil by the chisel working organs in the transverse-vertical plane is shown in Fig. 16. Qualitative loosening with the lowest energy costs occurs under the condition that the given working depth H does not exceed the value of the so-called critical cutting depth H_{cr}. Otherwise, the zone of deformation of the soil decreases, and the energy intensity of the process increases.

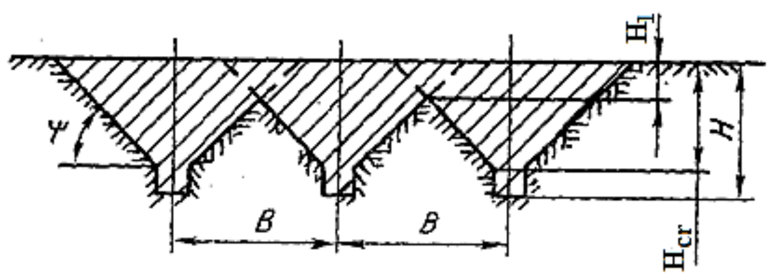


Fig. 16. The profile of the loosening zone of the soil by the working Bodiesof chisel plow

The working element, moving below the critical depth of cut, creates an additional soil compaction. Therefore, to ensure high-quality soil cultivation with a minimum of energy consumption, the parameters of the chisel organ are chosen such that the required depth of loosening does not exceed the critical depth of cutting, which is the boundary of the transition zones of the cutting process with the separation of the chips when the soil is crushed by the nosepiece.
At this depth, the resistance of the soil to the separation of the elements of the formation is equal to the resistance of the soil to crushing. Assuming that separation of the chips under the influence of a dihedral wedge at a greater depth of cutting occurs by separation, the dependence of the critical cutting depth on the physico-mechanical properties of the soil and on the parameters of the nosepiece is found.
When the chisel working element is operating in the soil monolith, the resistance of the formation to the detachment of the doctor under the condition of a maximum of normal stresses is determined by the dependence [20, 21]

$$R_{xr} = \frac{2\,\sigma_{\text{отр}}\,kH\,(b + 0{,}5\,\pi\,kH + 2a)}{\sin\theta + \mu\cos\theta}\,, \qquad (22)$$

where: $\sigma_{\text{отр}}$ – temporary soil resistance to stretching;

H – depth of loosening;
b – width of wedge;
a – the length of the path traversed by the wedge until the moment of detachment of the element of the soil;
θ – the angle of inclination of the resultant forces of soil resistance to the horizon;
μ – Poisson's ratio (0,30 - 0,35);
$k = (H - h)/h$ = 0,70-0,87 – coefficient characterizing the ratio of the depth of the zone of cleavage of the soil to the depth of the wedge;
h – the minimum height of the wedge lift, necessary for chipping the soil with a thickness equal to the depth of loosening H.
The strike angle ψ of the formation in the longitudinal and lateral sections, in which the maximum normal stresses are observed, can roughly be assumed equal to 45°.
The resistance of the soil to wedge crushing is determined by the equation

$$R_x = b\,h\,T, \tag{23}$$

where: T – specific resistance to crushing.

For approximate calculations, the value T can be taken from a solidogram recorded using a standard technique using a hardness meter.
Equating the forces of resistance to the crushing and tearing of the seam of the soil with a dihedral wedge and substituting the value $a = H(1 + \text{ctg}\,\beta)$ and
$h = H_{cr}\,(1 - k_{бок})$, as well μ = 0,33 and $k_{бок}$ = 0,8, found

$$H_{cr} = \frac{b\,[0{,}1\,(1 + 3\,\text{tg}\,\psi)\,T/\sigma_{отр}] - 2{,}5}{4{,}2 + \text{ctg}\,\beta}, \tag{24}$$

where: β – angle of crumbling of nosepiece.
From this equation it can be seen that the critical depth of processing is directly proportional to the width of the nosepiece and the ratio $T/\sigma_{отр}$, characterizing the mechanical properties of the soil. For dry solid soils, the critical depth of cut is greater than for loose ones.
Fig. 17 shows the dependence of the critical depth of loosening on the angle of crumbling and the width of the nosepiece, obtained with the value of the angle of cleavage of the soil

$$\psi = \frac{\pi}{2} - (\beta - \varphi)\,,$$

where: φ – angle of soil friction against steel.

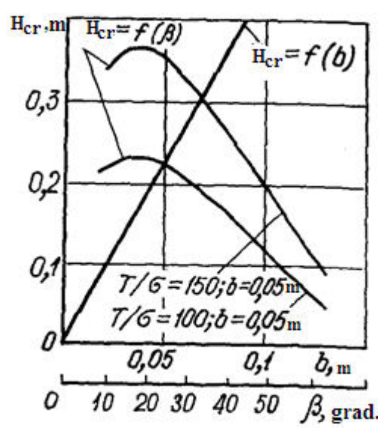


Fig. 17. Critical depth of loosening depending on the angle of crumbling and the width of the handle of the chisel working organ

The highest values of H_{cr} at β = 20 - 30^0 Obviously, this value of the cutting angle of the nosepiece will be optimal from the point of view of the complete loosening of the formation.

For a given value of the depth of processing using formula (24), can be determined the optimal width of the nosepiece - b. Taking a ratio $T/\sigma_{\text{отр}}$equal to an average of 125 for a processing depth of 20 and 40 sm, we obtain the optimal width of the doctor, corresponding to 30-40 and 40-60 mm. One of the main indicators of the quality of soil cultivation by chisel tools is the depth$H_1$of the continuous processing of the upper layer (see Fig. 16). This indicator determines the stable work for the operations of presowing soil preparation and sowing. The value of H_1 mainly depends on the total depth of loosening and the distance between the traces of the chisel working tools. The minimum depth of loosening should be no less than the depth of the presowing soil treatments (12 - 14 sm).

The calculated width of the in-between can be determined from equation

$$B = 2\,(kH - H_1)ctg\,\psi. \tag{25}$$

Fig. 18 shows the calculated and experimental values of the thickness of the H_1loosened layer as a function of the width of the intersection at different depths of treatment$H$. With a 30-40 sm inter-crossing, the thickness of the loosened top layer is in the range 16 - 20 sm, and with an increase in the cross-section over 40 sm it decreases to 5 - 10 sm.

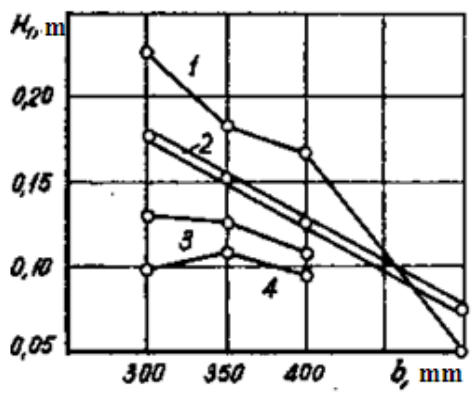


Fig.18. The thickness of the loosened layer of soil by the chisel working organ H_1, depending on the width of the interfacial and the depth of treatment: 1,2-H = 40 sm; 3 - H = 30 sm; 4- H = 25 sm; 1,3,4 - experimental, 2-computed (width of the nosepiece 70 mm)

The tractive resistance of the chisel working tools operating in semi-free cutting conditions is 83%, and in free-cutting conditions it is 46% of the tractive resistance of the chisel working member operating in conditions of blocked cutting. At the same time, the depressurizing force of the chisel working tools with free and semi-free cutting is reduced by approximately 1,5 - 2 times.

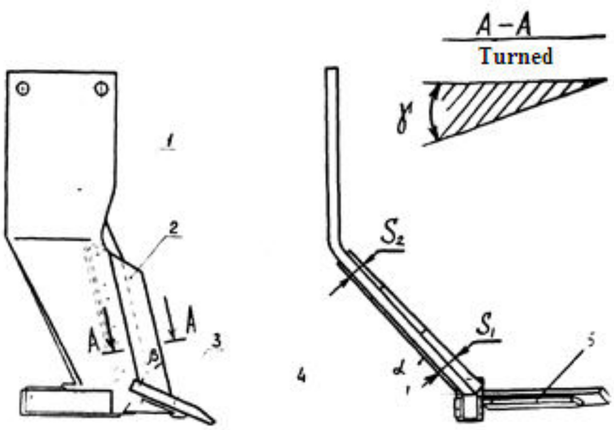
Based on the results of the research [20,21], the following values of the main parameters of the chisel working tools for soil tillage to a depth of up to 45 sm are recommended:

- struts of working elements - straight with an angle of inclination to the vertical 25 - 30^0);
- height of the rack 700 - 750 mm;
- width of the nosepiece 60 - 75 mm;
- the angle of installation of the nosepiece to the bottom of the furrow is 25 - 30^0;
- the width of the grip of lancet paws is 240 - 350 mm;
- for additional crumbling of the soil, use needle discs in the form of separate sections for each rack.

Chisel working tools with such parameters are the basis for the creation of the plow chie-selPCH-4,5.

Thus, with the soil protection system of agriculture, tools for deep loosening of the soil without turnover of the seam are used for wide-grasped lancet paws and chisel working organs with the above parameters.

However, in the dry years or with long-term application of minimum or zero soil treatments, as well as the movement of very massive agricultural aggregates along the field at high speeds, soil is compacted. This causes a decrease in the infiltration capacity and aeration of the soil and leads to a violation of the thermal and biological regimes, and, consequently, to a decrease in the yield of the cultivated crop. At the same time, the technical and operational performance of the units is also sharply worsening - the traction resistance and fuel consumption for processing one-hectare increase. Therefore, at the present time, chisel working bodies of the Paraplou type are widely used for deep loosening of compacted soils without recirculation of the formation, the principal scheme of which is shown in Fig. 19.



1-pillar; 2-cutting element; 3-chisel; 4-field board; 5-plane cutting knife

Fig. 19. Schematic diagram of a working organ of the Paraplou type

The Paraplou-type workpiece consists of a post 1, to which the chisel 3, a field board 4, a cutting element 2 and a flat-cutting knife 5 are attached. The stand has a complex shape - the upper part of it is vertical, and the lower part is bent to an angle $\alpha$to the horizon in a transversely vertical plane and to an angle$\beta$from the vertical in the direction of motion. Cutting elements with a cutting angle $\gamma$are installed on the inclined lower part of the rack, and it also provides for the possibility of installing a flat-cutting knife.
In order to optimize geometric parameters, the laboratory of mechanization of cultivation of agricultural crops of KazNIIMESKH [40] produced a set of such working elements, in which the parameters varied within the limits of: α= 24°-45° and γ= 24°-38°. In this case, the angle β= 75° was left constant, and the parameters of the plane-cutting knife-the capture width of 27,2sm, the height of the rise of the layer of 6,5 sm, the angle of installation of the blade of the knife to the direction of motion 50°, and the angle of inclination of the knife to the horizon in a plane perpendicular to its blade 25°.
The energy parameters of the experimental working bodies were determined in the soil channel by means of a spatial dynamometric frame at speeds of 6 - 10 km/h, a soil loosening depth of 25 sm with its humidity in this layer 45 - 47% of PPV and hardness 1,8 - 2,4 MPa.
The minimum tractive resistance is noted at the working element with the parameters α = 45°; γ= 24°. The index of deepening capacity (k=R_z/R_x), regardless of the parameters of the working element, was in the range 0,17 - 0,20. Wide trials of such a labor organ in various field conditions have shown that it satisfies the specific agrotechnological (initial) requirements for this operation and provides: a loosening depth of up to 35 sm with a standard deviation ± of 4,0 sm; continuous loosening of the upper soil layer to a depth of more than 20 sm; the height of the ridges not destroyed in the cross-track is less than 20% of the maximum depth of loosening; average margin of the surface of the field within 1,5 - 2,0 sm; preservation of crop residues on the field surface is at least 85%; the content in the treated soil layer of the fraction 0 - 5sm 80 - 90%.
A labour organ of the Paraplou type with such parameters was used in the development of the

ripper of the chisel РЧ-4,4 and the universal soil-cultivating tool УРО-4.

2.2.3. Working bodies of the PSh-3; PSh-5

Depending on the physical and mechanical properties of the soil and climatic conditions, each soil-climatic zone needs its own optimal technology for basic soil cultivation. For some zones, improvement of the soil structure and increase in the yield of agricultural crops in comparison with planar cuttings of the soil to a depth of 12-16sm or 22-27sm is noted when using a combined machine - a plane-splitter.

Its application in comparison with deep loosers and plane cutters promotes the creation of a looser layer along the entire depth of the layer being treated. In the zone of the Southern Urals, the density of the soil after processing it in autumn with a flat-blade or deep-loosener is usually 1,03 - 1,09 g/sm^3, and after a plane-cutter it decreases to 0,95 - 0,97 g/sm^3. This ensured better retention of moisture during melting of snow and increased its total reserves in a meter layer of soil for 500 - 630 t/ha, which gave an increase in the yield of cereals by 0,15 - 0,22 t/ha.

The working bodies of the plane-cutter are a flat-topped foot and a slit, from geometric parameters of which and their relative location depends not only the fulfillment of the agrotechnological initial requirements for planing the soil and splitting, but also the traction resistance of the machine. The rationale for the parameters of the planar paws is set out above.

In order to select the optimum values of the slit parameters, formulas (22) - (25) can be used, in principle, used to justify the parameters of the working part of the chisel plows. However, the work of a slit-cutter in combination with a flat-topped paw and the absence of a rigid need for continuous loosening of soil lying below its plane of blades impose a number of significant restrictions on the application of these dependencies. In this connection, CHIMESKH conducted studies to substantiate the parameters of the plane-cutter [22]. To study the influence of the structural parameters of the slit on its traction resistance, a calculation scheme was compiled (Fig. 20).

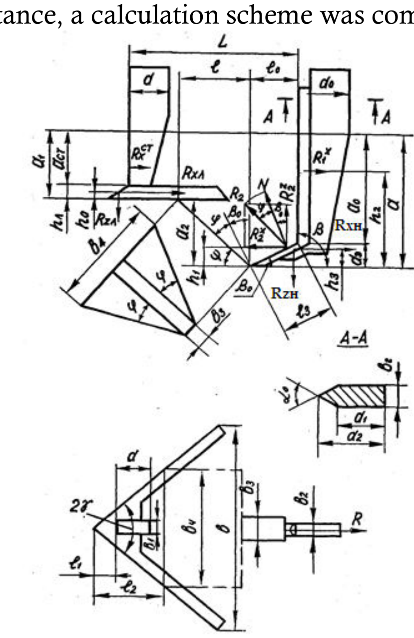


Fig. 20. Calculation scheme of a plane-cutter

Analysis of the forces acting on the plane-cutter made it possible to obtain the following dependences for determining the horizontal $R_{xн}$ and vertical $R_{zн}$ components of soil resistance to the slit-cutter:

$$R_{xн} = \frac{a_0 b_2}{\sin\beta}\left(\rho_1 + \rho_1 f\, ctg\,\frac{\alpha_0}{2} + \rho_2 f\cdot\frac{2d_0}{b_2} - \frac{\rho_2 f}{tg\,\frac{\alpha_0}{2}}\right) + +0{,}5\, q\, b_3 h^2 \frac{tg\,\beta_0 \sin(\beta_0+\varphi)}{\cos\varphi}; \quad (26)$$

$$R_{zн} = 0{,}5\ q\ b_3 h^2 \frac{tg\ \beta_0 \sin(\beta_0 + \varphi)}{\cos\varphi}; \tag{27}$$

where: ρ_1 and ρ_2 – specific soil pressure, respectively, in front and laterally on the rack of a slit-cutter;

f – coefficient of soil friction against steel;
φ – angle of friction;
q– coefficient of volumetric crushing of the soil;
h – the path traversed by the bit during compression;
α_0, b_2, β, d_0, b_3, β_0 – constructive parameters of the plane-cutter (see Fig. 20).

Calculations on dependences (26) and (27) and the results of experimental studies show the following.

With an increase in the thickness of the rack of the slitb_2, the traction resistance of the tool increases, the soil's crests and the width of the furrow furrow increase, which leads to a decrease in the degree of stubble conservation on the field surface.

It has been established that the angle of inclination of the slit to the bottom of the furrow β influences the traction resistance of the tool and the deformation of the soil in front of the working organ. It is recommended to take the inclination angle β equal to 90°.

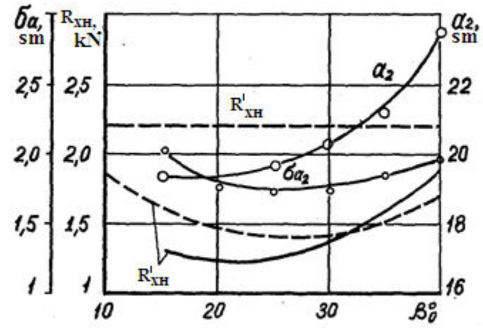


Fig. 21. Traction resistance of the slit cutter $R_{хн}$ and the depth of soil treatment (a_2, σ_{α_2}) depending on the angle of the groove of the groove with respect to the furrow bottom β_0

Analysis of Fig. 21 shows that by changing the angle of bit placement to the bottom of the furrow β_0 , it is possible to reduce the pull resistance of the slit-cutter $R_{хн}$ by 20 - 30% compared to the pull resistance of a slit-cutter $R'_{хн}$ without a chisel. The minimum value $R_{хн}$ and the best stability in depth (σ_{α_2}- the minimum) provides the angle of bit setting β_0= 23 - 27°.

As the bit width b3 increases, the forces $R_{хн}$ and $R_{zн}$ increase, which leads to a certain increase in the depth of the slit a_2 and its stability σ_{α_2} (Fig. 22).

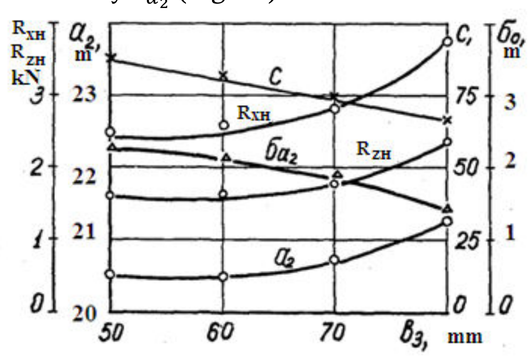


Fig. 22. Power and agrotechnological parameters of a buckler, depending on the width of the bit

However, the indicator C_1- the degree of stubble conservation on the surface of the field - is reduced. It is recommended to adopt the bit width $b_3 \approx$ 50-70 mm.

The minimum distance between the legs of the paw and the slit is determined from the condition that the zone of deformation of the soil in front of the slit does not have to reach the blade of the feet. At recommended CHIMESKH [22] constructive parameters of the sclerosis this distance should be not less than 0,57 m at a depth of loosening of the soil with a scraper of 15 sm and not less than 0,76 m at a depth of loosening of the soil with a 20 sm slit-cutter.

So, high-quality planar cutting of the soil simultaneously with its creasing at the minimum traction resistance is possible with the following values of the slit-cutter parameters: the angle of the racking of the splitter to the bottom of the furrow is 90 °; angle of sharpening of the leading edge of the rack of the slit 45-60°; thickness of the rack of the slit 20 - 25 mm; the width of the rack of the scraper at the base is 100 mm, and in the upper part 100 mm; angle of bit placement to the bottom of the furrow 23-27 °; the width of the bit is 50.. 70 mm; length of the chisel 150-200 mm; the thickness of the bit is 20 mm.

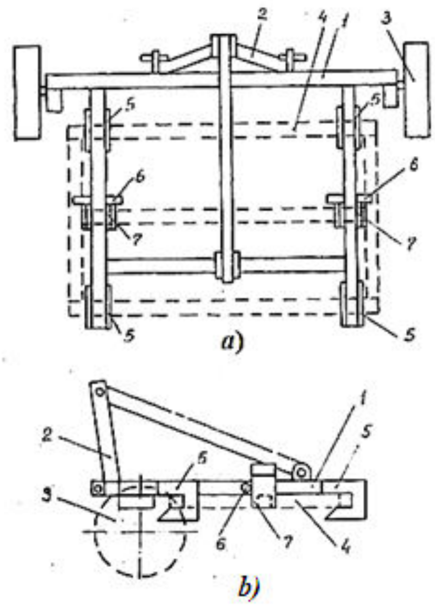
These parameters are the basis for the creation of flat-cutters PSh-3, PSh-5 and YPO-4.

It follows from the foregoing that under the soil protection system of agriculture for deep loosening of the soil without rotation of the bed, the flat-grooved plows PG-5, plow chisels PCh-4,5, subsoiler RCh-4,4 and flat cutters PSh-5 are used. These implements are energy-intensive and are only aggregated with class 5 tractors.

The use of tools with different types of working tools is due tothe fact that when performing specific technological operations in the cultivation of crops on different types of soils, their moisture and hardness, as well as the weediness of fields, are determined by weather conditions that change not only every year, but also during the agrotechnical period of operations. Depending on the condition of the soil and fields, tools are used with one or another type of working organ that provides the required quality of soil treatment in these specific conditions. It is not possible to create a single working body for the performance of all specific technological operations.

The disadvantage of each of these tools is the impossibility of installing on their frames other types of working organs for processing soils that have different physical and mechanical properties, humidity, density, and contamination. The construction of the frame of each of these tools provides for the installation of only one type of working tools for a specific technological operation for the cultivation of soils, which makes each of these tools not universal, which forces the farms to acquire them for high-quality execution of technological operations, depending on the state of the field. In this connection, attempts have been made to create a universal frame with the possibility of installing replaceable above-mentioned working bodies. However, placing them on a single frame of different types of replaceable working elements for tillage without a turnover of the bed causes the cumbersomeness and complexity of the construction of the frame, the high material consumption of the tool, and the process of replacing one type of working tools by another is very difficult and almost impossible to accomplish by one tractor driver because of the large mass of each of them (up to 100 kg).

In connection with the foregoing, it appeared expedient to develop a universal tillage tool (Fig. 23), which makes it possible to apply the appropriate working tools to perform various technological operations for processing soils, depending on their type and physico-mechanical state [10].



a) a species in excess*; b)* side view

Fig. 23. Universal tillage tool YPO-4

The tool consists of a frame 1, a hinge 2, support wheels with mechanisms for their adjustment in height 3, replacement frames 4, each of which has appropriate types of working tools for performing technological operations for processing soils.
Disengagement and attachment of exchangeable frames to the frame of the universal attachment takes place with the help of grippers 5, latches 6 mounted on this tool frame, and traps 7 installed on each of the exchangeable frames and allowing the gun frame to center relative to the longitudinal axis of symmetry of the replacement frames.
The track of the support wheels on the attachment - the coupling is constant and equal to 3600 mm. The support wheel mechanism provides a stepless adjustment of the required soil loosening depth from 15 to 45 sm.
Thus, the universal soil-cultivating machine YPO-4, containing a frame of a universal hitch device, a hitch and support wheels with mechanisms for their adjustment in height, in contrast to the single-shot tools PSh-5,PG-5, RCh-4 and RCh-4,5, is equipped with modular frames, each of which has different types of working elements for performing a certain technological operation for tillage, and the frame of the universal attachment - the coupling and the frame of the interchangeable modules are made with a removable connection with other. The introduction of plug-in tools into the design has made it possible to give the gun a versatility, and a detachable assembly consisting of guide brackets (grips), grippers and retainers provide the ability to replace modules with one tractor driver.
The advantage and competitiveness of the proposed universal tillage tool in comparison with several single-operation tools is provided mainly by reducing the material consumption from 1080 to 3715 kg (Table 3) and, consequently, the cost of their production.

Table 3 - Indicators of the material consumption of single-purpose and universal tool

Single-operation tools			Universal tool		
				Weight, kg	
Make	Working width, m	Weight, kg	Working width, m	of module	of suspended coupling device
PSh-5	4,40	1420	4,57	770	
PG-5	5,30	1830	5,10	800	600
PCh-4	4,66	2750	4,00	945	
PCh-4.5	4,50	1860	4,00	1000	
TOTAL		7830		3515	600

In addition, due to the qualitative and agrotechnical terms of performing deep loosening of the soil, the yield and quality of agricultural products will increase, i.e. an indirect effect will be obtained.

2.2.4 Disk and needle operative parts

Disc and needle operative parts have found wide application in machines intended for cultivation of agricultural crops in soil-protecting agriculture.

Solid spherical disks are used on seed drills (for example, LDS-6) when sowing grain crops simultaneously with the treatment of stubble backgrounds with high soil moisture, as well as on the stubble cleanersLDG-10, LDG-15 and LDG-20 and others when performing them operations of stubble peeling, spring surface treatment of stubble backgrounds, and presowing treatment of heavy soils with a mechanical composition. Performing the same operations on lightly mechanically treated soils is also carried out by lancers, where solid spherical discs are replaced by flat ones, since the latter considerably less spray the treated soil layer and leave more residue on the surface of the field. Before the flat-topped paws on the tools for the non-waste treatment of the perennial grasses (OPT-3-5), flat discs are also installed, which cut the sod in front of the paws, which allows to significantly reduce the furrows behind them. Cut-out spherical disks are used on heavy disc harrows such as BDT-7, which are used for discarding soil after tilled crops.

Needle disk working bodies are used on harrows (BIG-3A, BMSh-15, BMSh-20, etc.) used for stubble peeling, spring surface tillage, which is carried out to close the moisture, smooth the field surface and weed seeds to the soil with purpose of acceleration of germination and their destruction at preseeding processing, and also for care of crops of perennial grasses.

Investigations of the processes of interaction between disk working organs and the soil made it possible to identify the main parameters of these bodies that affect the qualitative and energy performance indicators. Methods and analytical relationships are proposed that allow to optimize the values of geometric parameters and operating modes of disk working bodies both from the point of view of ensuring the specified quality of the technological operation being carried out and in terms of ensuring minimum energy costs.

The main parameters of spherical disks (Fig. 24) are: D - is the diameter of the disk; r - is the radius of the sphere of the disk; α - angle of attack (angle of installation of the disk to the direction of movement of the unit); β - the angle of inclination of the plane of rotation of the blade to the vertical; 2θ - the central angle of the arc of the circle, formed as a result of the section of the disk by the equatorial plane; ε - the given angle, i.e., the angle between the back side of the cutting edge of the disc and the furrow wall; γ - cutting angle; i - angle of sharpening of the cutting edge of the disk.

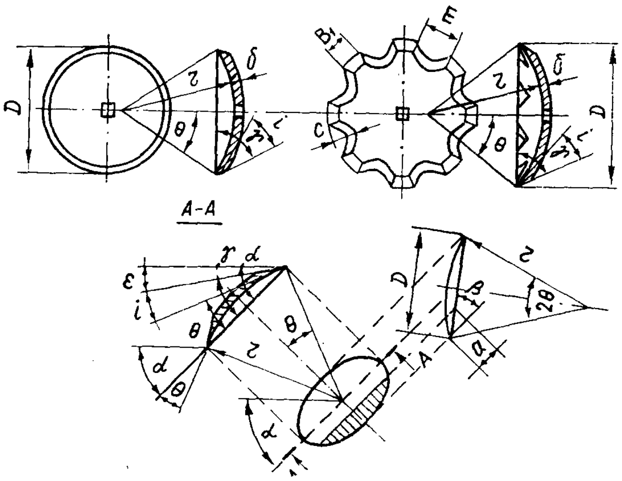


Fig. 24. Spherical disks and their geometric parameters

The nature of the deformation and displacement of the soil under the influence of spherical disks depends on the diameter and curvature of the discs, the angle of their installation in the horizontal and vertical planes and the dimensions of the seam cut out by each disc, as well as the speed of the machine and the soil properties. The choice of the values of the above parameters can not be arbitrary, since for each group of disk working bodies there are their own practical limits. In addition, individual geometric elements are interconnected by functional dependencies.
The basis for the theoretical analysis of the interaction of a spherical disk placed at an angle to the direction of motion and to the vertical of the disk is the basic relations of spherical trigonometry and equations characterizing the motion of any point on the surface of the sphere [11, 13, 14].
The motion of a point located on the surface of a spherical disk with $\alpha \neq 0$ and $\beta \neq 0$ characterizes the following parametric equations:

$$\begin{aligned} x &= \upsilon_n t + r \sin\theta \cos\alpha \cos\beta \cos\omega t + (r - r\cos\theta) \times \\ &\times \sin\alpha \cos\beta + r \sin\theta \sin\alpha \sin\beta \sin\omega t; \\ y &= (r - r\cos\theta) \cos\alpha \cos\beta + r \sin\theta \sin\beta \cos\alpha \sin\omega t - \\ &- r \sin\theta \sin\alpha \cos\omega t; \\ z &= (r - r\cos\theta) \sin\beta - r \sin\theta \cos\beta \sin\omega t, \end{aligned} \qquad (28)$$

where: υ_n – forward speed;
t – time of motion;
ω – angular velocity of the disk;
ωt – angle of disk rotation.

If in (28) we put $r \sin\theta = R_i$, and the arrow of the spherical segment (r- $r\cos\theta$) equate to zero, we obtain the equations defining the trajectory of the point of the blade of the spherical disk

$$\begin{aligned} x &= \upsilon_n t + R_i \cos\alpha \cos\beta \cos\omega t + R_i \sin\alpha \sin\beta \sin\omega t; \\ y &= R_i \sin\beta \cos\alpha \sin\omega t - R_i \sin\alpha \cos\omega t; \\ z &= -R_i \cos\beta \sin\omega t, \end{aligned} \qquad (29)$$

where: R_i – distance from the axis of rotation of the disk to the point under consideration, the maximum value $R_i = R$ – radius of a disk.
The absolute velocity of a point on the surface of a sphere is equal

$$\upsilon = \upsilon_n [1 + \lambda^2 \sin^2\theta + 2\lambda (\sin\alpha \sin\beta \cos\omega t - \cos\alpha \sin\omega t) \sin\theta]^{\frac{1}{2}}, \qquad (30)$$

where: λ – kinematic parameter and is equal to $\lambda = \upsilon_0/\upsilon_n$;

υ_0 – circumferential velocity of a sphere point.

Since the spherical disks of the ravines, the harrows and the seed drills are installed vertically, then taking in (28), (29) and (30) $\beta = 0$, we obtain equations that will characterize respectively the motion of any given point of the spherical surface, the points of the disk blade and the speed points on the surface of the sphere.

The diameter and radius of curvature of the disk are related

$$r = \frac{D}{2\,sin\theta}. \tag{31}$$

The diameter of the disk, depending on the operating conditions, should be chosen the smallest of the allowable values, since the increase in diameter sharply increases the load necessary for its penetration.

The diameter of the disc depends to a certain extent on the given depth of soil cultivation α, since D and α are interconnected by a proven practice dependence [14]

$$D = k{\cdot}\alpha, \tag{32}$$

where: k – coefficient of proportionality.

For harrows, stubble cleaners and seeders, when selecting k, it is necessary to take into account their adaptability to the unevenness of the surface of the field, the presence of weeds and stubble on it, since the possibility of pressing the soil between the discs depends on these factors. The heavier the working conditions, the greater the diameter of the disc. For lakes, the value of k should be taken equal to 5 - 6, and for disc harrows 4 - 6.

After choosing the diameter of the disk, the required radius of the sphere is found from formulas (32) and (31). In order to determine the angle θ geometric elements are used in the sections of the disk by horizontal planes at different heights from the bottom of the furrow (Fig. 25). For any section of a spherical disk by a horizontal plane, the geometric elements are related by

$$\alpha = \gamma_a \pm \varepsilon_a = \gamma \pm \varepsilon = const, \tag{33}$$

where: $\gamma = \theta + i$; γ and ε – geometric elements of a disk in the equatorial plane;

γ_a и ε_a – geometric elements of the disk in a secant horizontal plane located at a distance from the bottom of the furrow, equal to the depth of processing – a.

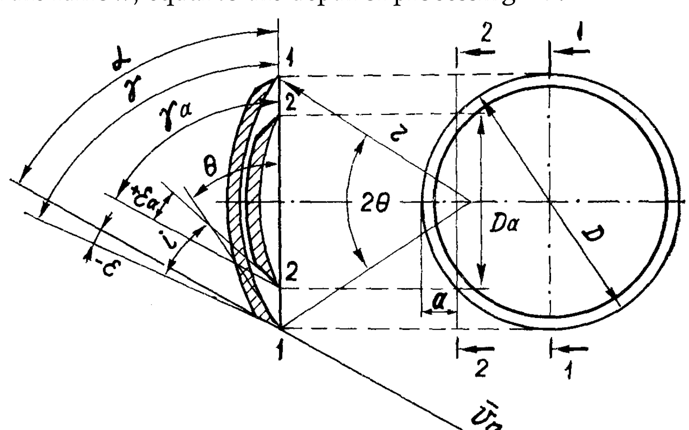


Fig. 25. Geometrical elements in sections of a spherical disk horizontal planes at different heights from the bottom of the furrow

Angles γ и γ_a are related to each other by the relation

$$tg\,\gamma = \frac{D}{D_a}\,tg\,\gamma_a\,, \tag{34}$$

where: D_a – the chord of the immersion of the disk to depth a and can be found by relation

$$D_a = 2\sqrt{a\,(D-a)}. \tag{35}$$

Analysis of dependencies (33) and (34) shows that for a given disk installation determined by an angle α, the values γ_a, and, consequently, also ε_a vary in the height of the sections. With increasing height a_a the value γ_a increases, and ε_a - decreases. Therefore to ensure normal operation of the disc at depth a angle ε_a must be positive ($\varepsilon_a > 0$). For $\varepsilon_a < 0$, the disc rests with the back side on the bottom of the furrow and will tend to roll out onto the surface of the field. Considering the crushing of the soil by the back side of the disk, the minimum allowable value of the occipital angle in the plane of the disk section by the horizontal at a distance a from the bottom of the furrow should lie within the range ε_a = 3-5°. The angle of sharpening of the cutting edge of the discs of the harrows and scallops is usually taken to be equal to i = 10-20°.
From the given values ε and α with the help of formula (33) we determine γ_a, taking into account which, by (34) we find γ. At $\theta = \gamma - i$ (see Fig. 25) and the received sharpening angle of the disc (i) determine the angle θ. Knowing the angle $\theta$and diameter D from (31), find the radius of curvature of the disk sphere. The thickness of spherical disks is determined by empirical dependence δ = 0,008D (mm). For disks on heavy duty soils δ = 0,008D +1 [11].
The cut-out parameters of the cut-out spherical disks (see Fig. 24) along the edge of the blade are equal: E=1,5· $B_1\eta$cutting height c = $D/8$. The number of cuts is 8, the shape of the notches is round [13].
When the discs move in the soil at a depth a bottom of the furrow is uneven, since each disc forms a furrow grooved profile. The theoretical height of ridges is determined by the formula [11]

$$C = \frac{D}{2} - \sqrt{\frac{D^2}{4} - \frac{b^2}{4\,sin^2\,\alpha}}\,, \tag{36}$$

where: b - is the distance between the disks (B) in the projection to the frontal plane.
This formula is used for the depth of processing

$$a > \sqrt{\frac{D^2}{4} - \frac{b^2}{4\,sin^2\,\alpha}}\,, \tag{37}$$

but at $a < \sqrt{\frac{D^2}{4} - \frac{b^2}{4\,sin^2\,\alpha}}$ ridge height $C = a$.

When the condition (37) is satisfied, the area of the cross-section of the soil formation by the disk is given by formula

$$S = b(a-c) + \frac{D^2}{4} sin\,\alpha\ \times \left\{arcsin\left[\frac{b}{D^2 sin^2\alpha} \times \left(D\,sin\,\alpha - \sqrt{D^2 sin^2\alpha - b^2}\right)\right]\right\}. \tag{38}$$

When the stubble cleaner or harrow is working in conditions a = c

$$S_1 = \frac{D^2}{4} arc\,cos\frac{D-2a}{D} - \frac{b}{4}(D-2a). \tag{39}$$

Elemental soil resistances, arising on the working surface and blade of a vertically mounted spherical disk, do not have a single resultant force [14,15]. They can be brought to the dynamo, as well as to two crossing forces $R'$and R'' (Fig. 26).

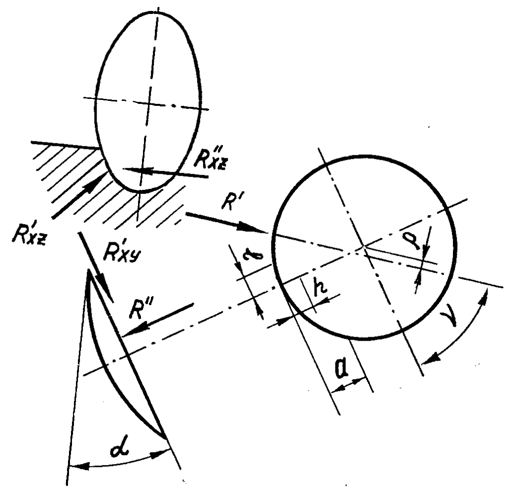


Fig. 26 - Forces applied to a spherical disk

Force R' lies in a vertical plane and passes at a distance of ρ from the axis of rotation of the disk. Because of the small size of the arm ρ, which is the radius of the bearing friction circle, we can assume that the force R'passes through the axis of rotation of the disks. The force $R''$is parallel to the axis of rotation of the disks. It is located at a distance h from the bottom of the furrow, equal to approximately half the depth of the disks a, and at a distance ℓ ahead of the vertical plane drawn through the axis of rotation of the disks. The segment ℓ is small and can be equated to zero.
Since the starting value for the force calculation of a machine is usually its tractive effort, which is determined by linear dynamometry or reference data, it is important to establish methods for determining the values of the impedance components of the disk R_y and $R_z$by a known value R_x. If in designing the power characteristic a of the disk is specified by forces R'andR'', then to find the components R_x, R_y and R_z forces of general resistance R_{xyz}it is recommended to use the dependencies [15]

$$R_x = R'' \sin\alpha + R' \cos\nu \cos\alpha$$
$$R_y = R'' \cos\alpha - R' \cos\nu \sin\alpha, \tag{40}$$
$$R_z = R' \sin v,$$

where:v – angle formed by force $R'$with a horizontal plane. The magnitude of this angle is determined from the equation of moments of forces $G$and R' in the vertical plane perpendicular to the axis of rotation of the disks (Figure 27) using the formula:

$$v = arc\sin\frac{G\,l_1}{R^1 l_2}. \tag{41}$$

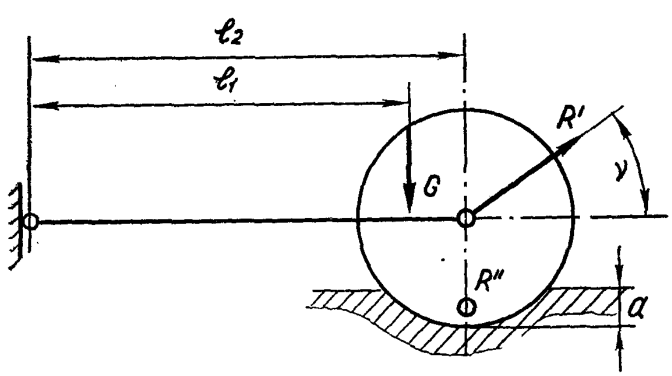


Fig. 27. Determination of angle v, formed by force $R'$with a horizontal plane

If the given forces are R_x,$R_y$and R_z which can be determined experimentally by spatial dynamometry, and the desired $R', R''$and angle ν, then according to Fig. 28 for their definition we find

$$R' = \sqrt{R_z^2 + \left(R'_{xy}\right)^2},$$

$$R'' = \sqrt{R_x^2 + R_y^2 - \left(R'_{xy}\right)^2}, \tag{42}$$

$$\nu = arctg\frac{R_z}{R'_{xy}}$$

where:R'_{xy}=$R_x cos\alpha - R_y sin\alpha$.

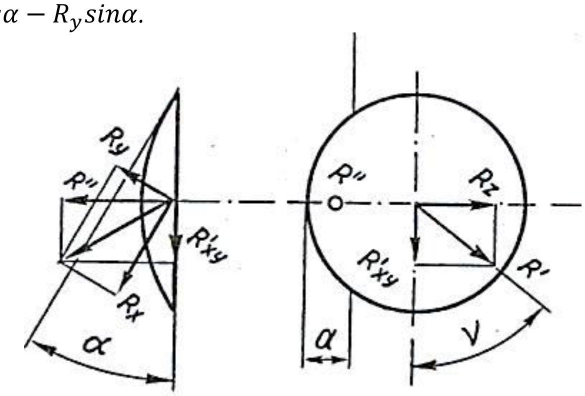


Fig. 28. Scheme for determining forces R' and $R''$and angleν

To determine the components R_yи$R_z$harrows and stubble cleaners according to the known traction resistanceR_x , which requires no complicated instruments and equipment, it is recommended [15] to use the coefficients $m = R_z/R_x$ and $n = R_y/R_x$, the averaged values of which are given in Table 4.

The technological process of deformation of the soil by a needle disk working with a given angle of attack proceeds in the following sequence. Each disc needle that operates in a closed or open on one side groove, penetrating into the soil, creates in it a gradually increasing compressive stress. When the compression stress of the value corresponding to the ultimate strength of the soil is reached, a sliding body of curvilinear form is extracted from the treated formation leaving a deformation shell on the bottom of the furrow. With further advance of the needle in the soil, the sliding body is crushed, and its individual parts are shifted away from the line of motion of the center of the disk. Then the needle goes out of the soil, describes an arc in the air, and then the process repeats itself many times.

The dimensions and location of the shells determine the relief of the bottom of the furrow (Fig. 29), and, onsequently, the uniformity of the depth of the treatment; The fractional composition of the treated layer depends on the fragmentation of the sliding body; the displacement of the soil elements determines the quality of the leveling of the surface profile of the field, the percentage of stubcble conservation and the quality of seeding of weeds in the soil.

Table 4 - Average values of the coefficients of the disc harrow spines andstubble cleaners

Diameter of the disc, mm	Angle of installation α, grad.	Depth of processing, a, sm	$m=\frac{R_z}{R_x}$	$n=\frac{R_y}{R_x}$	Diameter of the disc, mm	Angle of installation α, grad.	Depth of processing, a, sm	$m=\frac{R_z}{R_x}$	$n=\frac{R_y}{R_x}$
Harrows					Stubble cleaners				
450	10	6	1,38	0,40	450	25	6	0,72	1,24
		9	1,28	0,16			9	0,58	1,13
		12	1,50	0,00			12	0,56	1,24
	15	6	1,06	0,92		30	6	0,53	1,15
		9	1,03	0,73			9	0,44	1,07
		12	1,10	0,56			12	0,50	0,94
	20	6	0,92	1,20		35	6	0,44	0,98
		9	0,78	1,03			9	0,40	0,97
		12	0,80	0,88			12	0,37	0,84
510	10	6	2,00	0,54	510	25	6	0,76	1,11
		10	1,57	0,21			10	0,61	1,00
		14	1,56	0,12			14	0,56	0,96
	15	6	1,55	0,79		30	6	0,58	1,09
		10	1,20	0,52			10	0,50	1,03
		14	1,17	0,38			14	0,45	0,91
	20	6	1,10	0,97		35	6	0,44	0,98
		10	0,83	0,79			10	0,40	0,91
		14	0,76	0,69			14	0,37	0,76

When the needle discs are operating under open furrow conditions (under these conditions, the last row of disks is used when they are arranged in multiple rows on the frame of the machine), to ensure the admission of agricultural demands for uneven processing depths, the needles must be buried in the soil in areas already loosened by the foremost working organs. Therefore, they cannot create limiting stresses in the same volume of the reservoir as the foremost discs. Their task is reduced to a local deformation, consisting in cutting ridges that remain at the bottom of the furrow after the front row of discs, additional crumbling and displacement of the crushed soil particles (see Fig. 29).

The nature of the deformation of the soil with a needle disc changes with the growth of soil density and the angle of attack of the batteries. In the overwhelming majority of working conditions, it deforms the soil, chipping its elements. With this in mind, the following dependencies are proposed [16] for determining the dimensions of individual wells

$$L_p = 2\cdot a\cdot e^{\left(45+\frac{\varphi}{2}\right)tg\,\varphi}\cdot sin\left(45+\frac{\varphi}{2}\right)-(a-\Delta)tg\left(45+\frac{\varphi}{2}\right), \quad (43)$$

$$l_p = sin\sqrt{2r\Delta-\Delta^2}; \quad (44)$$

$$b_p = (d+2\Delta)K_\phi, \quad (45)$$

where: L_p – estimated length of the shell in the cleavage area of the soil; a – depth of processing; φ– angle of internal friction of soil; Δ – admission of agricultural requirements for irregular depth of processing; α– battery angle of attack; r – radius of disk; d – needle width; l_p– estimated length of the penetration zone of the needle; b_p– estimated width of the loosening zone; $K_\text{ф}$ – coefficient that takes into account the effect of the curvature of the needle on the width of the deformation zone.

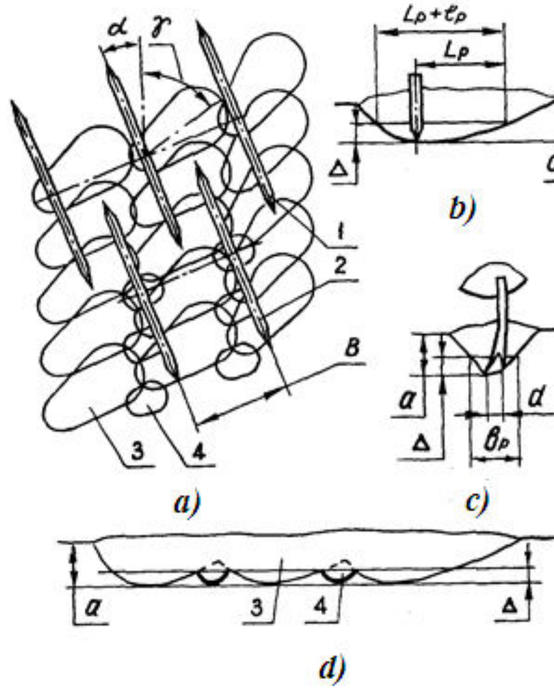


Fig. 29. Scheme of soil deformation by needle batteries: *a)* scheme of deformation in plan; *b)* the limiting section of the deformation shell formed by the needle of the front row of batteries; *c)* the cross section of the deformation shell; *d)* the cross-section of the treated formation (1, 2 is the front and rear row of batteries, respectively; 3, 4 is the deformation shell created by the needle of the front and rear row of batteries).

For the needle disk, the harrows BIG-3A, BMSH-15, BMSh-20 with passive operation (the curvature of the needle is set forward in the course of rotation of the disk) $K_\text{ф}$is proposed [16] to be found from the relation

$$K_\text{ф} = -0{,}0005a^2 + 0{,}0394a + 0{,}3589, \qquad (46)$$

and, with active work

$$K_\text{ф} = -0{,}00006a^2 + 0{,}0084a + 0{,}0855. \qquad (47)$$

The location of deformation zones, determined by the angle between the direction of motion of the tool and the axis of deformation γ (see Fig. 29), has a great influence on the qualitative characteristics of the needle disk. The results of the research showed that the value of the angle γ is affected by the sliding of the needle discs, the soil moisture, the proximity of the open sulcus, which depends on the interdisk distance.

Analytically, the magnitude of the angle γ can be determined from expression

$$\gamma = arc\ tg\ (sin\alpha\ cos\alpha). \qquad (48)$$

Taking into account the formulas (43), (44) and (48) for determining the parameters of deformation shells formed by the disc needles, and for a given interdisk distance (B), the angle of attack for single-row batteries is chosen from the relation

$$B \cdot cos\alpha = L_p sin\,\gamma + sin\,\alpha\,\sqrt{2r\Delta - \Delta^2}, \tag{49}$$

but, for multi-row

$$B \cdot cos\alpha = (p-1)L_p sin\,\gamma + (p+1) sin\,\alpha\,\sqrt{2r\Delta - \Delta^2}, \tag{50}$$

where:p - is the number of consecutively arranged rows of batteries on the machine.

The width of each needle should be taken within 20 - 30 mm, because with a smaller width the needle will simply crush the soil into the furrow walls without chipping it, and with a larger width, the metal capacity of the disc grows faster than the width of the deformation zone, which is economically impractical. Increase the width of the loosening zone with a needle can be achieved by using the effect of parallel profiles, in which neighboring disks form a common zone of deformation of each individual needle. With a circumferential step of about 90 mm, the disk deforms the soil almost as a continuous one. The critical step, when the effect of parallel profiles ceases, is a step of 180 mm. Directly to determine the number of needles on disk$z$follows from the dependence

$$z = \frac{2\pi r\lambda}{t\cos\alpha} \tag{51}$$

where: λ - coefficient of sliding of the needle disk; t- is the circumferential spacing of the needle.

Optimum value λ for needle discs of harrows BIG-3A, BMSH-15 and BMSh-20 is recommended to be equal to 1,02.

According to the experimental data [16], the optimal value of the circumferential pitch is 150 - 160 mm, the disc diameter is 550 mm, the interdisk distance is 200 mm, the angle of attack of the batteries is 17 - 22°. Taking into account these studies, the geometrical parameters of the needle disk (Fig. 30) are chosen, which is currently installed on the harrows BIG-3A, BMSh-15 and BMSh-20.

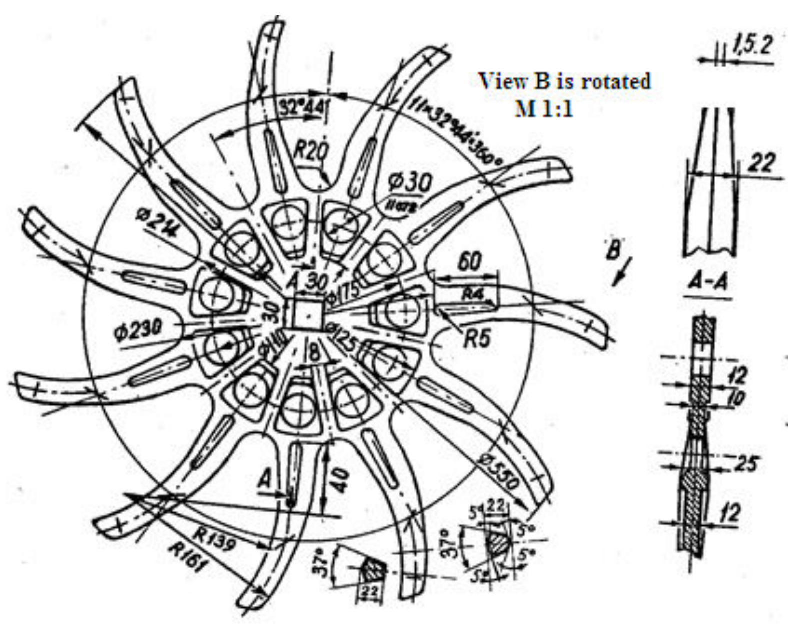


Fig. 30. Working organ of harrowsBIG-3A, BMSh-15,BMSh-20

At present, research works are continuing to refine the shapes and geometric parameters of the disk needle working organs. It is believed that the rational shape of the needle at a disk radius of 275 mm for treating the soil at an average depth of 8 cm is a hyperbola. At the same time, not only qualitative performance indicators are improved, but the tool will have better soil penetration and greater stability of the working parts in depth.

2.2.5. Rod operative tool

The rod operative tool is a horizontal shaft (square, hexagonal or circular cross-section) installed perpendicular to the direction of movement of the unit. The shaft is supported by several bearings located in the shoes of the mills (racks). With the help of a propshaft or chain transmission, the rod is rotated from the tool wheel in the direction opposite to the direction of rotation of the drive wheel. The length of the path of the machine for one turn of the rod is 0,7 - 1,3 m.
The rod body is used on machines designed for preseeding till the soil to a depth of 5 - 10 sm and care for couples. In the fields with smooth terrain and loose soil, the quality of the rod cultivator is better than the paw cultivator. It eliminates weeds without passes, leaves a flat bottom of the furrow and the surface of the field, does not mix wet and dry soil layers and is almost unaffected by weeds even in heavily clogged fields. For these reasons, the rod member can be used to create special anti-erosion machines for tillage, i.e., it will insignificantly fill up the stubble residues in the soil and spray the treated soil layer.
When the rod hits the root, the weed is first bent and then pulled out, after which the bent weed begins to move along with the soil. The bar is gradually released from the weed during rotation. With a shallow arrangement of roots and a greater depth of the rod, there are cases when the weeds will be almost undamaged. If the bar goes finely, and the weed has a long root, then only the "smoothing" of the weed occurs - the rod damages its stem and leaves but does not cause significant harm to the root.
The most suitable for work is the square and hexagonal section of the rod [17]. Ribs on the surface of a square or hexagonal bar contribute to a better grip of the weed rod and pull them out. But because of the variable angle of inclination of the lower face of the rotating bar of square or hexagonal sections, the compaction of the bottom of the furrow proves to be unstable, cyclically changing during the turn of the rod by 90° or 60°. The rod of the same round section more stably affects the soil and more evenly compacts the bottom of the furrow. This is especially important in cases where the work of the machine with the bar immediately precedes the sowing.
Bucket cultivators poorly copy the relief of the field because of the large width of the rod and practically do not recline on compacted soils. The experience of their operation indicates the advisability of combining in one unit the paw and rod-type working organs.
To smooth the surface of the field and destroy the weeds after the passage of the lancets of the heavy cultivator KPE-3.8A type, a removable device PSHP-3.8 was designed for it, the rod of which had a round profile of the cross section. Its drive is carried out from two needle discs from the harrow BIG-3A. However, this device, satisfactorily performing agrotechnical functions, is metal consuming, energy-intensive and makes it difficult to transfer the unit to the position of long-distance transportation.
Investigating the process of interaction of the rod of the circular section with the soil, came to the conclusion that the drive to it is not necessary and it itself will rotate in the desired direction [18]. It is justified by the following. The circumferential velocity of the points of the rod surface is approximately one order of magnitude less than the speed of its translational movement. Therefore, the instantaneous center of rotation of the rod (M) is located above its center (0) on the extension of the diameter perpendicular to the direction of motion, and the absolute velocities of the points of the rod surface($\bar{v}_a$) will be directed along the normal to the radius vector (MA), connecting the instantaneous center of rotation of the rod and the point under consideration (Fig. 31). Then the angle α between the normal to the surface at point A and the vector ($\bar{v}_a$) will be greater than the angle φ, since the angle γ, determining the position of the radius vector MA, is greater than zero. This means that the resultant of the normal pressure force and the frictional force $\Delta R_{\text{ш}}$ applied to the soil particle at point $A,$ is deviated from the normal to the surface at this point by a maximum angle equal to the friction angle φ and coincides in direction with the direction of rod movement.

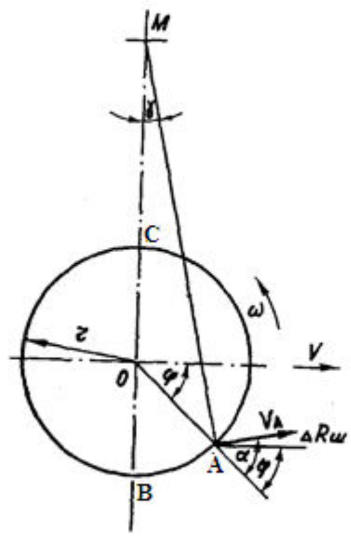


Fig. 31. The interaction of a rod with a circular cross-section with soil

On the surface of the AB rod, the angle between the direction $\Delta R_{ш}$ and the horizontal radius at each point varies from 0° to (90° -φ). Therefore, the soil particles adjacent to the rod in this section will move in the direction $\Delta\bar{R}_{ш}$ and be crushed by the bar. On the section of the surface of the rod AC, the angle between the vectors ($\bar{v}_a$) and$\Delta\bar{R}_{ш}$ also varies from 0° to (90° - φ), and the soil particles adjacent to the rod in this section will be lifted by it and pass under the bar. The layer at the height of the point A , is usually called the separating layer. Its position is determined by the angle φ.

It was established [18] that when the rod moves, the position of the soil separation layer at the height of point A (see Fig. 31) is determined by the internal angle of friction of the soil, both when the surface of the rod is stuck with soil, and without it. Therefore, the height of this layer is found from formula

$$h = r(1 - \sin\varphi). \tag{52}$$

This phenomenon is explained by the fact that during movement the rod acts on the soil particles lying in front of it not by its surface, but by the soil layers adjacent to the rod, since the movement of the particle begins much earlier than the bar approaches it. In this case, in the initial period of the motion of the particle, the distance to which it moves is considerably less than the path of the rod during the same period of time. This suggests that at first compression takes place, the compaction of soil layers adjacent to the rod, and then the compacted layers act on other, lying ahead of them soil particles. Thus, a plane of sliding of the particles relative to each other is formed, and the densified core itself takes the form of a wedge, the vertex of which is in the separating layer. In the future, the rod acts on the forward lying soil particles by means of this soil wedge. The filming of this process [18] showed that the soil wedge under the action of the forward lying soil rotates and in its rotation outruns the rotation of the rod, and the soil particles that make up the soil wedge move relative to the surface of the rod with slip.

Therefore, it was assumed that the soil wedge, rotating under the action of a counterflow of soil, can rotate the rod at the expense of the clutch forces without supplying the torque from outside.

In both sections of the BAandAC rods the soil particles move with slip and the resulting tangential friction forces the rotation of the rod and will be proportional to the forces of normal pressure. Therefore, knowing the law of normal pressure on the surface of the rod, depending on the angle of its rotation, it is possible to determine the amount of torque that arises on the rod from frictional forces.

The relative diagram of the dependence of the specific normal forces on the rod surface on the angle of its rotation is shown in Fig. 32.

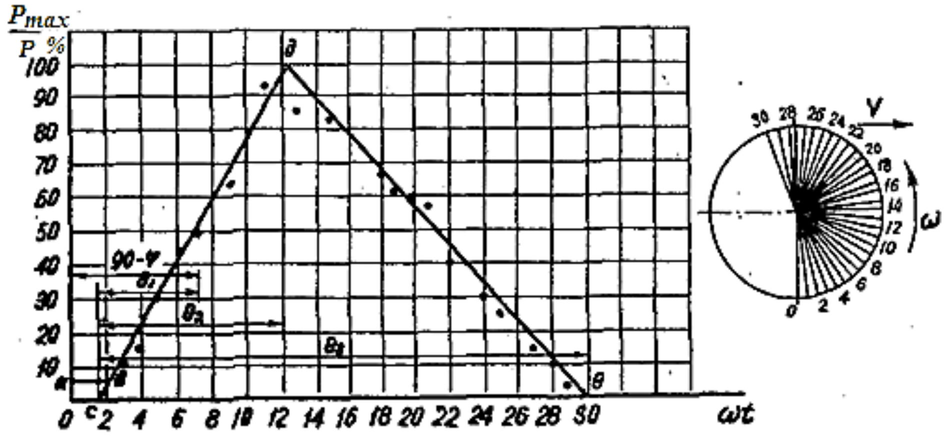


Fig. 32.Relative diagram of specific normal forces on the surface of the rod depending on the angle of its rotation

An analysis of this diagram shows that the maximum pressure practically coincides with the direction of rod movement and has a linear dependence on the angle of rotation.
Taking into account the numerical values of the central angles characterizing the sections of the rod circumference below and above the distribution layer and in general the whole section, determined from this diagram, the following dependence is proposed for finding the magnitude of the torque on the rod [18]

$$M_{\text{кр}} = \frac{r^2 l_{\text{ш}} P f}{2{,}28}[4{,}06 - (1{,}39 - \varphi)^2 - 2(1{,}39 - \varphi)^3] \qquad (53)$$

,

where: r- is the radius of the rod; $l_{\text{ш}}$- the length of the rod; P- maximum hardness of the soil in the horizontal direction; f- coefficient of adhesion.
The idea of the non-drive rod is realized in the form of adaptation to heavy cultivators KTS-10-1 and KTS-10-2 and machines for preseeding soil cultivation OP-8 and OP-12. The test results showed that a non-drive rod with a diameter of approximately 30 mm rotates in the layer of soil loosened by lancet paws and satisfactorily copes with its functions.

2.3. Placement of labor organs on framesof tillage machines and tools

From the parameters of the location of working organs on the frames of machines and implements of soil-protecting agriculture, it depends not only on the fulfillment of the agrotechnological (initial) requirements to the quality of soil cultivation (the profile of the bottom of the furrow and the surface of the treated field, weed cutting, stubble conservation on the field surface, etc.) but also their patency (working capacity). The latter is explained by the fact that they work on stubble backgrounds containing a significant amount of crop residues on the surface of the field. In these conditions, the passability of machines in which the working bodies are flat-moped lancet paws is provided only if the distances from the plane of the paw blades to the lower point of the gun frame, between the working elements in the row and between the rows of paws are sufficient for passage of stubble residues and deformed soil. Otherwise, either the soil is jammed between the working organs, or a "grabbing" effect from the paws appears.
The distance between the working elements in the series depends on the number of rows, the width of the grip and the zone of deformation of the soil under the influence of the paw, the dimensions of which depend on its geometric parameters, the physical and mechanical properties of the soil and the depth of loosening. Therefore, knowing the width of the grip of the paw, which has the optimal values of geometric parameters, it is possible to find the minimum permissible distance between the paws in the row, and, consequently, the minimum allowable number of rows on the frames of flat cutting machines.

The distance between rows of flat-topped paws with their multi-row arrangement on the frame of the machine is predetermined by the size of the overlap, the zone of deformation of the soil by the paw and the range of flight of the soil bed after its descent from the working surface of the knife. It should be noted that in determining the range and height, the velocity of the formation was assumed to be equal to the translational velocity of the working body. However, such a case is possible only if an ideal, i.e., without friction and deformation, occurs, the advance of the soil layer along the knife of the working organ. In addition, the cutting angle in these studies was assumed to be equal to the angle of setting the share to the bottom of the furrow. It has been proved above that the translational velocity of the working element will be much greater than the relative speed of movement of the soil along the working surface of the knife, i.e., the rate at which the soil layer leaves it, and the cutting angle of the paw is variable, and its magnitude depends on many factors (see formula 16)). In addition, the area of propagation of soil deformation by a flat-topped paw is determined not by the angle of setting of the share to the bottom of the furrow, but by its actual cutting angle. In view of the foregoing, the following formula is proposed for determining the minimum permissible distance between the paws in a row [8]:

$$D \geq 2b^{*} + 2atg\frac{\theta_{\alpha'}}{2}/\cos(\alpha' + \varphi), \tag{54}$$

where: b^{*} - half of the width of the claw; a- soil loosening depth;
$\theta_{\alpha'}$- angle of soil displacement by the paw determined by formula (6); $\alpha'$is the actual cutting angle of the soil by the paw defined by formula (16).
To increase the distance between the paws in the row and ensure the overlap between them, a multi-row arrangement is made on the frame of the machine. The distance between rows (L_0) is chosen from the following considerations. To ensure complete cropping of the weeds, especially in the case of deviation of the machine from rectilinear movement, the feet are located with some overlap $A_1K = \Delta b$ (Fig. 33). Therefore, one leg is displaced in the direction of movement of the implement relative to the other by a distance $B_1B_2 = A_1A_2$. The wing of the flat-topped paw has a definite area of propagation of the soil deformation, the value of which is found from the relation

$$d = \alpha \cdot tg\frac{\theta_{\alpha'}}{2}/\cos(\alpha' + \varphi). \tag{55}$$

To exclude the possibility of overlapping deformation zones from adjacent paws, which leads to an increase in soil sputtering, a decrease in stubble conservation on the surface of the field, and to clogging machines with crop residues, they are shifted in the direction of travel so that $KM \geq 2d$,that is, the paw is additionally displaced by a distance$AA_2 = BB_2$.

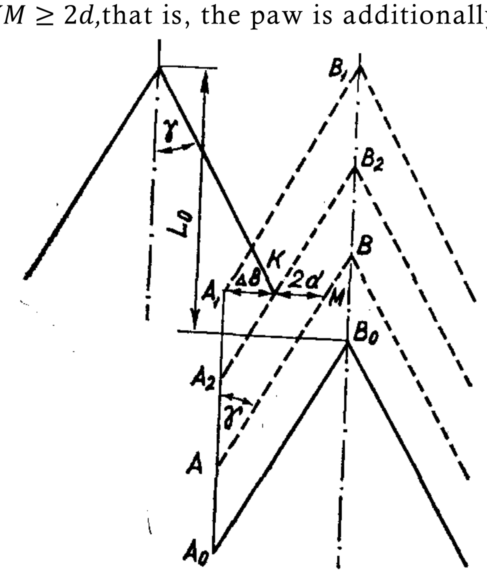


Fig. 33. To determine the minimum allowable distance between rows of paws

The soil layer descends from the surface of the paw knife at a certain angle to the horizon and at a certain speed, that is, as the body is cast at an angle to the horizon. At the same time, it is necessary that, after flying a certain distance, the formation fell before the beginning of the zone of deformation of the soil by the working organ of the subsequent series. If this condition is not observed, the soils of the soil are "lapped" with all the ensuing negative consequences: increased spraying of the treated soil layer, stamping of stubble residues in the soil, etc. Therefore, the working member must be offset by a distance $AA_0 = BB_0$.

Consequently, the total displacement by the flat-paw of the rear row relative to the front row will be

$$L_0 = A_1A_2 + A_2A + AA_0. \tag{56}$$

From Fig. 33, taking into account formula (55), it follows that

$$A_1A_2 + A_2A = AA_1 = \operatorname{ctg}\gamma\left[\Delta b + \frac{2a \cdot \operatorname{tg}\left(\frac{\theta_{\alpha'}}{2}\right)}{\cos(\alpha' + \varphi)}\right]. \tag{57}$$

Distance AA_0 is determined by the range of flight of the soil, coming at an angle to the horizon with some initial velocity, and is equal to

$$AA_0 = \frac{(\upsilon_r^*)^2 sin2\alpha'}{g}, \tag{58}$$

where: g - is the acceleration due to gravity.

In the final analysis, the quantity L_0 is determined from relation

$$L_o = \operatorname{ctg}\gamma\left[\Delta b + \frac{2\alpha\operatorname{tg}\theta_{\alpha'}/2}{\cos(\alpha' + \varphi)}\right] + \frac{(\upsilon_n^*)^2 \cdot \sin 2\alpha' \cos^2\gamma}{g} \mathrm{x}$$

$$\mathrm{x}\left[\frac{\operatorname{tg}^2\gamma}{(\cos\alpha_o + tg\rho_{a_o} \cdot \sin\alpha_o)^2} + \left(\frac{f}{f^1 + f}\right)^2\right], \tag{59}$$

where: ρ_{a_o} - is defined by formula (13).

The distance between rows of paws for cultivators-flat-croppers, heavy cultivators, etc. found by formula
(59) should lie within 400 - 600 mm.

To determine the minimum permissible overlap (Δb), consider the following. Let the cultivator-planar having a two-row arrangement of working elements with a distance between their rows - $L_o =$, under the influence of some external disturbance deviate from the direction of motion in the horizontal plane so that the overlap Δb =0 (Fig. 34),i.e. the point of the foot C of the front row and the point of the foot $A$of the rear row coincided with the direction of the speed of movement, i.e. $\boldsymbol{C'A'}$ υ_n^*. Then from the triangle$\boldsymbol{A'C'B}$.

$$\Delta b = L_0 tg\Delta\delta, \tag{60}$$

where: $\Delta\delta$ - is the angle of deviation of the machine from the direction of motion in the horizontal plane at which Δb = 0.

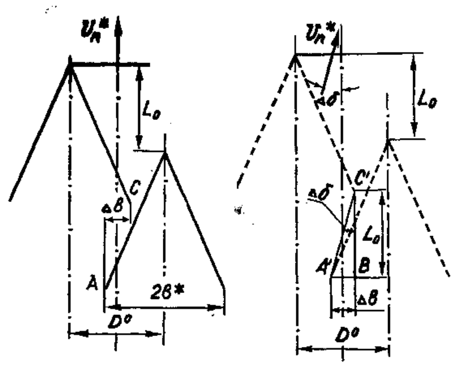


Fig. 34. To the definition of theminimum permissible overlap of adjacent planar paws

Consequently, the overlapping of adjacent paws depends on the distance between the rows and the angle $\Delta\delta$. It is known that the maximum angle of deviation of the movement of tillage aggregates reaches 8 - 10°. Then at L_0 = 400 - 600 mm, the minimum allowable overlap should be 50 - 110 mm, respectively.
The penetration and soil-cultivating machines and tools will depend not only on the distance between the paws in the row, but also on the distance from the plane of their blades to the lower plane of the frame - *H.* At a small distance *H* crop residues in the form of stubble, separately encountered heaps of straw or evenly scattered over the field, are clogged between the soil surface and the frame of the machine, which leads to a disruption in the technological process. The total height *H* should be determined from the following expression

$$H = a + h_0 + h_1 , \qquad (61)$$

where: a - depth of processing (6 - 30 sm); h_0 - is the altitude of the flight trajectory of the soil after the removal of the paw from the surface of the knife; h_1 - is the maximum height of stubble, small heaps of straw, or the thickness of a scattered layer of straw on the surface of the field (approximately 25 - 30 sm). Taking into account the altitude of the trajectory of the flight of the soil after its descent from the knife of the paw, the formula for determining$H$ becomes

$$H = a + h_1 + \frac{(\upsilon_n^*)^2 \cdot \cos^2\gamma \sin 2\alpha'}{2g}\left[\frac{tg^2\gamma}{(\cos\alpha_0 + tg\rho_{a_0} \cdot \sin\alpha_0)^2} + \left(\frac{f}{f^1 + f}\right)^2\right]. \qquad (62)$$

Calculations show that the minimum allowable distance *H*for plane cutters is 500 mm, and for deep loosers - 700 mm.
It should be noted that if two or three-row frontal arrangement of working bodies on the frames of machines intended for loosening of soils on dump backgrounds satisfied the agrotechnological (initial) requirements to some extent, then in the case of flat cutting such a layout, in view of the presence on the field surface of a large number stubble residues, contributes to clogging the machine, i.e., the appearance of a "grabbing" effect. The appearance of this effect is also facilitated by the fact that the first, second and third rows of flat-topped paws operate in absolutely unequal conditions: the first row - in conditions of blocked cutting, the second - semi-blocked, and the third - free cutting.
The low depth of loosening, which determines the presence of a small amount of backup, and the work of the last row of working elements in free cutting conditions cause an uneven surface of the treated field. In some cases, the working organs of the last row are almost completely, as shown by the results of tests of heavy cultivators and seeder-cultivators (without support rollers), open the bottom of the furrow. Therefore, from the point of view of providing a level surface of the treated field and the patency of the machine when working on stubble backgrounds, it is necessary to find such a scheme for arranging the working bodies so that the conditions of their operation in all rows are identical or use the leveling surface of the treated field device.

To test the impact of the working conditions of the paws (free, semi-free, blocked) on the performance of the machine and its patency, a laboratory installation was made (Fig. 35), which allows the working bodies to be placed on the frame in various ways.

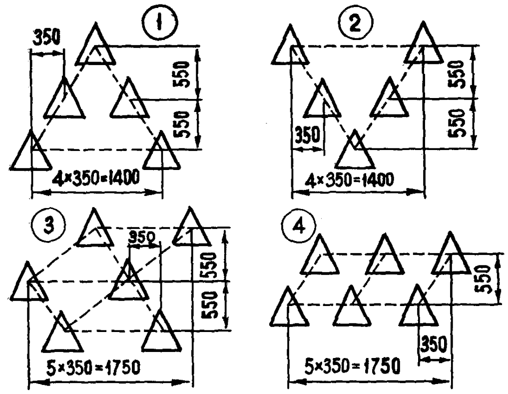


Fig. 35. Variants of placement of paws on the frame of the field installation

Variants 3 and 4 provided for the conventional arrangement of the paws on the frame of the machine, and variants 1 and 2 positioned them at an angle to the direction of travel. In option 1, a larger number of paws, with the exception of one, operate under the same conditions - semi-free cutting. In option 2, two paws work in conditions of blocked, two - semi-free and one free cutting. The parameters of the working bodies are the same ($2\gamma = 75°$, $\alpha_2 = 16° 40'$; $2b^* = 420$ mm), the overlap between the legs $\Delta b = 70$ mm, the height of the racks is 500 mm, the distance between the rows of paws is 550 mm.

The results of experimental studies have shown that, depending on the location of the paws, the surface profile of the field after treatment significantly changes. The most leveled surface is formed after the passage of the installation with the placement of the paws according to the first variant - the average height of the crests is 2,8 sm. When the paws are placed at an angle backwards (version two) behind the last working member operating under free cutting conditions, i.e. with insufficient back- A furrow with a depth of 4,8 sm is formed after the passage of the installation, on which the paws are placed in the third variant; behind the feet of the first row a furrow is formed with a depth of up to 2 sm (blocking cutting conditions), behind the second row (semi-blocked cutting condition) furrow depth virtually not different from the depth of the furrow formed by the front row of claws. Behind the paws of the third row, working under conditions of free cutting, the groove depth reached 6sm. Consequently, from the point of view of the leveling of the field surface, the paws on the frame of the machine should be placed with an angle forward.

Approximately the same results were obtained when studying the technological schemes of a heavy paw cultivator [19]. The investigations were carried out on the fields of the SKF VIM with the help of a special installation, the design of which allowed changing the patterns of placement of the paws (A, B, C) and the distance between them in the longitudinal (l) and transverse (b) directions (Fig. 36). Variants were tested: A (700x310); A_1 (400x350); B_n (700x310) and B (400x310), where the first factor indicates the distance (in mm) l, and the second factor - b.

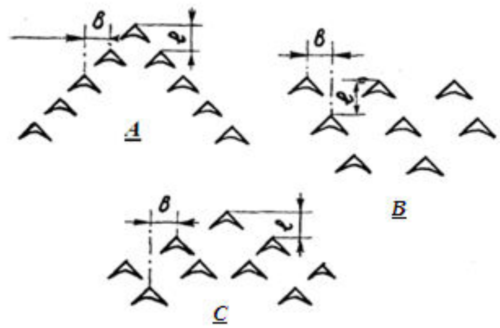


Fig. 36. Technological schemes of heavy cultivator: *A*) wedge-shaped single-row; *B*) frontal three-row; *C*) double wedge

Comparison of the results of the agrotechnical evaluation of these options showed that the quality of the cultivator at l= 700mm is much better than at l = 400 mm: by the combing of the field in 1,3 - 2,7 times, the width of the furrows is 1,1 - 1,75; according to the depth of the sulcus 1,1 - 1,8, to maintain stubble in 1,1-1,75 times. Further studies of the circuits were carried out at l = 700 - 800 mm.

The results of the investigation of the circuit variants at$l$ = 700 - 800 mm showed that the best leveling of the field surface was after the passage of the installation with a wedge arrangement of the working bodies. However, as noted, none of the studied schemes for this indicator, as well as the degree of crumbling of soil on solid backgrounds, does not correspond to agrotechnological (initial) requirements. Therefore, heavy cultivators must, in order to improve these indicators, be equipped with a special device. In the presence of such a device, three-row paw placement on the frames of heavy cultivators is preferred. This arrangement of the paws was used not only on heavy cultivators (KPE-3.8A, KTS-10-1, KTS-10-2, but also on machines for presowing soil cultivation on stubble backgrounds (OP-8 and OP-12).

When working on dry soil with a large amount of plant residues, in order to prevent clogging of the chisel plow, it is necessary to place the operating elements in the row at a distance of not less than 500 mm and along the way - at least 700 mm. In this case, the height of the racks of the chisel working element must be at least 700 - 750 mm.

It is established [20, 21] that the arrangement of the chisel working bodies affects the quality of work and the patency of the machine. Thus, a frontal two-row arrangement of chisel working tools can be used on chisel cultivators designed to work at a depth of not more than 27 sm. The three-row frontal arrangement of chisel working tools on the frame with the above parameters of their placement provides the machine with a high-quality technological process and patency at a loosening depth of up to 40 sm. The main drawback of this arrangement of working tools is the bulky tools and the large material consumption.

The wedge-shaped arrangement of the chisel working bodies allows to shorten the length of the tool and reduce its material consumption. However, when treating heavy compacted soils to a depth of more than 33 sm, the soil is blocked in the spaces between the posts of the three middle paws following the track of the tractor. Clogging is eliminated if the front chisel body is pulled back.

The parameters of the relative location of the flat-topped paws and the scrapers depend not only on the qualitative characteristics of the machine as a whole, but also on its traction resistance [22]. The rational depth of the splitting should be 10 - 20 sm below the depth of the flat-paw, and the slit is located behind the paw at a certain distance L'_{min} (see Fig. 20).

The minimum distance L'_{min} between the legs of the paw and the slit is selected from the condition that the zone of deformation of the soil in front of the splitter does not have to reach the blade of the foot. Therefore, L'_{min}it is proposed to determine by the formula [22]

$$L'_{min} = l_o + a_2 \mathrm{tg}(\beta_0 + \varphi) - l_1 + \frac{b_3}{2tg\gamma} + \frac{a_2 \mathrm{tg}\varphi}{\cos(\beta_0 + \varphi)tg\gamma} l\,. \qquad (63)$$

1.

The decoding of the quantities included in this formula is given in Section 2.2.3 "Labor organs of a plane cutter". The value of the constructive parameter of the slit-cutter L'_{min}, found from (63), will be 0,38; 0,57 and 0,76 m with the difference between the depth of loosening of the paw and the slit-cutter a_2 respectively equal 10;15 and 20 sm.

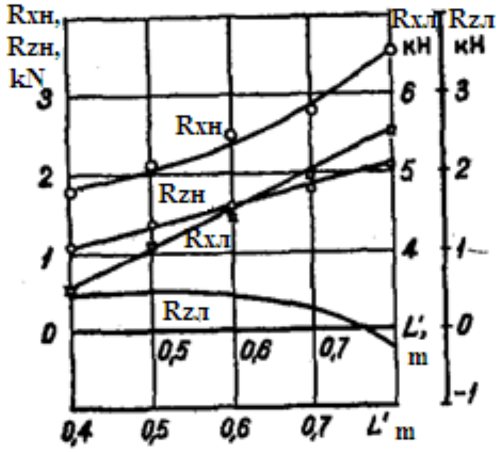


Fig. 37. Horizontal $R_{xн}$и$R_{xл}$ and vertical $R_{zн}$и $R_{zл}$components of the resistance of the soil of the slit and paw, depending on the limiting distance between them L'

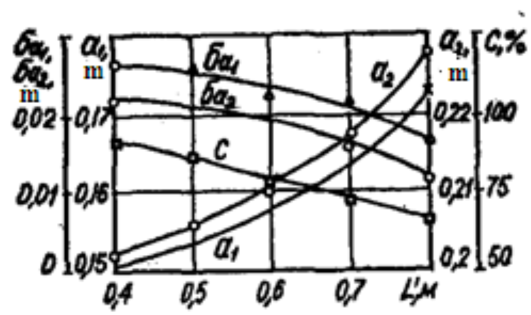


Fig. 38. Influence of the longitudinal distance between the slit-cutter and the paw L' on the depth and stability of loosening of the soil with the paw (a_1; σ_{a_1})) and the slit-cutter (a_2; σ_2)), and the effect on stubble preservation.

From the analysis of the results of experimental studies to determine the effect of the magnitude L' on the power (Fig. 37)and agrotechnological (Fig. 38) performance of the plane slit-cutter follows that with increasing distance, the traction resistance increases, and the stability of the stroke of the working organs is improved in depth. However, this increases the width of the furrow groove and reduces the degree of stubble conservation. Thus, if the depth of the splitting is greater than the depth of loosening of the planar cutting foot by 10 - 20 sm, then it is recommended to place the slit behind the paw at a distance of 35- 54 sm.
Presence of stubble residues and weeds on the surface of the field contributes to the possibility of molding the soil between the discs. Therefore, the heavier the working conditions, i.e., the probability of occurrence of pressing the soil between the discs, the greater the coefficient k in determining the diameter of the disc (see formula (32). In order to avoid seizing the formation and individual blocks of soil between the discs, the size B should be greater than the depth of the disc stroke [14], i.e., $B \geq 1{,}5a$(Fig. 39).
However, when choosing the value for D and b the harrows and stubble cleaners, one should keep in mind not only the clogging of the discs, but also the receipt of the bottom of the groove of the given profile. The height of the ridges C at the bottom of the furrow depends on the diameter of the disc, the angle of installation α and the distance$B$ is theoretically determined by formula (36).

With the battery placement of disks:

$$Bcos\alpha = D_c sin\alpha. \tag{64}$$

Butin$D_C$there is an average proportionality between $D - C$ and C (11,14,15)

$$D_c = 2\sqrt{C(D - C)}. \tag{65}$$

Solving jointly (64) and (65), we obtain

$$B = 2tg\alpha\sqrt{C(D - C)}. \tag{66}$$

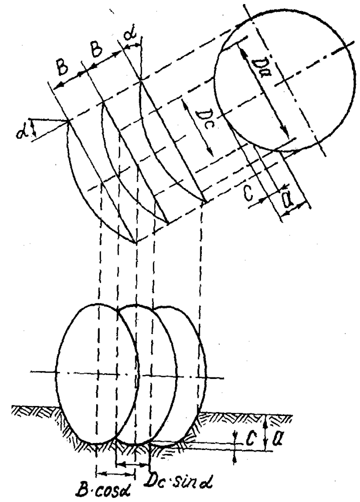


Fig. 39. To definition of placement of disks on the battery of stubble cleaners

When designing, it is often necessary to determine one of the values B, α, C and D from the selected values of the other three. To facilitate the selection of values B, C, D and α, corresponding to the technical specification, it is advisable to use the nomogram (Fig. 40) [14].

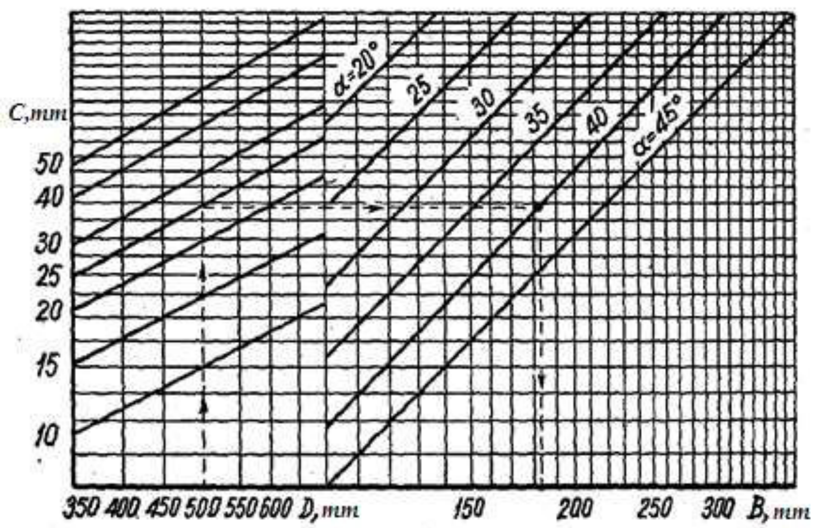


Fig. 40. Nomogram for selecting the values of the parameters B, C, D, and α of the battery of the disc stubble cleaner.

In the case of disc lenses, the height of the ridge $C \leq 0{,}5a$. Non-observance of this relation when choosing C for stubble cleaners adversely affects the quality of performance - remains uncleared part of the weeds. Because of the small value of the angle α for disc harrows, the condition $C<a$would lead to an excessive rendezvous of the discs, so the harrows are made two-sided, shifting the discs of the rear batteries in the transverse direction with respect to the discs of the first row by the amount $B/2\ cos\ \alpha$.
The angle of attack α of disk stubble cleaners as a function of the physical and mechanical properties of the soil and the type of technological operation is set at 15°- 35° [11,14,15].
Needle disks on the harrows BIG-3A, BMSH-15, BMSH-20 are also assembled into batteries. Taking into account the parameters of deformation shells formed by the disc needles, the interdisk distance and the angle of attack for a single-row and multi-row battery are selected by the formulas (49) and (50).

For a two-row battery, the distance between the needle discs is assumed to be 200 mm, and the angle of attack is 17 - 22°.[16].To increase the number of weeds seeded in the soil and to reduce the traction resistance, the angle of attack of the second-row needle discs is reduced compared to the first for 5° (Fig. 41).

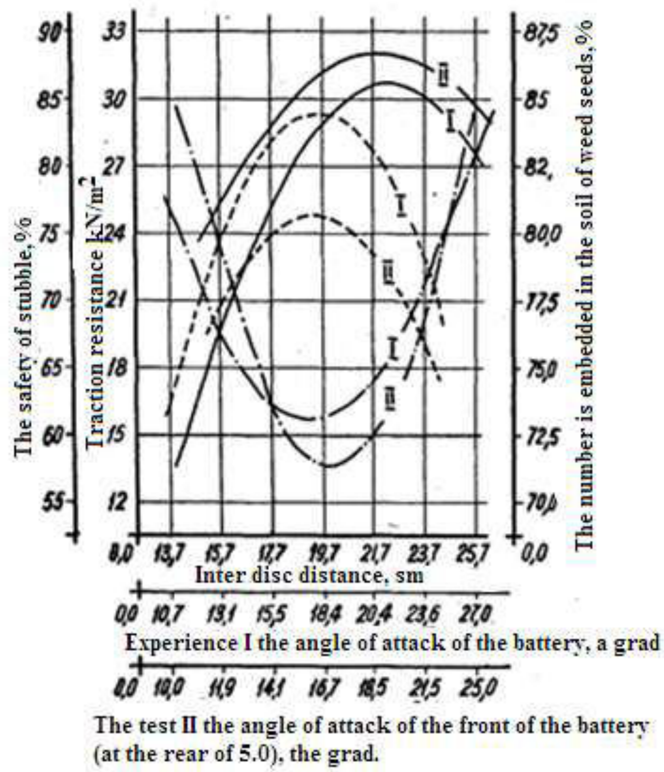


Fig. 41. Stubble conservation (- - -), specific traction resistance(– · – · – · ·) and sealing seeds (-) of a two-row battery with needle discs into the soil, depending on the parameters of their placement

When combining in one-unit lancet paws and rod rod, the bar should be placed behind the paws. The minimum distance between the tips of the paws of the paws and the bar will be determined by the range of flight of the soil layer after the descent from the paw and is found by formula (59).

2.4. Stability of the course of working organs in the depth of soil-cultivating machines and implements of soil-protecting agriculture

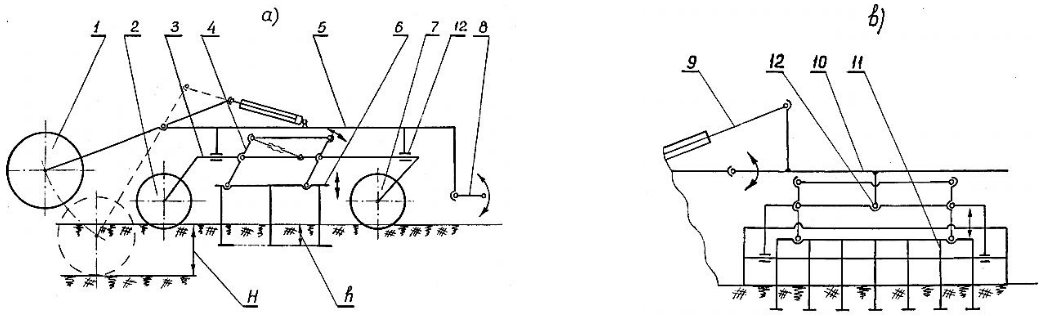
Agro-technological (initial) requirements for soil-cultivating machines and implements of soil-protecting agriculture and the conditions of their operation, the level of energy intensity that determines the width of capture, as well as strict limitations on deviations from a given depth of loosening, predetermine the need to select the type and justification for the parameters of the principle constructive scheme of the most promising combined tools for shallow soil treatment [24].

The purpose of creating combined tillage tools is to provide in a timely manner, in agrotechnical terms, the qualitative performance of several technological operations for processing soils with minimal energy and labor costs, which is achieved, as a rule, in two ways.

The first - the compilation of aggregates by the sequential combination of several single-use tools. Such units are not widely used due to the complexity of their operation.

The second is the placement on the frame of the gun of several types of working bodies. The types of working organs, their number and placement on the tool frame, the desire to simultaneously

perform several technological operations qualitatively and in agrotechnical terms and predetermine in principle a basic constructive scheme of combined tools for shallow processing of soils.
The most widely used semi-mounted combined tools for shallow soil treatment with the arrangement of transport wheels behind the frame (Fig. 42). According to this scheme, a number of models of soil-cultivating combined tools have been developed: ESA TUME 4000; Agropak-RABE; MARS-UNIA; EUROPAK AMAZONE; AKSh-6; KPZ-9.7; models K450F, K500A, K600A, Gigant 800, Gigant 1000 from LEMKEN and others.



a) side view;*b)* back view

Fig. 42 Schematic diagram of semi-mounted combinedtools for shallow processing of soils

Such tools consist of a central section (5), on the rear bar of which the transport wheels (1) are installed, providing a given ground clearance, and on the front bar the scele (8) with a height-adjustable trailer point. To increase the width of the grip, the side sections (10) are hingedly attached to the frame of the central section, which, when the implement is moved to the position of long-distance transport, is lifted and fixed in a transversely vertical plane by means of the mechanism (9). In the near-transport position, the side sections are supported by stops (restraints) rigidly connected to the frame of the central section.
The frames (3) with the front (7) and rear (2) support-rolling unrolling rollers are attached to the main frames of the central and side sections by means of longitudinal-horizontal joints (12).
The hinges (12) allow the frames (3) to oscillate in the transverse-vertical plane, which allows them to copy the relief of the field surface in a given plane. This makes it possible to attach several such frames to the wide and rigid frame of the central or lateral sections of the gun. The position of the rollers (2 and 7) relative to the main sections of the sections does not change and they are the supports in the process of work, between which are placed special frames (6) with the loosening working elements fixed to them (11). The specified depth of loosening of these working elements is established by means of a parallelogram mechanism (4).
Such tools have the following advantages:
- the weight of the implement does not depend on the load capacity of the tractor attachment;
- to frames (3) with rollers, it is possible to hinge frame (6) with various types of working elements for loosening, laying, additional crumbling of the treated soil layer;
- longitudinal-horizontal joints allow lateral sections (10) and frames (3) to oscillate in the transverse-vertical plane, which provides wide-cutting tools with copying the relief of the field surface in this plane;
- placement of working elements (11) between two supports (2 and 7) at the minimum distance allowed from them provides the tool with satisfactory copying of the field surface in the longitudinal-vertical plane;
- hinged articulated frames of the central and lateral sections and articulated suspension to them of frames (3) of small width of capture, as a rule, no more than 2 meters, allow to create block-modular highly-rated units to tractors of various classes.
The shortcomings of the guns executed under this scheme include:
- a large mass of the gun 550 - 760 kg per 1 m of capture due to the presence of guns located in three horizontal planes;
- a large number of the same type of adjustment mechanisms (4), the number of which is equal to the number of frames (6) with working elements (11);

- as the front row of rollers (7) compacts the soil, and afterwards the loosening working elements are installed, their traction resistance sharply increases.

A basic design diagram of some semi-mounted combined tools for shallow soil treatment, for example, UNIA VIKING, KPM-4, etc. (Fig. 43), differs from the one shown in Fig. 42 as follows.

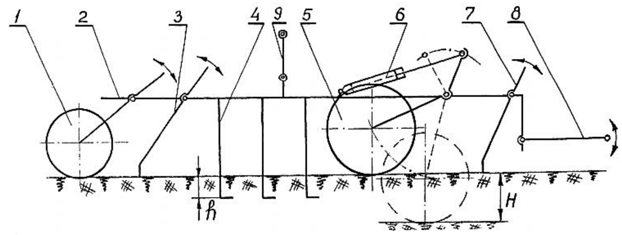


Fig. 43. Schematic diagram of semi-mounted combined implements for shallow soil treatment with height-adjustable height adjustmentof transport wheels

Instead of the front row of support rollers, support wheels for height-adjustable wheels (5) are installed, which makes it possible to set the required depth of loosening of the soil (h) and to transfer the implement to the transport position with a given ground clearance (H). In this case, as a rule, a pair of support wheels is installed in the middle section, and on the side sections, pivotally connected to the middle section, one at a time.

There are no frames located in different horizontal planes hingedly connected to each other. The loosening work tools (4) are rigidly attached to the tool frame (2), and the leveling (3 and 7) and supporting-tightening (1) operating members are pivotally attached to the frame (2), spring-loaded and height adjustable. In this case, the depth of loosening of the working elements (4) only changes with the help of support wheels (5).

In case the roller (1) can be rigidly fixed with respect to the frame (2), the required soil loosening depth is set by adjusting the position of the individual roller (1) and the wheel (5) separately.

Advantages of such tools:

- replacement of the front row of support rollers with support and transport wheels eliminates the compaction of the soil throughout its width, before loosening it, which reduces energy consumption;
- the absence of frames in different horizontal planes allows to reduce the material consumption;
- the adjustment of the loosening depth of the soil is simplified if the roller (1) is not rigidly fixed with respect to the section frames, since in this case the depth of loosening is established only with the help of wheels (5).

The shortcomings of the guns executed under this scheme:

- the absence of frames hingedly connected to the main frames of sections, limits the possibility of increasing the width of the tool, the copying of the relief of the field surface in the transversely vertical plane will deteriorate;
- the depth of loosening of the soil becomes more complicated if the roller (1) is rigidly fixed with respect to the frames of the implement;
- if the soil loosening depth is established only by means of the wheels (5) and the rear rollers (1) are spring-loaded, then when the distance of the working elements (4) from the wheels (5) increases in the longitudinal plane, the stability of the loosening depth will deteriorate;
- a large load on the wheels (5) in the transport position, because the center of gravity of the gun will be shifted back relative to them by a considerable distance.

A schematic diagram of the hinged combined tillage tools is shown in Fig. 44. According to this scheme, Combinator II guns of Kverneland, APK-3, KPSh-8, KBM-4,2 etc. were created. Loosening work tools (9) and the planner (8) are attached to the main frames of the central (3) and side sections. The rolling roller (1) is pivotally attached to the same frames and spring-loaded. Therefore, the depth of loosening is regulated only by means of support wheels (4) and a special mechanism (5).

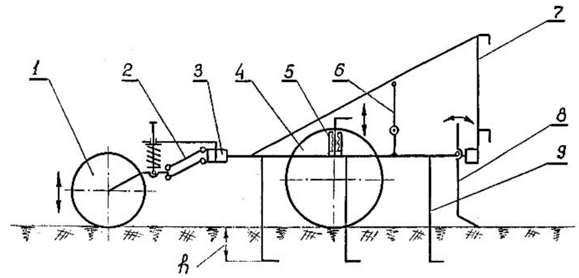


Fig. 44. Schematic diagram of hinged combined guns for shallow soil treatment

The gun, made on a hinged scheme, has the following drawbacks:
- strict requirements are imposed on the total mass of the implement, since it is limited by the weight of the tractor hitching mechanism and, in general, the controllability of the implement. At the same time, such a scheme is acceptable for the creation of narrow-block block-modules, which allow the creation of wide-swing aggregates with the help of a special hydroficated coupler equipped with mechanisms for hinging them. But such an approach is effective only when such units for tractors of different traction classes will be manufactured in the factory conditions, i.e. at the request of the consumer;
- to ensure the stability of the movement of the unit in the longitudinal-vertical plane and its controllability, the center of gravity of the implement should be as close as possible to the axis of suspension of the tractor attachment mechanism or to the suspension axis of the hitch linkage mechanism.
In order to increase the stability of the working bodies in depth along the longitudinal and transversely-vertical planes, many firms (ECA TIME of Soume Sakeri, UNIA KOMBY, System Korund of LEMKEN, etc.) developed hinged combined tillage tools according to the scheme shown in Fig. 45.

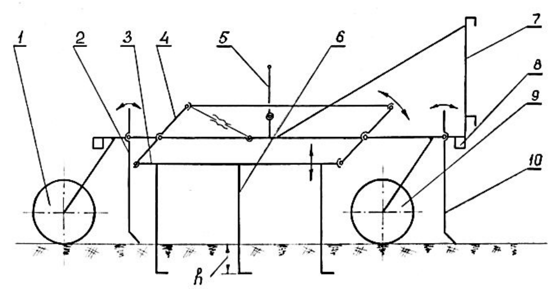


Fig. 45. Schematic diagram of hinged combined tools for shallow soil treatment with a height-adjustable frame with loosening working bodies

The stability of the loosening of the soil in such tools is achieved in them due to the fact that the loosening working elements (6) are connected to a frame (3), adjustable in height relative to the frame (8) by means of a parallelogram mechanism (4). The frame (3) is located between two rows (1, 9) of support rollers rigidly attached to the frames (8) of the main or side sections to which the planners (2, 10) can also be attached.
The main shortcomings of guns executed under this scheme are:
- increased material consumption of the tool due to the use of a double frame. This imposes restrictions on the increase in the width of the tools because of the weight of the tractor hitching mechanism or the hitching mechanism of the special hydraulic coupling and ensuring proper stability and controllability of the unit;
- the energy intensity of the loosening process of the soil increases, since the soil is compacted before the loosening working organs.
Many firms and enterprises (Models TTA-firm Kverneland, COIL TINE MULCHER-firm CASE CANADA CORPORATION, Mulch Finisher-firm Iohn Deere, founders of tools APK-7,2, PVK-3,6, PVK-7,2) developed trailed combined tools for shallow processing of soils, which differ from those described above only by the way of connection with the tractor.The shortcomings of the guns executed under this scheme include: In this case, the depth of tillage by loosening working

bodies is regulated by a change in the height of the support wheels relative to the frames of the main and side sections. The position of the rollers and the planners, hingedly connected to the frame sections and spring-loaded, with respect to the frames is set taking into account the depth of soil loosening by means of a special mechanism.

Trailed combined tools for shallow soil treatment have the following drawbacks:

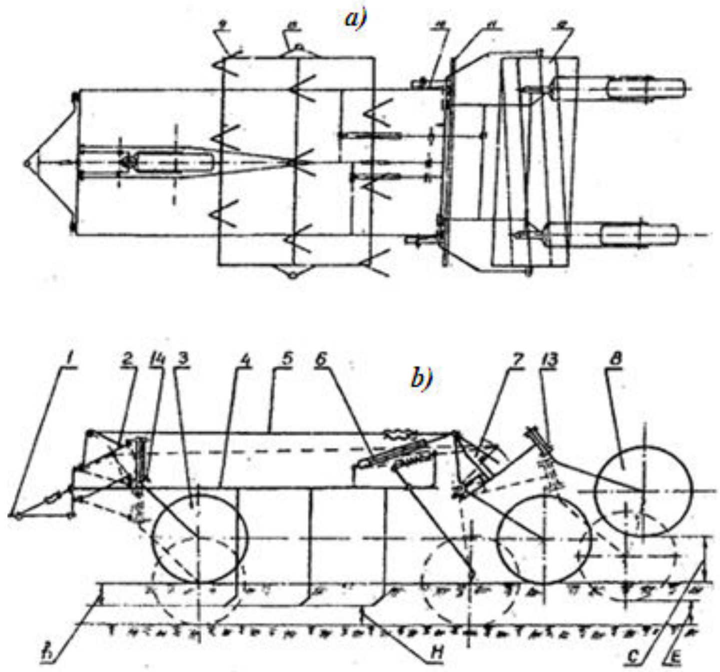
- the supporting, height-adjustable wheels should be as close as possible in the longitudinal-vertical plane to the loosening working bodies, otherwise their depth stability will deteriorate;
- the number of the same type of adjustment mechanisms is increasing, since the position of the rollers and the planners relative to the frame must be changed with the next change in the depth of loosening of the soil;
- the working width of the sections is limited due to the need to ensure the proper stability of the stroke of the working bodies in depth in the transverse-vertical plane;
- when transferring the side sections to the position of long-distance transport, the load on the support wheels sharply increases, so the wheels are placed on the central section either reinforced or doubled (tandem).

Thus, the analysis showed that the creation of combined tools for small-scale soil cultivation is promising, justified, and they have found wide application in the practice of farmers in various countries of the world. However, the analysis made it possible to identify a number of significant shortcomings of such tools, carried out according to various fundamental constructive schemes, which predetermines the need to search for new schemes in which the advantages of existing ones are used and their shortcomings are eliminated as much as possible. So, for example, it is advisable to use the block-modular principle in creating wide-spread combined tools for shallow tillage, which allows creating aggregates to tractors of different traction classes with the help of unified narrow-clamping modules and couplers. To ensure the necessary stability of the stroke of the working organs in depth, the module must have a working width of 1,5 - 2,0 meters. This will eliminate the need to use multi-tiered tool frames in the horizontal plane and reduce the material consumption. To simplify the connection of the module to the hitch, it must be trailed.

To transfer the module to the position of long-distance transport, use special transport wheels mounted on one axis of the frame of the rear skid roller. During operation, the transport wheels are installed with a special mechanism so that they do not touch the surface of the field.

In order to avoid collision of adjacent frames of trailing modules in the process of work, to prevent the formation of flaws at their joint, to provide for the possibility of transporting wide-angle aggregates in the transverse direction of the frame of adjacent modules, they must be connected by special hinges that enable the modules frames to move relative to each other in at least three mutually perpendicular planes.

To implement these provisions, the principal constructive scheme of the trailed narrow-gripping module is proposed (Fig. 46).



a) - top view*; b)* - side view

Fig. 46. Schematic diagram of a narrow-grabbed trailing module of a combined tool for shallow soil treatment

The basis of the module is a flat frame (4), on which are mounted:
- a sledge (1) with a trailer height adjustable;
- the front support-transport self-aligning wheel (3), moved in height relative to the frame by means of a parallelogram mechanism (2), adjustable along the length of the rod (5), the hydraulic cylinder (6);
- loosening (9) and leveling (11) working bodies;
- the supporting and compacting roller (12), hingedly connected to the frame of the module, whose height position is adjusted by means of a pull rod (5) and a hydraulic cylinder (6) in synchronization with the front wheel-drive wheel (3).

A frame with transport self-aligning wheels (8) is pivotally attached to the suspension axis of the frame of the rollers (12). These wheels with the help of a special mechanism (7), locks of self-aligning wheels (13, 14) and intermodular hinges (15) allow to transfer the wide-grip unit to the position of long-distance transport. The location of the center of gravity of the trailing module between the three points of support in the position of long-distance transport provides him with the necessary stability and allows to exclude the mechanism for blocking the frames of adjacent modules during transportation of the gun to a long distance.

Wide-gripping units from such modules are formed with the help of trailing hinge-sectional couplings or wheelless (coupling units of seedersSZS-6/12) or with support wheels (coupling type SP-16).

The use of a coupling without support wheels necessitates the installation of markers with a large shoulder of the furrow-forming body at the extreme narrow-clamping modules (sections), which sharply worsens the stability of the course of their paws in depth in the transverse-vertical plane. Therefore, to exclude a permanent tilt in this plane, the extreme modules in the unit are mounted on them from the markers side, height-adjustable support wheels, and with adjacent modules they are connected by additional connecting devices with a limited number of degrees of freedom.

In the position of long-distance transport, this coupling does not allow detaching modules from it. In this case, the adjacent modules in the horizontal plane are connected by a rigid triangle - two points of the trailer on the coupling and a connecting link between the modules, which drastically reduces the maneuverability of the unit in the horizontal plane and its safety.

The use of a single hydraulic cylinder on the module to extract the paws at the end of the paddock and when transferring it to the position of long-distance transport by means of a single rear transport wheel causes the use of a special and kinematical mechanism that increases the time required to perform this operation and reduces the overall reliability of the unit. The presence of only two points of support in the position of long-distance transport causes its instability in the longitudinal-vertical plane, which predetermines the need for rigid fixation of the frames of the extreme modules with adjacent special locking devices.

The above disadvantages of such units are eliminated by replacing the unsupported coupling with a trailing hinge-section hitch with the support wheels, and also by replacing one rear module transport wheel with two.

The presence of support wheels on the coupler makes it possible to mount markers on its outer sections when forming an assembly of 4, 5, 6 modules, and in the middle section, when forming an assembly of 2-3 modules. This makes it possible to eliminate the misalignment of the extreme modules in the transverse-vertical plane during operation and to exclude from the module design the support-limiting wheels and special connecting devices between the frames of the extreme and adjacent modules. When the unit is moved to the position of long-distance transport, the modules are disconnected from the coupling, which eliminates the presence of a rigid triangular coupling in the horizontal plane between adjacent modules and increases maneuverability of the entire unit in the horizontal plane and safety during transportation.

The use of two transport wheels instead of one in the module ensures a stable position of the module, since its center of gravity will be located within the three support points and allows to eliminate the blocking device connecting the rigid frames of the extreme and adjacent modules from the unit construction.

The use for lifting and lowering the transport wheels of a separate mechanism with a hydraulic cylinder, kinematically not connected with the mechanism and the hydraulic cylinder of recess and penetration of the working elements, greatly simplifies the design and reliability of the module and shortens the time for performing the transfer of the unit to the position of long-distance transport.

The use of the hinged sectional coupling with the support wheels makes it possible to install a centralized tank for seeds and fats and dosing, distribution and pneumatic conveying devices to the openers on its middle section. it is possible to create wide-block block-modular tillage-seeding units for tractors of various classes.

The stability of the progress of the working organs in the depth of soil-cultivating machines and implements of soil-protecting agriculture depends not only on their type of constructive scheme and its parameters, but also on the statistical characteristics of external perturbations acting on them in the course of work.

With the soil protection system of agriculture, changes in the height of the unevenness of the fields and the hardness of the soil in the treated layer are random quantities with a distribution close to the normal law. The height of the unevenness of the investigated fields varies from 2,9 to 10 sm with a variance of 0,74 - 11,5. The hardness of the soil in the treated layer varies from 0,82 to 4,35 MPa with a dispersion of 0,39 - 3,75. The frequency of the perturbing influences is characterized by a wide range: for uneven fields, it lies in the range 0 - 9 m^{-1}, and for soil heterogeneity by hardness - 0 - 8 m^{-1}. With the movement of soil-cultivating machines and tools in fields with a soil hardness of more than 0,1 - 1,0 MPa, the nature of the variation in the irregularities as an external disturbing factor under the influence of a support wheel with a load of 4,0 - 5,0 kN remains practically unchanged.

It is assumed that the width of the section of tillage machines, in which the working members are rigidly fixed to the frame, is limited by the uniformity of the loosening depth in the transverse-vertical plane. The width of the grip of the machine with a rigid attachment of the working element to the frame should not exceed 2,5 m, and for fields with an even microrelief - not more than 3,5 m [14].

A further increase in the width of the grip of the machine leads to a deterioration in the lateral stability of the movement of the working organs in depth, especially in shallow processing. On the other hand, it is almost impossible to achieve the optimum loading of the tractor engine and obtain the most advantageous economic and operational indicators with such a small catch width of tillage machines and tools. To reconcile these two opposite but fundamental requirements, one can either proceed along the path of creating narrow-engaging trailer machines, or along the path of creating hinge-section no-tractor machines to tractors of different traction classes. But in both cases, it is necessary to solve the problem of the permissible width of the gripper of the machine (section) with a solid frame with rigid attachment of working parts to it.

When justifying the recommended capture width of a rigid section of conventional cultivators, the random nature of the effect of unevenness on the support wheels [23]. The replacement of microroughness by macroroughness is illegal for the steppe zone, since the length (radius) of the latter is incommensurable with the width of the grip of the machine. The length of the arc, described by such a radius, will fit the machine with a width of 4 - 5 m, and the height of the segment will be several times less than the permissible deviation from the average depth of loosening. From our point of view, the deviation of the depth of loosening, for example the cultivator-plane, in the transverse-vertical plane is largely influenced by the microroughness of the field surface, which can be copied by supporting wheels.

To justify the allowable width of the grip of the rigid section and the track of the support wheels, taking into account the unevenness of the fields and their random nature, it is expedient to use the relations [37]:

$$D_{y_c} = \frac{2D_{z_n}}{B_0^2}[a_C^2 + a_c B_0 + B_0^2 + (a_c B_0 - a_C^2) \cdot \rho(B_0) - a_c B_0 \cdot \rho(B_0 - a_c) - $$

$$-(B_0^2 - a_c B_0) \cdot \rho(a_c)] \tag{67}$$

but for the console (radius)

$$D_{y_b} = \frac{2D_{z_n}}{B_0^2}\left[l_b^2 + B_0^2 + B_0 l_b - \left(l_b^2 + B_0 l_b\right) \cdot \rho(B_0) - \left(B_0 l_b + B_0^2\right) \cdot \rho(l_b) + \right.$$

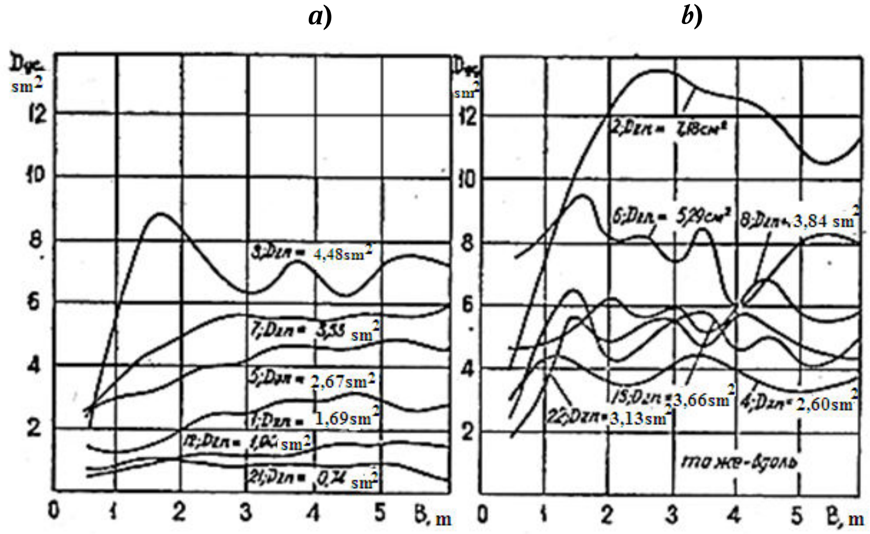
$$\left. + B_0 l_b \cdot \rho(B_0 + l_b)\right] \tag{68}$$

where: D_{z_n}- dispersion of the height of the unevenness of the field; D_{y_c}-dispersion of the vibration of the midpoint of the frame relative to the surface of the field, provided that the wheel gauge is equal to the width of the tool;D_{y_b}- dispersion of the vibration of the frame point at the outset with respect to the support wheel; $\rho(B_0)$,$\rho(a_c)$, $\rho(B_0 - a_c)$, $\rho(B_0 + l_b)$ - the values of the normalized correlation functions of the irregularities for the corresponding values $L_i$equal B_0, $B_0 - a_c$etc.

The corresponding values$\rho_i(L_i)$ are taken in the direction perpendicular to the movement of the tool. Using the values of the functions $\rho_i(L_i)$for the unevennesses of the fields investigated, the statistical characteristics of which are given in our works [8,26], dispersion D_{y_c} is calculated as a function of the width of capture of the rigid section both along and along the previous treatments (Fig. 47).

Analysis of these data shows that the oscillations of the mid-point of the frame relative to the surface of the field, and, consequently, the depth of loosening, depend mainly on the variance of the height of the unevenness of the field. With an increase in the width of the tool, the gun increases from 0,5 to 1,5 m, D_{y_c} while at B_0<1,5 m it remains practically unchanged. When the machine moves across the previous machining, i.e., when the profile of the unevenness of the fields of the previous treatments is applied to the support wheels, the variance D_{y_c} is substantially smaller than when the machine moves along the machining. This once again confirms the feasibility of implementing the generally accepted truth that the forthcoming treatment must be conducted across the previous one. With an allowable deviation of the loosening depth from the specified± 1.5 sm (D-2,25 sm^2) machine with a working width of 0,5 - 6 m, it will not meet the demand if the dispersion of the unevenness of the fields is more than 1,5 - 2 sm^2. Therefore, to obtain acceptable deviations it is not advisable to reduce the width of the gripper of the machine, but it is necessary to level the field surface. This will allow not only to create wide-spread tillage machines of simple construction and convenient in operation, but also will allow the units to operate at higher speeds.

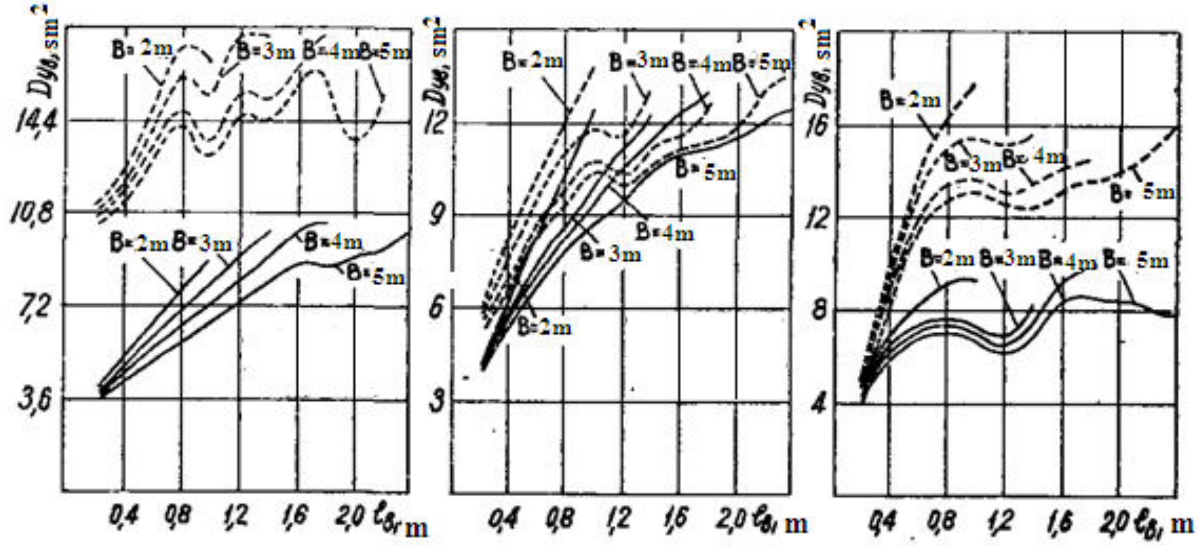
Otherwise, the tolerance is ± 1,5 sm, which is clearly visible from Fig. 47. is unreal and needs to be clarified, since the variance of the height of unevenness of the stubble backgrounds lies mainly in the range 3 - 6 sm^2.



a) - movement of the aggregate across the direction of the previous treatment
b) - movement of the aggregate along the previous treatment

Fig. 47. Dispersion of the deviation of the mid-point of the frame relative to the surface of the field, depending on the width of its capture
(1-8 - stubble of wheat after sowing with various seeders;
12, 13 - uncontrolled plowing; 21,22 - five-year-old fat)

The insignificant effect of the width of the grip of the machine with a rigid frame on the dispersion of the depth of the working bodies is confirmed by the state tests of flat-top machines with different working widths (from 2,2 to 4,5 m) conducted by the Tselinnaya, Siberian, Altai and Pavlodar MIS in 1962 - 1975. Thus, for example, according to long-term data, the dispersion of the loosening depth in the transverse-vertical plane of the plane with a capture width of 2,2 m lies in the range 2,27 - 12,32 sm^2; from 3,0 m – 1,7 - 13,76 sm^2; 3,5 m -1,48 - 8 sm^2 and with 4,5 m – 1,21 - 8 sm^2. Of course, these data were obtained without taking into account the unevenness of the field surface in each specific case, but in general they confirm the assumption that the width of the capture is insignificant in these limits for the stability of the stroke of the working organs in depth in the transverse-vertical plane and the need for alignment (repair) of the fields.
To reduce the deviation of the stroke of the working bodies from a given depth of loosening in wide-angle machines with a rigid section, it is possible by selecting the optimal track of the support wheels. The deviation of the extreme point of the frame, depending on the length of the departure relative to the support wheel and the width of the gripper of the machine, is reflected in Fig. 48.



—–the movement of the unit transversely of previous treatment (profile 5, Fig. 47)
– – – the movement of the Assembly along the previous treatment (profile 6, Fig. 47)

—–the movement of the unit across the preceding treatment (profile 7, Fig. 47)
– – –the movement of the unit along the upstream processing (profile 8, see Fig. 47)

—–the movement of the unit across upstream processing (profile 3, Fig. 47)
– – –the movement of the unit along the upstream processing (profile 4, see Fig. 47)

Background – the stubble of wheat, sowing of SZS-9, North of Kazakhstan

Fig. 48. Dispersion deviation of the frame console is relatively surface of the field, depending on the shoulder of its departure and thewidth of the section

As can be seen, with the increase in the arm of the arm of the console, the dispersion D_{y_b} increases substantially: when it is out, equal to half the track wheel track, it reaches its greatest value; With a larger machine width, the arm of the console can be extended. Dispersion D_{y_b} depends, as in the previous case, on the direction of travel of the machine, hence, the arm of removal of the console relative to the support wheel should be minimal. However, this minimum value should, with an adjacent passage of the machine, ensure that the support wheel rolls over the untreated surface and that there is a sufficient overlapping area, i.e., the ratio of track to the width of the gripper should be approximately 0,8 - 0,9.
The stability of the stroke of workers, organs along the depth of the trailing section of the plane plane depends not only on its width and the track of the support wheels, but also on the parameters of the structural scheme. Therefore, we [8,26] carried out research to determine the optimal values of the parameters of the trajectory section of the plane cutter (Fig. 49).
Considering the forces acting during the operation of the trailing section of the plane cutter (see Fig. 49), using the assumptions made for vibrational systems with small deviations, and also assuming that the perturbing forces change according to a periodic law, the physical values of which in the analysis of such systems can be replaced spectra of the first harmonic we obtain the differential equation of motion of the machine

$$\Delta\ddot{\varphi}' + 2n\Delta\dot{\varphi}' + k^2\Delta\varphi' = U_1 f(t), \quad (69)$$

where:$n = B/2l_n$- index characterizing the attenuation of the oscillations of the trailing section (1/s);$k = \sqrt{C/J_n}$- natural frequency of the system (1/s); B- coefficient taking into account the resistance of the soil when the position of the section is changed (H·m·s/rad);C- hardness coefficient

($H{\cdot}m/rad$); J_n— moment of inertia of the section relative to the trailer point (H·m·s^2/rad);$J_n = J_c + \frac{G^0}{g}$; $l^2 = l_c^2 + H_c^2$; U_1- shoulder of application of perturbation forces (m);$f(t) = \sum F_i(t)/I_n$ - perturbing function of a periodic nature; g - acceleration of gravity.

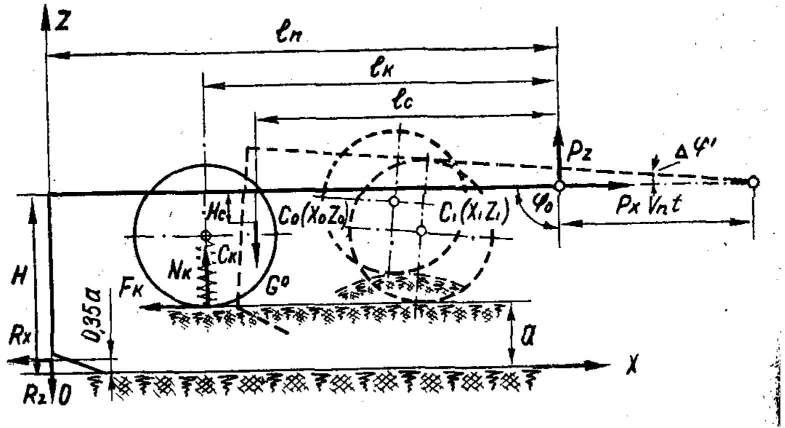


Fig. 49. Schematic diagram of a trailed plane cutter

The stationary solution of this equation, which describes the forced oscillations of the steady motion under the influence of the disturbing force, made it possible to establish a relationship between the displacement of the machine in the longitudinal-vertical plane and its parameters

$$\Delta\varphi'(t) = \rho^* \cdot F(\mathrm{t}). \tag{70}$$

The value of ρ^* is the displacement of any of the points of the frame of the machine (amplitude) in radians under the action of a perturbing force F=9,81 H, applied on the arm $U_{\,i} = l$ (m), and is equal

$$\rho *= \eta/[(k^2 - \omega^2)^2 + 4n^2\varpi^2]^{\frac{1}{2}} \tag{71}$$

where: ω - frequency of the disturbing force (1/s); η - generalized constructive parameter, which, with the accepted U_r=l, depends on the mass of the machine and the moment of inertia of it relative to the center of gravity

$$\eta = \frac{l}{\left(J_c + \frac{G^O}{\mathrm{g}} \cdot l_2\right)} \tag{72}$$

Analysis of the dependencies (71) and (72) shows that to reduce the displacement, it is necessary to increase either the mass of the machine (to load with ballast) or the moment of inertia of it relative to the center of gravity. At l> 1,9 m, the constructive parameter decreases more intensively with increasing G^0 than with increasing J_c but at l<0,8 - otherwise. Consequently, the displacement value ρ * also changes accordingly. The displacement decreases with an increase in the damping coefficient and a larger difference between the circular frequency of the natural oscillations of the system and the frequency of external perturbations.

The effect of the values of the parameters of the scheme of flat-top machines on the position of the center of gravity, the magnitude of the moment of inertia, and the frequency of natural oscillations is investigated [26]. It is established that changing the parameters of the plane cutter l_k and l_n (see Fig. 49) has a significant effect on the position of the center of gravity in the longitudinal-vertical plane and its moment of inertia, but it does not significantly affect the mass and frequency of natural oscillations.

A comparison of the frequencies obtained for the natural vibrations of the plane with the frequencies of external perturbations shows that, depending on the speed of its motion and the parameters of the circuit, a resonant phenomenon may arise. To eliminate it, it is necessary to increase the frequency of the natural oscillations of the plane cutters and to reduce the frequency of external disturbances.

In laboratory studies carried out with the help of a special installation with various parameters of the planar plane scheme, for each variant of which are determined J_n, $\frac{G^0}{g}$, k, η. Their analysis showed that the deviations of the section, as a function of the values of the parameters and the frequency of external disturbances, are analogous to those calculated from (70), (71), and (72), (Figures 50, 51).

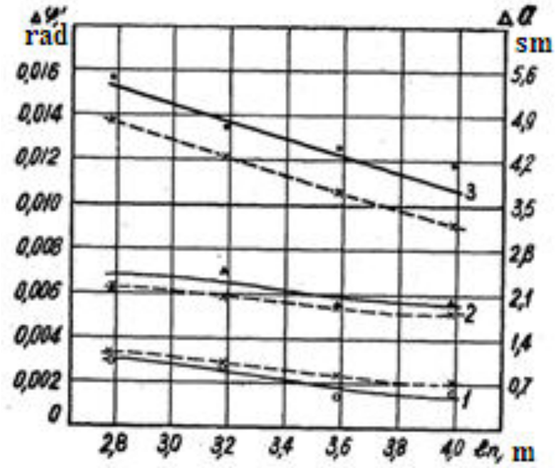


Fig. 50. Calculated (- - -) and experimental (-) changes$\Delta\varphi'$ and Δa depending on the size of the circuit parameters $l_n$and $l_k$at external perturbation from the unevenness of fields (1, 2, 3- respectively
l_k- l_n = +0,29; -0,21; -0,71 m)

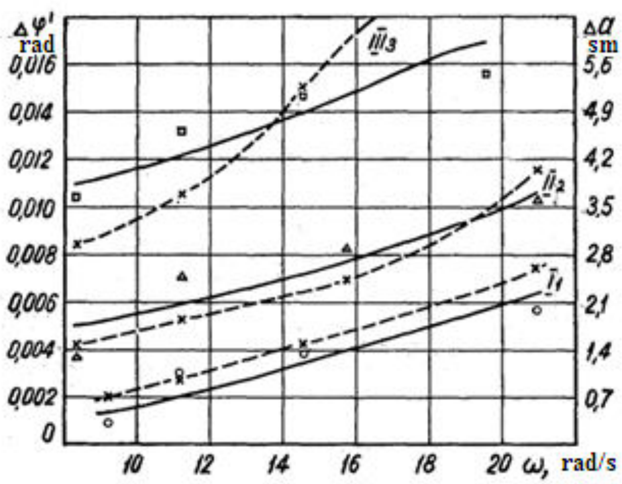


Fig. 51. Calculated (- - -) and experimental (-) changes $\Delta\varphi'$ and Δa depending on the frequency of external perturbations of the unevenness of fields and the parameters of the circuit l_n and l_k

I - l_n= 2,78 m; l_k = 3,07 m;
II - l_n= 3,18 m; l_k = 2,87 m;
III - l_n= 3,58 m; l_k= 2,97 m.

Laboratory studies have established that the deviation of the section under external disturbance from the heterogeneity of soils is almost 2-3 times less in hardness than in the unevenness of the fields. Therefore, in the field conditions, the effect of the values of the circuit parameters on the deviation of the section was studied along the channel by the profile of the surface of the field-the profile of the bottom of the furrow, i.e. $Z_{h_i}(t) \rightarrow Z_6(t)$. From the analysis of the nature of the changes in the correlation functions and the spectral densities of these processes and their numerical statistical characteristics obtained under real operating conditions, it follows that the trailing

section of the plane cut is a dynamical system with a narrow frequency bandwidth that enhances the low-frequency components at $l_n \approx l_k$.
The results of field experimental studies have shown that the change in the parameters of the circuit does not have a significant effect on the processes of the vertical and horizontal components of the paw resistance. However, the magnitude of the circuit parameters has a large effect on the response coefficient of the reaction on the support wheel N_k, and, consequently, on the stability of the course of working organs in depth, which, when $l_n \approx l_k$and l_n>3,5 m is in the allowable zone (Fig. 52).
On the channel $Z_{h_1}(t) \rightarrow Z_\delta(t)$, the correlation coefficient between them reached 0,4, for each installation variant, the amplitude-frequency characteristics of the dependence$S_{xy} = W(i\omega) \cdot S_x(i\omega)$, where$S_x(\omega)$ and $S_{xy} = (\omega)$ respectively, the spectral density of the process and the mutual spectral density of the input and output processes, $W(i\omega)$ - frequency function of the system. Based on these amplitude-frequency characteristics, the values of time coefficients T_1 and T_2 , characterizing the dynamic properties of the trailing section of the plane cutter are found (T_1 - inertial, T_2 - damping). An analysis of these coefficients shows that if l_k/l_n> 1, then $T_2/2T_1$> 1, i.e. section of the plane cutter works like an aperiodic link, and when l_n/l_k<1 the ratio $T_2/2T_1$<*1* - in this case the system works as an oscillating link.

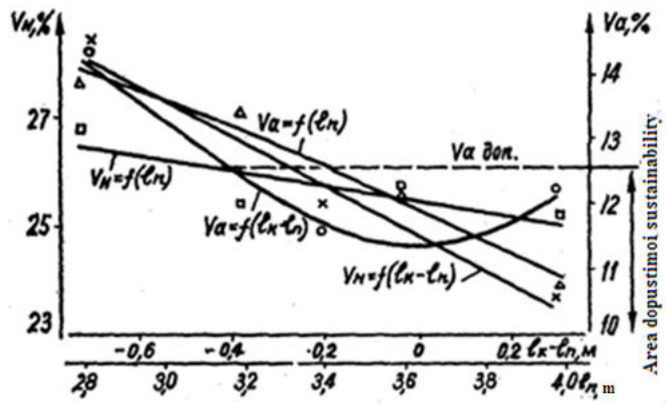
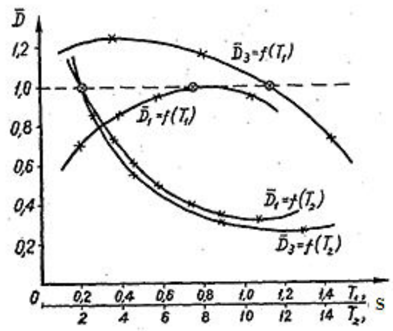


Fig. 52. The change in the vertical reaction on the support wheels and the depth of loosening of the soil, depending on the values of the parameters plane cutter scheme $l_n \approx l_k$

To determine the influence of the values of the time coefficients of the transfer function $T_1$and $T_2$to change the dispersion of the output process, an analysis of the work of the model of the section of the plane cutter on the ABM was carried out. The results of these studies (Fig. 53) showed that the coefficient T_2has a greater effect on the variance of the output process and its numerical value for v = 2,1-2,4 m/s should lie within 8 - 12 s, and the magnitude T_1chosen out of the working condition of the tool as an aperiodic link.



⊕ the experimental values
x the values obtained from AVM

Fig. 53. Dispersion of the output process (bottom profile of the furrow)depending on the values of the time coefficients $\boldsymbol{T_1}$ ***and*** $\boldsymbol{T_2}$

The results of these studies made it possible to recommend for the trailing section of the plane the optimal values of the distance from the point of the trailer to the working bodies $l_n \approx$ 3,5-4,0 m, mutual arrangement of support wheels of working bodies $l_k/l_n \approx 1$, position of the center of gravity relative to the axis of the support wheels (in front of the support wheels not more than 0,2-0,3 m), the moment of inertia (3500 - 4000 N · m · s^2), as well as the mass per 1 m of the tool (270-320 kg), which ensure the permissible stability of the stroke of working organs in depth.
The analysis of existing designs of domestic and foreign samples of wide-spread soil-cultivating machines for continuous tillage showed that one of the promising is the unchainable hinge-sectional scheme. In this case, it is advisable to transfer the machine to the position of long-distance transport by planting the side sections relatively to the middle back. In this case, the unit's height and dynamic loads on the middle section are reduced, the structure is simplified, and the safety is enhanced when transporting the machine over long distances.
However, wide-section hinged sectional cultivators-flat cutters, made on the frontal pattern, do not provide a stable course of working organs at a given depth of processing. In such machines, with an increased hardness of the soil, an irreversible phenomenon of "fracture" occurs in the areas of articulation between sections (Fig. 54).
Therefore, there was a need to study the dynamics of a wide-grip, unshaped, hinge-sectional cultivator-plane cutter and to select its optimal parameters ensuring a stable course of its working organs at a given processing depth.

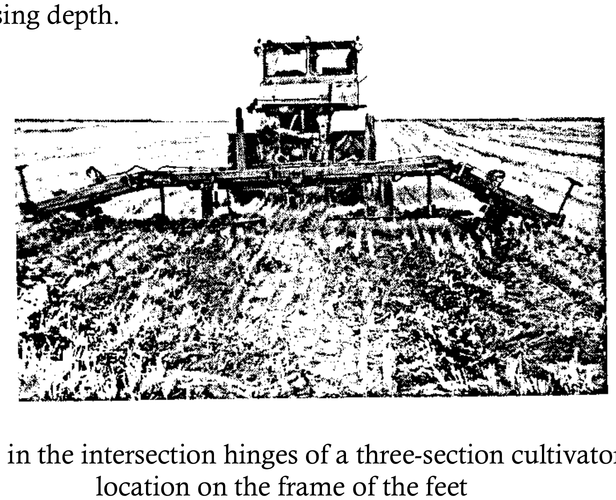


Fig. 54. A "break" in the intersection hinges of a three-section cultivator-plane with a front location on the frame of the feet

For this purpose, problems of ensuring the stability of motion in the longitudinal-vertical plane of machines with a small working width and a rigid one-section frame were considered, and, as a rule, these studies were carried out using static methods. Only recently there have been works on justifying the parameters of the hinge-sectional tillage tools. In them, soil-cutting flat-saw machines were considered as non-conservative systems for which the question of Lyapunov instability does not arise, since in such systems the accumulated energy is dissipated mainly due to the damping properties of the soil.

However, during operation, the operating elements rigidly fixed to the frame of the sections can change their position in the longitudinal and transverse-vertical planes. In this regard, the forces arising on them can introduce additional energy into the dynamic system, which is clearly insufficient to dissipate the damping properties of the soil. The account of such forces can lead to a qualitatively new conclusion about the operation of flat-topped sectional machines, since these forces can create instability. Therefore, the results of previous studies do not allow to fully justify choosing the optimal parameters of a hinged sectional cultivator-plane cutter.

The dynamics of a wide-angle hinge-section cultivator-plane moving along the field, which is inhomogeneous both in the hardness of the soil and in the microrelief, is considered in [8, 38]. Therefore, most of the forces acting on the machine during operation vary in magnitude and direction, thus causing deviations of the working bodies from the specified depth of loosening. Such forces include the components of the resultant forces of soil resistance to the working bodies and the rolling of the support wheels of the cultivator-plane along the unevenness of the field. In this case the cultivator-plane is considered as a three-section system in which the soil resistance forces to each working organ are concentrated in the reduced centers of their resistance with the coordinates l_{pix}, l_{piy}(Fig. 55).

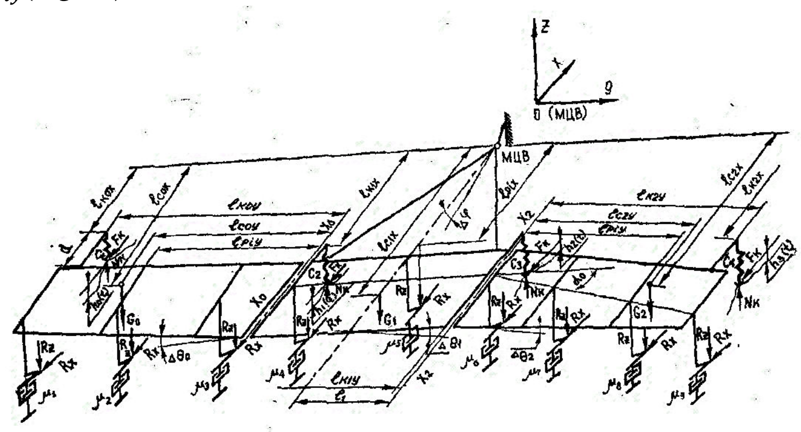


Figure 55 - Calculation scheme of the hinge-sectional flat cutter with three sections

In this case, the following assumptions are accepted: the soil layer is pushed to the machine along the OX axis with a constant speed of the gun υ_n and is cut by the working bodies to a given depth a, which is limited by support wheels rolling over the uneven surface of the formation; the deflection of the working elements, and consequently of the rigidly connected frames of the machine sections relative to the steady-state position (φ = 90° and θ = 90$^{0)}$) remain sufficiently small, that is, the linearization of the nonlinear dependences is permissible; the rigidity of the tires of the support wheels has a practically linear characteristic and during operation they do not detach from the field surface and, finally, the influence of the oscillations of the tractor on the point of the trailer of the machine is insignificant. The considered system in the adopted coordinate system XYZ will perform forced oscillations with respect to the MCW. The movement of the cultivator-plane is characterized by four generalized coordinates (see Fig. 55):

$\Delta\varphi$ - angular movements of sections of the machine in the longitudinal-vertical plane relative to the axis OY;

$\Delta\theta_1$ - angular movements of the central section of the machine in the transverse-vertical plane with respect to the axis passing through its center of gravity and the MCW;

$\Delta\theta_0$and $\Delta\theta_2$are the angular displacements of the left and right lateral sections of the flat-blade cultivator in the transverse-vertical plane cutter relative to the axes passing through the centers of the longitudinal joints connecting these sections to the central one. The origin of the coordinate system is connected with the MCW of the machine - point 0. The dynamics of such a system is most fully described by means of the Lagrange equation of the second kind.
Taking into account the values of the functions T, Π, Φ and Q_{di}, obtained using known methods, the Lagrange equations take the form:

$$\left.\begin{aligned}
&J_y\,\Delta\ddot{\varphi} + b_{10}\Delta\dot{\varphi} + c_{10}^*\Delta\varphi + b_{11}\Delta\dot{\theta}_0 + c_{11}\Delta\theta_{0+} b_{12}\Delta\theta_1 + c_{12}\Delta\dot{\theta}_1 + \\
&+b_{13}\Delta\dot{\theta}_2 + c_{13}\Delta\theta_2 = F_1(t); \\
&J_{ox_o}\ \ \Delta\ddot{\theta}_0 + b_{20}\Delta\dot{\theta}_1 + c_{20}^*\Delta\theta_{0+} + b_{11}\Delta\dot{\varphi} + c_{11}\Delta\varphi + b_{21}\Delta\dot{\theta}_1 + \\
&\qquad +c_{21}\Delta\theta_{12} = F_2(t); \\
&J_c^*\Delta\ddot{\theta}_1 + b_{30}\Delta\dot{\theta}_1 + c_{30}\Delta\theta_1 + b_{12}\Delta\dot{\varphi} + c_{12}\Delta\varphi + b_{32}\Delta\theta_0 + \\
&+c_{32}\Delta\theta_0 + b_{33}\Delta\dot{\theta}_2 + c_{33}\Delta\theta_2 = F_3(t); \\
&J_{2x_2}\ \ \Delta\ddot{\theta}_2 + b_{40}\Delta\dot{\theta}_2 + c_{40}^*\Delta\theta_{2+} + b_{13}\Delta\dot{\varphi} + c_{13}\Delta\varphi + b_{41}\Delta\theta_1 + \\
&+c_{33}\Delta\theta_1 = F_4(t),
\end{aligned}\right\} \tag{73}$$

where:

$$b_{10} = \sum_{i=1}^{9} \mu \cdot l_{pix}^2 \qquad b_{11} = \sum_{i=7}^{3} \mu \cdot l_{pix} \cdot l_{piy}$$

$$b_{12} = l_1 \sum_{i=6}^{9} \mu \cdot l_{pix} - l_1 \sum_{i=1}^{4} \mu \cdot l_{pix} \qquad b_{13} = \sum_{i=7}^{9} \mu \cdot l_{pix} \cdot l_{piy}$$

$$b_{20} = \sum_{i=1}^{3} \mu \cdot l_{piy}^2 \qquad b_{21} = -l_1 \sum_{i=1}^{3} \mu \cdot l_{piy}$$

$$b_{30} = l_1^2 \sum_{i=1}^{4} \mu + l_1^2 \sum_{i=6}^{9} \mu \qquad b_{32} = -l_1 \sum_{i=1}^{3} \mu \cdot l_{piy}$$

$$b_{33} = l_1 \sum_{i=7}^{9} \mu \cdot l_{piy} \qquad b_{40} = \sum_{i=1}^{3} \mu \cdot l_{piy}$$

$$c_{10} = \sum_{i=1}^{4} c \cdot l_{k1x} \qquad c_{11} = c \cdot l_{kox} \cdot l_{koy}$$

$$c_{10}^* = c_{10} - A_0 \qquad c_{12} = c \cdot l_1 (l_{k2x} - l_{kox})$$

$$c_{13} = c \cdot l_{k2y} \qquad c_{20} = c \cdot l_{koy}^2$$

$$c_{20}^* = c_{20} - A_1 \qquad c_{21} = c \cdot l_1 \cdot l_{koy}$$

$$c_{30} = l_1^2 \sum_{i=1}^{4} c_1 \qquad c_{30}^* = c_{30} - A_2$$

$$c_{31} = c \cdot l_1 (l_{k2x} - l_{k1x}) \qquad c_{32} = c \cdot l_1 \cdot l_{koy}$$

$c_{33} = c \cdot l_1 \cdot l_{k2y}$

$c_{40}^* = c_{40} - A_3$ $\qquad$ $c_{40} = c \cdot l_{k2y}^2$

$$A_0 = \sum_{i=1}^{9} K_{0pi} \cdot l_{pix} \qquad A_1 = \sum_{i=1}^{3} K_{1pi} \cdot l_{piy}$$

$$A_2 = \sum_{i=4}^{6} K_{1pi} \cdot l_{piy} \qquad A_3 = \sum_{i=7}^{9} K_{1pi} \cdot l_{piy}$$

$$F_1(t) = c[l_{koo}h_1(t) + l_{k1x}h_2(t) + l_{k1x}h_3(t) + l_{k2x}h_4(t)] - \sum_{i=1}^{9} \Delta R_{zit} \cdot l_{pix};$$

$$F_2(t) = c \cdot l_{koy}h_1(t) - \sum_{i=1}^{3} \Delta R_{zit} \cdot l_{piy};$$

$$F_3(t) = c[l_1 h_4(t) + l_1 h_3(t) - l_1 h_2(t) - l_1 h_1(t)] + \Delta R_{z4t} \cdot l_{p4y} - \Delta R_{z6t} \cdot l_{p6y};$$

$$F_4(t) = c \cdot l_{k2y}h_4(t) - \sum_{i=7}^{9} \Delta R_{zit} \cdot l_{piy};$$

where: J_y ,$J_{ox_1}J_{2x_2}$, J_c^*- moments of inertia, respectively of the whole machine relative to the axis *OY*lateral sections relative to the axes x_o—x_o and x_2 —x_2, the given moment of inertia of the machine with respect to the axis passing through the CT and the *MCW*, $N \cdot m \cdot s^2$; b_{ij} – coefficients that take into account the dissipative forces of the system, $N \cdot m \cdot s$; c_{ij} – coefficients that take into account the rigidity of the system, $N \cdot m$; A_i - the forces acting on the working parts of the cultivator-plane cutter, which appear when the sections deviate from the equilibrium state, $N \cdot m$; $F_i(t)$- generalized forces acting on the cultivator-plane cutter, $N \cdot m$.
The solution and analysis of the system of differential equations (73) presents a significant difficulty. To simplify its solution, the following assumptions are made.
1. Under the support wheels of the middle section, the perturbations are the same in amplitude and phase, that is, the machine moves across the previous treatment. Therefore, the depth of loosening 4 and 6 of working bodies will be equal. Assuming that ΔR_{zit}it is mainly a function of the change in the depth of loosening, $\Delta R_{z4t} = \Delta R_{z6t}$.Therefore, $\Delta\theta_1 = 0$.
2. Since the side sections are identical and are relatively symmetric about the mean, in the system (73) the corresponding coefficients of the second and fourth differential equations and their non-homogeneous parts $F_1(t)$ and $F_4(t)$are equal to each other, and the functions $\Delta\theta_0(t)$ and $\Delta\theta_2(t)$satisfy the same differential equation. In addition, the initial conditions are zero, hence, for t≥ 0 is a true condition $\Delta\theta_0(t) = \Delta\theta_2(t)$.The identity of these functions makes it possible to replace in the first equation $\Delta\theta_0(t)$ by $\Delta\theta_2(t)$ when solving system (73) and take into account only one of the two equations, for example, the fourth one.
In view of the foregoing, system (73) simplifies and takes the form

$$\left.\begin{aligned} J_y\, \Delta\ddot{\varphi} + b_{10}\Delta\dot{\varphi} + c_{10}^*\Delta\varphi + b_{14}\Delta\dot{\theta}_2 + c_{14}\Delta\theta_2 = F_1(t); \\ J_{2x_2}\ \ \Delta\ddot{\theta}_2 + b_{40}\Delta\dot{\theta}_2 + c_{40}^*\Delta\theta_2 + b_{13}\Delta\dot{\varphi} + c_{13}\Delta\varphi = F_4(t) \end{aligned}\right\} , \qquad (74)$$

where:$b_{14} = b_{11} + b_{13}$; $\quad c_{14} = c_{11} + c_{13}$.

From this system of differential equations, it can be seen that all the oscillations of the dynamical system under consideration are related, that is, the generalized coordinates ($\Delta\varphi$ and $\Delta\theta_2$) will vary simultaneously. This relationship can lead to instability, that is, system (74) for certain values of its parameters will become inoperative and its oscillation amplitudes will increase with time. In connection with this, it is necessary to investigate the given system for stability, which can be estimated using the Routh-Hurwitz criteria, which for the system under consideration have the form:

$$a_1 - a_5 > 0$$

$$\Delta_1 = a_2 a_3 - a_1 a_4 > 0;$$

$$\Delta_2 = a_2 a_3 a_4 - a_2^2 a_5 - a_1 a_4^2 > 0; \quad (75);$$

$$\Delta_3 = a_2 a_3 a_4 a_5 - a_2^2 a_5^2 - a_1 a_4^2 a_5 > 0;$$

where:

$$a_1 = J_y \cdot J_{2x_2};$$

$$a_2 = J_y b_{40} + J_{2x_2} b_{10};$$

$$a_3 = J_y c_{40}^* + b_{10} b_{40} + J_{2x_2} c_{10}^* - b_{13} b_{14};$$

$$a_4 = b_{10} c_{40}^* + b_{40} c_{10}^* - b_{13} c_{14} - b_{14} c_{13};$$

$$a_5 = c_{10}^* c_{40}^* - c_{13} c_{14}.$$

Criteria allow in general to assess the stability of a dynamic system for various values of its parameters and determine the range of their admissible values. However, this does not mean that this dynamical system, provided that conditions (75) are met, will have the necessary stability of the stroke of the working organs in depth.

In the study of dynamic systems, amplitude-frequency characteristics (AFC) have the greatest practical significance. To find them, we represent the system of differential equations (74) in Laplace images, that is,

$$\left.\begin{array}{l} \left[L_y S^2 + b_{10} S + (c_{10} - A_0)\right] \cdot \Delta\varphi + [b_{14} S + c_{14}] \cdot \Delta\theta_2 = F_1 S \\ \left[J_{2x_2} S^2 + b_{40} S + (c_{40} - A_3)\right] \Delta\theta_2 + [b_{13} S + c_{13}] \cdot \Delta\varphi = F_4 S \end{array}\right\} \quad (76)$$

Then

$$\left.\begin{array}{l} \Delta\varphi(S) = W'_{\Delta\varphi}(S) \cdot F_1\, S - W''_{\Delta\varphi}(S) \cdot F_4(S) \cdot e^{-s\tau} \\ \Delta\theta_2(S) = W'_{\Delta\theta_2}(S) \cdot F_4\,(S) \cdot e^{-s\tau} - W''_{\Delta\theta_2}(S) \cdot F_1\,(S) \end{array}\right\}, \quad (77)$$

where $W'_{\Delta\varphi}(S)$; $W''_{\Delta\varphi}(S)$; $W'_{\Delta\theta_2}(S)$; $W''_{\Delta\theta_2}(S)$- operators of the system:

$$\left.\begin{array}{l} W'_{\Delta\varphi}(S) = \dfrac{I_{2x_2} S^2 + b_{40} S + (c_{40} - A_3)}{a_1 S^4 + a_2 S^3 + a_3 S^2 + a_4 S + a_5}; \\[2ex] W''_{\Delta\varphi}(S) = \dfrac{b_{14} S + c_{14}}{a_1 s^4 + a_2 S^3 + a_3 S^2 + a_4 S + a_5}; \\[2ex] W'_{\Delta\theta_2}(S) = \dfrac{J_y S^2 + b_{10} S + (c_{10} - A_0)}{a_1 S^4 + a_2 S^3 + a_3 S^2 + a_4 S + a_5}; \\[2ex] W''_{\Delta\theta_2}(S) = \dfrac{b_{13} s + c_{13}}{a_1 S^4 + a_2 S^3 + a_3 S^2 + a_4 S + a_5}. \end{array}\right\} \quad (78)$$

Formulas (77) contain transcendental terms of the forme^{-sr}, which show that perturbations from the surface profile of the field arrive with a delay on the support wheels of the sections by an amount$\tau = L/v$, where L - is the distance between the support wheels of the lateral and middle sections in the direction of travel. Replacingin (77) s by$(i\omega)$ and taking into account that $F_1(i\omega) = F_4(i\omega) = 1$, we obtain the amplitude-frequency characteristics (AFC)

$$W_{\Delta\varphi}(i\omega) = \frac{a_*\left[(c_{40} - A_3) - J_{2x_2}\omega^2 - c_{14} \cdot cos\omega\tau - b_{14} \cdot \omega sin\omega\tau\right] +}{a_*^2 +}$$

$$\frac{+b_*\,(b_{40} \cdot \omega - b_{14} \cdot \omega cos\omega\tau + c_{14} sin\omega\tau}{+b_*^2} + i\frac{a_*(b_{40}\ \cdot \omega - b_{14} . cos\omega\tau + c_{14} \cdot sin\omega\tau) +}{a_*^2 +}$$

$$\frac{+b_*\left[I_{2x_2}.\omega^2 + c_{14} \cdot cos\omega\tau + b_{14} \cdot \omega\tau - (c_{40} - A_3)\right]}{+b_*^2} \qquad (79)$$

$$W_{\Delta\theta_2}(i\omega) = \frac{a_*\left[(c_{10} - A_0)cos\omega\tau - J_y\omega cos\omega\tau + b_{10}\omega sin\omega\tau - c_{13}\right] +}{a_*^2 +}$$
$$\frac{+b_*\left[b_{10} \cdot \omega cos\omega\tau - (c_{10} - A_0)sin\omega\tau - J_y\omega^2 \cdot sin\omega\tau - b_{13} \cdot \omega\right]}{+b_*^2} +$$
$$+i\frac{a_*\left[b_{10} \cdot \omega cos\omega\tau - (c_{10} - A_0)sin\omega\tau - J_y\omega^2 \cdot sin\omega\tau - b_{13} \cdot \omega\ \ \right] +}{a_*^2 +}$$

$$\frac{+b_*\left[c_{13} - (c_{10} - A_0)cos\omega\tau + J_y\omega \cdot cos\omega\tau - b_{10}\omega sin\omega\tau\right]}{+b_*^2}, \qquad (80)$$

where:$a_* = \alpha_1\omega^4 - \alpha_3\omega^2 + \alpha_5;\ b_* = a_4\omega - a_2\omega^3$,
Denoting the real and imaginary parts of the frequency response in (79) and (80), respectively $R_{e\Delta\varphi}(\omega)$; $R_{e\Delta\theta_2}(\omega)$; $I_{m\Delta\varphi}(\omega)$; $I_{m\Delta\theta_2}(\omega)$, we obtain the dependences for determining the amplitudes of the vertical displacements of the working elements as a function of the frequency of external disturbances:

— for the working parts of the middle section

$$A_{ic}(\omega)=\left\{\left[l_{pix}R_{e\Delta\varphi}(\omega)\right]^2 + \left[l_{pix} \cdot J_{m\Delta\varphi}(\omega)\right]^2\right\}^{\frac{1}{2}},\ i = 4,5,6 \qquad (81)$$

- for the working parts of the side sections

$$A_{i\delta}(\omega = \left\{\left[l_{pix}R_{e\Delta\varphi}(\omega) + l_{piy}R_{e\Delta\theta_2}(\omega)\right]^2 + \left[l_{piy}J_{m\Delta\theta_2}(\omega) + l_{pix}J_{m\Delta\varphi}(\omega)\right]^2\right\}^{\frac{1}{2}} \qquad (81)$$
$$i = 1,2,3,7,8,9.$$

Dependencies (81) and (82) make it possible to estimate and select ranges of rational values for the parameters of hinge-sectional tillers with known numerical characteristics of harmonic external disturbances. It is advisable to refine the found values with allowance for real input disturbances by simulating the AVM system (74) and verify them in the real operating conditions of the machine. Therefore, before carrying out field studies, profilograms of surface irregularities and solidograms of the heterogeneity of the soil layer in terms of hardness were recorded. Statistical characteristics and these processes were by implementing well-known algorithms on a computer.

For experimental studies, a laboratory-field installation with varying parameters of the circuit was designed and manufactured (see Fig. 54): a^0 = +35^0; 0; -35^0; $l_{koy} = l_{k2y}$=1,05-2, 65m; d = 0,5-1,5m [8,38].

The coordinates of the center of gravity of the installation were determined by the method of suspension, and the moments of inertia by the method of pendulum oscillations. Determination of the coefficient characterizing the resistance of the soil to the flat-topped paw when it was deepened or buried was carried out in the soil channel by a specially developed method, the essence of which was as follows. In the process of work, a certain speed of movement of the working element in the vertical direction was set and vertical components of the total soil resistance were recorded at the same time. One way to obtain a given speed of movement, rigidly connected to the frame of a flat-topped paw, is to hit absolutely rigid support wheels onto absolutely rigid irregularities, for example having the shape of a trapezoid. By varying the parameters of the trapezoid and the translational speed of the machine, it is possible to obtain a predetermined rate of displacement of the paw when it is recessed or buried. Recorded on the oscillogram, the process of burial or burial of the paw was divided into equal time intervals Δt = 0,05 s. During this period of time, the path of the deepening (deepening) is determined by the difference in the ordinates of the process of changing the depth of the working organ at the beginning and at the end of the accepted time interval. According to the known path and time, the speed of moving the paw in the process of penetration (deeper penetration) is determined. At each of these time intervals, the ordinates of the vertical component of the soil resistance of the flat-topped paw, as a function of the vertical speed of its movement when receding or buried, were taken, respectively, with a change in the speed of movement.

Analysis of the dependences characterizing the change in soil resistance of a flat-topped paw as a function of the speed of its movement during burial or burial (Fig. 56) shows that with certain assumptions the form of this dependence can be adopted linearly viscous, and the numerical value of the coefficient μ is equal to 20 kN s/m.

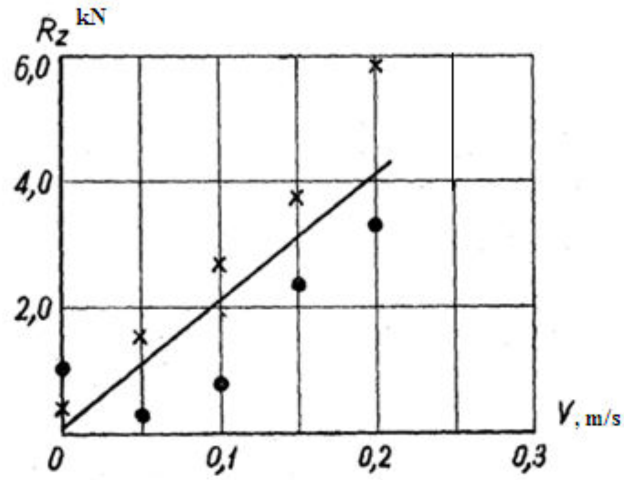


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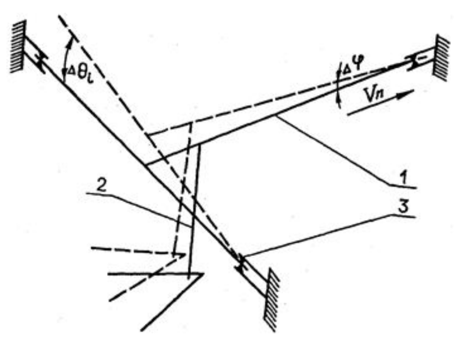
Fig. 56 Resistance of the soil to a flat-topped paw, depending on the speed of its movement in the process of penetration or deepening (the depth of loosening is 12-14 sm)

The determination of the vertical component on the working organ as a function of the angle of its inclination $\Delta\varphi$ was $\Delta\theta_2$ carried out in a soil channel in a special installation (Fig. 57). The angles $\Delta\varphi$ and $\Delta\theta_2$ varied within the range 0 - 6° with an interval of 2°. The process of change R_z was recorded with the help of strain gauges.



1 - spatial strain gage; 2 - working body; 3 - strain gauges

Fig. 57. Scheme of installation for determining the vertical component of soil resistance to the working body, depending on its position in space

Analysis of experimental dependencies $R_{zi} = f(\Delta\varphi); \Delta\theta_2$)shows (Fig. 58 *a, b*) that with an increase in the angle of inclination of the installation frames in the longitudinal and transverse-vertical planes, the vertical component of the total resistance of the soil $R_{zi} = f(\Delta\varphi)$; $\Delta\theta_2$increases in direct proportion. At the same time, an increase in the angle of inclination of the lateral sections in the transverse-vertical plane $\Delta\theta_2$has a greater effect on the increase and intensity of growth R_{zi} than with changing the angle $\Delta\varphi$.

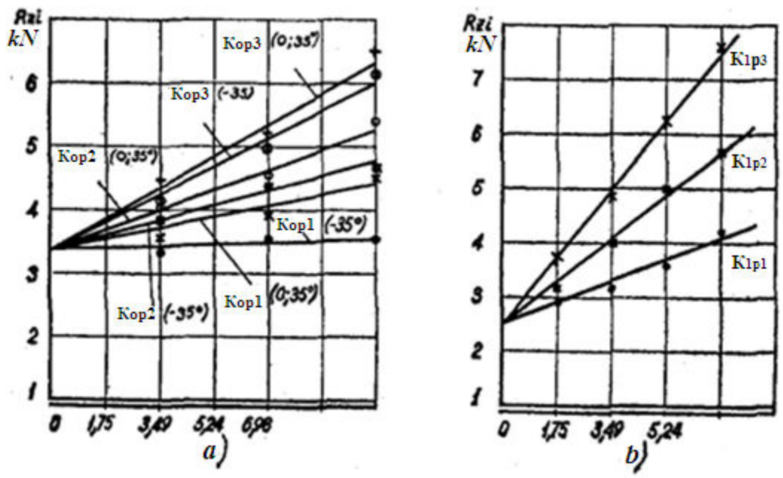


Fig. 58 Dependences of the vertical components of the soil resistance of the flat-topped paw ($k_{op4}=k_{op6}=k_{op8}$; $k_{op5}=k_{op7}=k_{op9}$) as a function of the angles $\Delta\varphi$ and, $\Delta\theta_2$ respectively, a) and b)

Taking into account the experimentally found values of the coefficients (μ, K_{opi}) and the parameters of the twenty-seven variants of the setup, the Routh-Hurwitz criteria were found. Their analysis showed that of all the variants of various combinations of the parameters of the setup circuit, the stability conditions are satisfied only for eighteen variants (1-6,10-15, 19-24), for which l_{koy}=$l_{k2y} \geq l_{coy} = l_{c2y}$(see Fig. 55) .

In these cases, the recovery moments from the support wheels of the side sections will be greater than the moments caused by the forces appearing on the working bodies, while in the other variants, on the contrary. Therefore, the lateral sections' working members in the latter variants will be monotonously buried and, acting through the intersection hinges, "squeeze out" the middle section of the cultivator-plane upwards, i.e., there will be a "fracture" of the sectional frame, which was confirmed by field tests of these variants (see Fig. 54).

For the sixth and ninth working bodies of all installation options for which the stability criteria (75) are fulfilled, the calculated AFCs were obtained from formulas (81) and (82), the analysis of which allowed to reveal the influence of one or another parameter of the circuit on the deviation of the working elements from the equilibrium position. As an example, Fig. 59 shows the calculated AFC response for 1, 4, 10 and 19 installation options.

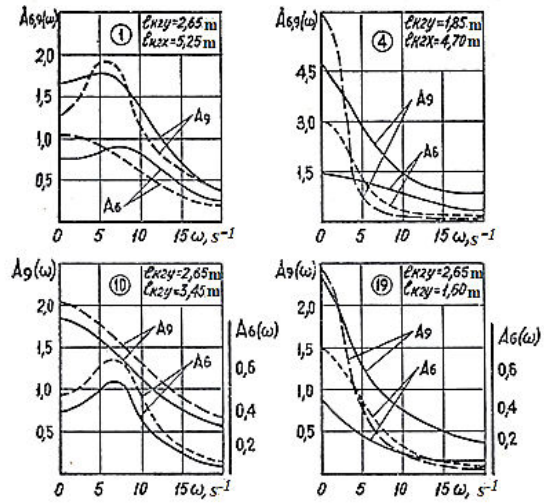


Fig. 59 Theoretical (-) and experimental (- - -) amplitude-frequency characteristics of the hinge-section cultivator-plane cutter, depending on its parameters from the c circuit (1,4,10 and 19 variants of the installation)

The maximum amplitude-frequency characteristic of the sixth working organ along the nel$h_3(t) \rightarrow Z_{Б6}(t)$is much less than the ninth organ working through the channel $h_4(t) \rightarrow Z_{Б9}(t)$. This indicates that, in comparison with the lateral, the middle section performs angular vibrations with a smaller amplitude, since its moment of inertia relative to the axis$OУ$ is in 2 - 3 times more than the moments of inertia of the lateral sections performing angular vibrations in the transverse-vertical plane relative to the axes of the intersection hinges x_0—x_0 and x_2—x_2. In addition, the damping properties of the middle section are enhanced by the side sections.
Using the calculated frequency response of the sixth and ninth operating bodies, the values of the time coefficients of the transfer functions were calculated, the analysis of which showed that the hinged sectional plane, cutter depending on the parameters of the circuit l_k/l_n, processes input effects as an oscillatory $\left(\frac{l_k}{l_n} = p = \mathrm{T}_2/2\mathrm{T}_1 < 1\right)$or an aperiodic ($p > 1$) link. To improve the quality of planing, it is necessary that the internal structure of the process $Z_{Бi}(t)$on the spectral composition in the low-frequency region approached the structure of the input process $h(\mathrm{t})$. At the same time, the amplification amplitudes of the displacements are desirable to have <1, since the plane plane must fulfill the role of the "scheduler". These conditions are most fully met by the hinged sectional plane, in which the side sections are displaced relatively to the average in the horizontal plane backwards ($\alpha° = 35°$), and their support wheels are as far removed from the intersection hinge axes$l_{koy} = l_{k2y}$ max.) and have a minimum outreach in the direction of movement relative to the working bodies, i.e. $l_{kix} \approx l_{pix}$.
Similar results were obtained in the simulation of the system (74) of the operation of the hinged sectional plane along the minimum of the dispersion of the sixth and ninth working bodies on the AVM, taking into account the real characteristics of the input disturbances. Dispersions of the depth of stroke of these working organs, operating under the most unfavorable conditions, can be obtained as

$$D_{a6} = \frac{1}{T}\int [a_6(t)]^2 \cdot dt$$
$$D_{a9} = \frac{1}{T}\int [a_9(t)]^2 \cdot dt. \qquad (83)$$

Discrete values of processes $a_6(t)$и$a_9(t)$

$$a_6(t) = Z_{Б.6}(t) - h_3(t); \qquad (84)$$

$$a_9(t) = Z_{Б.9}(t) - h_4(t),$$

where: $Z_{Б.6}(t)$, $Z_{Б.9}(t) -$ furrow bottom profiles for the sixth and ninth working bodies; $h_3(t), h_4(t) -$ field surface profiles entering the third and fourth support wheels.

Formation of a disturbing signal, analog of the process [h (t)], was carried out as follows. Parallel to the surface of the field, at a certain height, rails up to 60 m long were installed. Along them with$\upsilon_n = const$ rolled a rolling cart with a fixed rheochord, the wheel of which copied the relief of the field. The resulting electrical signal was applied to the magnetic tape. This analogue signal was then repeatedly fed to the input of the dynamic model. A signal analog of the process of changing the soil resistance [$R_z(t)$] was also recorded on a magnetic tape.

Drawing up the structural diagram of the dynamic model of the cultivator-plane cutter for the AVM study was carried out by the method of decreasing the order of the derivative.

As a result of the simulation of the system (74) on the AVM, it was established that the minimum of the variances of the vertical displacements of the working members of the hinged sectional plane cutter is observed under condition$l_{p6x} - l_{k1x} \approx$ 0,35 m for sixthworkingtool, for ninthtool $l_{p9x} - l_{k2x} \approx$0,25 m.

Field studies were conducted to verify the values of the parameters of a wide-angle hinged sectional cultivator-plane cut, selected on the basis of an analysis of the mathematical model of its functioning according to the Routh-Hurwitz criteria, AFC, the ratios of the time coefficients of the transfer function, and the results of the AVM studies. According to the obtained data, experimental frequency response curves for the sixth and ninth working organs have been constructed, the nature of their variation and their numerical values close to the calculated ones (see Fig. 59). By these frequency response, the time coefficients of the transfer functions and their ratios$p = T_2/2T_1$.In this case, the experimental and calculated values of p unambiguously show for what set of numerical values of the parameters of the circuit the hinge-sectional plane cuts out the input disturbances as an oscillatory or as an aperiodic link.

Analysis of the statistical characteristics of the changes in the depth of stroke for the sixth and ninth working bodies obtained in the field showed that the variants of a wide-angle hinged sectional cultivator-plane with the recommended values of the circuit parameters provide the uniformity of the course of their working organs in depth determined by the agrotechnological (initial) requirements.

Determination of the stability of the stroke of working organs in the depth of narrowly-mounted hinged ones (KPN-4, KPSh-5, OPT-3-5,KPG-250A, PG-3-100, PG-3-5) and trailed (KPG-2,2) cultivators plane cutters and flat-plows-deep-loosening is not difficult, because the methodological basis for its calculation is described in detail in the literature [14].

The stability of the stroke of the disk working organs is symmetric to the stubble cleaner, provided [11,15] $R_z \approx 0{,}5 \cdot R_x$, determined by the formulas (40) and (41). However, in order to ensure the installed depth of the discs, the working weight of the asymmetrical discs should be twice the value R_z.

Influence of the parameters of needle batteries (inclination and length of the battery lead to the center of its gravity -l_0, weight - G, working width - Ш), speed of movement (v) and the height of the profile of the unevenness of fields characterized by dispersion - D and obtained by approximating the normalized correlation functions of the process of changing the irregularities by the coefficients a and β, the uniformity of the depth of loosening and other qualitative indices was considered in [16]. The following is established (Fig. 60):

- the initial (constructive) angle of inclination of the frame, varying from 0 to 45°, does not affect the performance of the battery;
- lengthening the battery's battery frame improves the uniformity of the depth of processing, but does not affect the quality of the profile alignment of the field surface;
- increase in the translational speed of the machine improves the quality of the equalization of the field irregularities by the battery, but worsens the stability of the processing depth and the safety of the stubble;
- with the increase in the width of the battery, the quality of its equalization with the unevenness of the field surface improves, but at the same time the uniformity of the depth of processing is dete-

riorated more intensively.

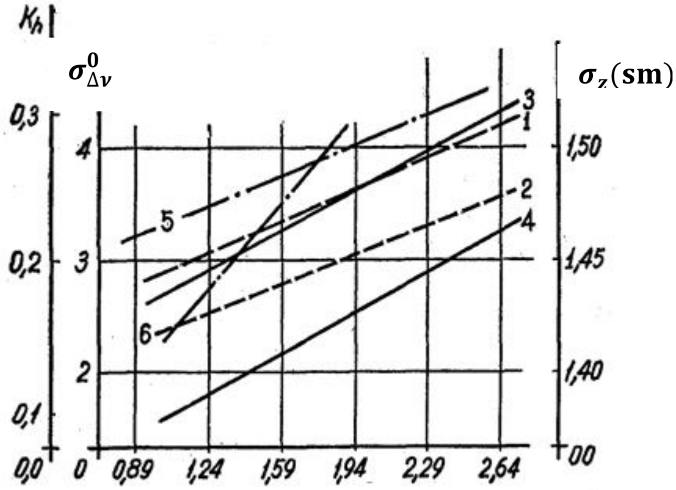


Fig.60. The dependence of the battery oscillations$\sigma^0_{\Delta v}$, the coefficient of decrease in the height of the unevenness of the field profile K_h and the uniformity of the depth of processing (σ_z)on the width of the battery: 1 and 2 - theoretical dependences ($\sigma^0_{\Delta v}$)respectively at a = 0,075 m and a = 0,0965 m; 3 and 4 - the coefficient (K_h) - respectively, in the presence of pressure springs and in their absence; 5 and 6 - uniformity of the depth of treatment (σ_z), respectively, in the presence and absence of pressure springs

Taking into account the most common statistical characteristics of stubble backgrounds, during the period of their processing with needle-shaped harrows, the width of the capture of batteries should be taken within 1,2 - 1,6 m.

2.5. Stability of the movement of soil-cultivating machines and tools of soil conservation in the horizontal plane

With the increase in the width of the grip of machines, the stability of their movement in the horizontal plane deteriorates. This is because the instant center of application of the reduced drag force of a wide-angle machine does not generally coincide with the traction line of the tractor due to soil heterogeneity in hardness and different depth of loosening along the width of its capture. In this case, a moment is created that tends to deflect the machine from rectilinear motion.
Taking into account the well-known assumptions made in the analysis of small oscillations of mechanical systems, the movement of the machine in the horizontal plane is described by a differential equation of the form (69). In this case, to determine the coefficients n and k^2, we proposed [39] the dependences:
for a trailer with symmetrical working elements, for example, lancet paws

$$n = d(\alpha_0 R \cdot d + \beta_0)/2J_0 \cdot v_0; \quad k^2 = (\alpha_0 R \cdot d + \beta_0)/2J_0 \tag{85}$$

but for an attached machine

$$n = D(\alpha_0 RD + \beta_0)/2J_{co} \cdot v_0; \quad k^2 = \frac{\alpha_0 RD + \beta_0}{J_{co}}, \tag{86}$$

where: d - the distance between the trailer's point and the center of application of the reduced resistance of the machine;R - the main vector of forces of resistance of working bodies;v_0 - forward speed of the machine; J_0 - the moment of inertia of the machine relative to the vertical axis passing through the trailer point; α_0, β_0- power parameters of the working body; D - distance from the instantaneous center of rotation of the machine to the center of application of the reduced force of resistance of the working bodies and is equal to $D = d + b\ cos\alpha + \frac{a}{tg\alpha}$; b - the length of the lower links of the tractor linkage; α - angle between the axis of symmetry of the tractor and the lower links of the tractor hitch mechanism; a- half the distance between the hinges of the lower links of the tractor hitch at the point of attachment to the tractor, J_{co}moment of inertia of the machine relative to the instantaneous center of rotation $J_{co} = J_0 + mH^2$; m - weight of machine; H- distance from the instantaneous center of rotation of the machine to its center of mass and is found from the relation$H = h + b\ cos\ \alpha + \frac{\alpha}{tg\alpha}$; h - distance between the axis of the suspension of the machine and the center of its mass.

It is seen from (85) and (86) that the coefficients n and k^2 of equation (69) are positive and constant for v_0 - const and R - const. Consequently, the motion of the machine for all values of these coefficients is asymptotically stable according to Lyapunov ($\Delta\varphi \to 0$ or $t \to \infty$). It is known that, depending on the ratio of these coefficients$\frac{n}{k} = \theta$, the nature of the machine's motion after obtaining initial perturbations $\Delta\varphi$and $\Delta\dot{\varphi}$ can be different: for $\theta < 1$, it is vibrational; $\theta = 1$- limit aperiodic; $\theta > 1$ - aperiodic. From the practical point of view, the degree of stability of the movement of the machine in the cases considered can not be considered the same. At different values of the ratio $\theta = n/k$, to achieve acceptable deviations, a different time is required, and the machine undergoes a different path during this time - S. The path for different values of the ratio $n/k$for the trailer is found from the dependences

$$S = \frac{2J_0 v_0^2}{(\alpha_0 Rd + \beta_0)d} \cdot lnN \qquad \text{with } \theta < 1;$$

$$S_{min} = v_0 lnN \sqrt{\frac{J_o}{\alpha_0 Rd + \beta_0}} \text{ with } \theta \to 1; \qquad (87)$$

but for an attached machine

$$S = \frac{2v_0^2(J_{co} + mH^2)}{(a_0 RD + \beta_0)D} \cdot lnN \qquad \text{with } \theta < 1;$$

$$S_{min} = v_0 lnN \sqrt{\frac{J_{co} + mH^2}{\alpha_0 RD + \beta_0}} \text{ with } \theta \to 1; \qquad (88)$$

where: N - is a number characterizing, under damped oscillations, a decrease in the amplitude after the i-th number of strokes.

For the case $\theta > 1$, the path is determined by the approximate formula

$$S = \frac{1}{2} dlnN\left(1 - \sqrt{1 - \frac{4J_0 v_0^2}{d^2(\alpha_0 Rd + \beta_0)}}\right). \qquad (89)$$

It follows from formulas (87), (88) and (89) that the minimum path - S_{min}, achieved at a given speed of the machine is proportional to the velocity and depends on the parameters of its circuit. The path S in the non-optimal mode of motion is proportional to the square of the velocity and also depends on the parameters of the circuit of the machine. To increase the stability of the machines in the horizontal plane, that is, to reduce the path S, it is necessary to strive to reduce the moment of inertia J_0, increase the length d and the resistance force R.

The angular deviations of the machine from the direction of motion for known n and k are determined by equations

$$\begin{aligned}&\text{with } \theta < 1: \ \Delta\varphi = e^{-nt}\left(c_1 cos\sqrt{k^2-n^2 t} + c_2 sin\sqrt{k^2-n^2 t};\right);\\ &\text{with } \theta \to 1: \ \Delta\varphi = e^{-nt}(c_1 + c_2 t);\\ &\text{with } \theta > 1: \ \Delta\varphi = e^{-nt}\left(c_1 e^{\sqrt{n^2-k^2 t}} + c_2 e^{\sqrt{n^2-k^2 t}}\right);\end{aligned} \tag{90}$$

where: $c_1 и c_2$ - constants of integration.

It is recommended that the parameters of the machine scheme be chosen under the condition $n \approx k$, that is, $\theta \approx 1$.
The numerical values of α_0 and β_0, which appear in formulas (85) and (86), are experimentally based on the following considerations. In a symmetrical planar paw, the system of soil reactions reduces to a single resultant force if it moves translationally in the plane of its axis of symmetry. If the velocity vector of the center of reduction of the forces of resistance to the motion of the flat-topped paw from the plane of its symmetry by an angle δ deviates, the system of forces acting on the paw is already reduced to the principal vector R and the principal moment M_0, and R deviates from the plane of symmetry by an angle χ (Fig. 61). The modulus of force R, the angle χ and the moment M_0 are functions of the angle δ characterizing the location of the velocity vector υ_0 with respect to the plane of symmetry of the plane cut paw as it rotates, i.e.

$$R = R(\delta); \quad \chi = \chi(\delta); \quad M_0 = M_0(\delta).$$

Expanding these functions in a series in powers of the small parameter δ, we obtain

$$R(\delta) = R(o) + R'(o)\delta + \frac{1}{2}R''(o)\delta^2 + \cdots$$
$$\chi(\delta) = \chi(o) + \chi'(o)\delta + \frac{1}{2}\chi''(o)\delta^2 + \cdots$$
$$M_0(\delta) = M_0(o) + M_0{}'(o)\delta + \frac{1}{2}M_o''(o)\delta^2 + \cdots$$

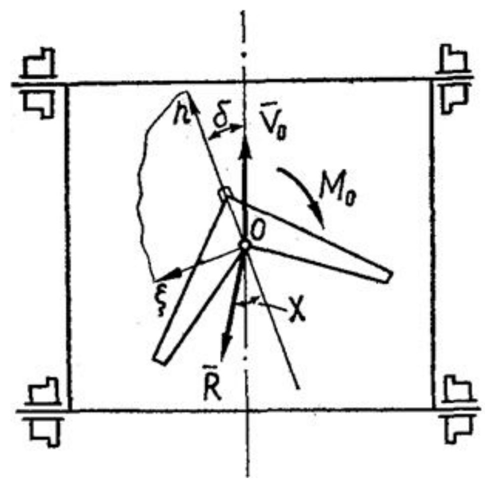


Fig. 61. Scheme of the installation for determining the force parameters α_0 and β_o of the planar paw

Since $R(o) = R_0$ - is a finite value, the remaining terms of this series, small first and higher orders, can be neglected. In the series for $\chi(\delta)$ and $M_0(\delta)$, there are no finite terms, i.e., $\chi(o)= 0$ and $M_0(o)= 0$, therefore we retain only terms of the first order of smallness in them. Then for small values of the angle $\delta R \approx R_0 = const; \chi \approx \alpha_0 \cdot \delta; \quad M_0 \approx \beta_0 \cdot \delta$, where $\alpha_0 = \chi'(o)$ and $\beta_0 = M'(o)$ are the constants for the given velocity (v_0) and the loosening depth (h).

Each of these coefficients is proportional to the tangent of the slope of the tangent to the corresponding curve [$\chi = \chi(\delta)$] or [$M_0 = M_0(\delta)$] at its initial point, that is,

$$\alpha_0^\delta = \left(\frac{d\chi}{d\delta}\right)_{\delta=o}; \quad \beta_0^\delta = \left(\frac{dM_0}{d\delta}\right)_{\delta=o}$$

Thus, to determine the force parameters of the plane cut foot α_0 and β_0 in the function of the angle δ, it is necessary to know the dependencies χ and M_0 in the function δ. However, χM_0 will depend not only on the angle δ, but also on the speed of movement (v_0)) and the depth of loosening (h). In this case, the force parameters of the planar paw are similarly found in the dependences

$$\alpha_o^{v_o} = \left(\frac{d\chi}{dv_o}\right)_{v_o=o}; \qquad \beta_o^{v_o} = \left(\frac{dM_0}{dv_o}\right)_{v_o=o};$$

$$\alpha_o^h = \left(\frac{d\chi}{dh}\right)_{h=o}; \qquad \beta_o^h = \left(\frac{dM_0}{dh}\right)_{h=o}.$$

Theoretical determination of the dependences $x = f(\delta, v_0, h)$ and$M_0 = f(\delta, v_0, h)$ is complicated because of the difficulties of the process of interaction of the planar paw with the soil. These dependencies should be found experimentally.

For this purpose, a laboratory installation was developed (see Fig. 61), which is a trolley moving along a rail track [8,38,40].

To the trolley is attached a strain gauge, on which a planar paw is mounted. The design of the mounting of the strain gauge allows the installation of the paw at any angle δ to the direction of movement of the trolley $\bar{v}_0$, and the variation within certain limits of the loosening depth of the soil h.

When moving the trolley, the values of the components R_ξ and R_η the main vector of the horizontal forces of resistance to the motion of the flat-topped paw were determined, as well as the principal moment M_0 of these forces with respect to the vertical axis of its rotation. From the data obtained, for each value of the angle δ, the angle $\chi = arctg\frac{R_\xi}{R_\eta}$ and modulus of the principal vector$R = \sqrt{R_\xi^2 + R_\eta^2}$were determined. Angle δ varied from 0 to 12° with an interval of 3°.

In Fig. 62*a*shows the experimental dependences $R = R(\delta), \chi = \chi(\delta)$ and $M_0 = M_0(\delta)$ at a constant depth of tillage (12 sm) and speed (2,5 m/s). The graphs of the function $x = \chi(\delta)$and $M_0 = M_0(\delta)$ have a form close to linear. Therefore, it is not difficult to determine the force parameters α_0^δ = of 2,98 and β_0^δ = 2,18 Nm. Analysis of the dependence $R = R(\delta)$ shows that the increase in R with increasing values of the angle δ is explained by the deterioration of the cutting conditions of the soil with a flat-paw, although the accepted working width of its gripping somewhat decreases.

Then the values $R_\xi, R_\eta$and M_0 were determined at a constant speed of the trolley v_0 and a constant angle δ, but at different depths of tillage (h),which varied within 8 - 20 sm with an interval of 4 sm. Fig. 62*b* shows the dependencies $R = R(h),\ \chi = \chi(h)$ and $M_0 = M_0(h)$. They determined the power parameters α_0^h = - 1,1 andβ_0^h =1,8 H·m. An analysis of these relationships shows that with increasing depth of soil treatment, the values of R and M_0increase, but the values of the angle χ decrease. The increase in R and M_0 is associated with an increase in soil density when the paw is buried, and a decrease in the values of the angle$\chi$is due to a redistribution of the values of the components R_ξ and R_η the principal vector R, that is, a decrease $R_\xi$and an increase R_η.

Then the values R_ξ, R_η and M_0were determined at constant values of the angle δ and the depth of soil treatment h. The speed of the trolley's movement varied from 1,0 m/s to 2,5 m/s every 0,5 m/s. Fig. 62*c*shows the dependencies$R = R(v_0)$. They determined the parameters $\alpha_0^{v_0}$= 3,33 and $\beta_0^{v_0}$ = 9,33 Nm. Analysis of these dependencies shows that when the rate of soil cultivation is increased by the flat-topped paw, the values of$R$, χ and M_0 increase.

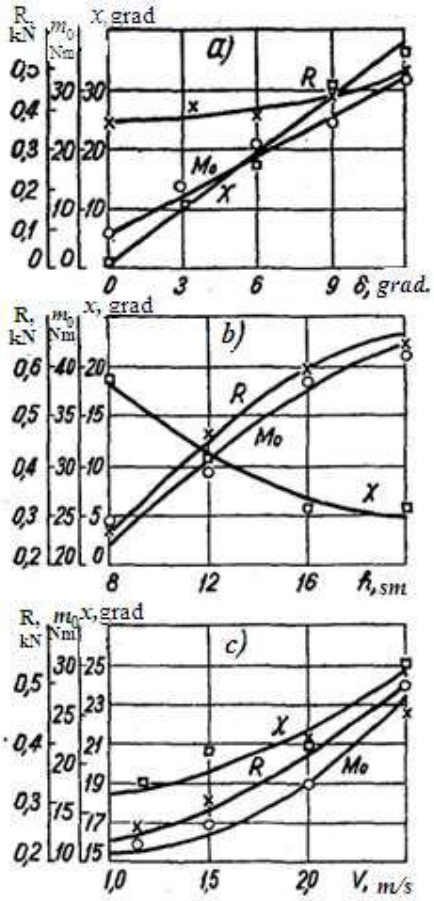


Fig. 62 Dependency graphs $R = R(\delta, h, v_0), \chi = \chi(\delta, h, v_0),$ и$M_0 = M_0(\delta, h, v_0)$:

a) with constant h=12 smand v_0=2,5 m/s;

b) with constant v_0=2,5 m/s and δ=9^0;

c) with constant h=12 sm and $\delta = 9^0$.

The above dependencies are used to analyze the influence of the type of connection with the tractor of flat-top machines (hinged or trailed) and their design parameters (the width of the grip, the location of the working bodies, etc.) on the stability of movement in the horizontal plane. Calculations and experimental studies were carried out on a laboratory installation, the design of which allowed it to be aggregated with tractors of the K-700 type in a hinged or trailed version, to change the working width from 2,8 to 13,6 m, to place the front bodies in two rows and at an angle to the direction movement (wedge), work with supporting self-aligning wheels or with fixed in the direction of motion [8,38,40].

The results of experimental studies on the determination of the root-mean-square value of the deviation angle of the installation from the direction of travel and its path in the deflected state are given in Tables 5 and 6.

Summarizing the results of calculation and experimental studies on the stability of the motion of flat-milling machines in the horizontal plane, we can draw the following conclusions:

- with the increase in the width of the capture of the flat-cutting machine from 3,6 to 13,6 m, the stability of its movement decreases;
- clinical placement of working elements on the frame of the flat-cutting machine, in comparison with the front one, reduces the root-mean-square values of the angular deviations from the direction of motion, but at the same time its path in the deflected state increases;
- when the flattening machine is trailed to the tractor, the root-mean-square value of the angle of deviations from the direction of movement is reduced in comparison with the hinged machine, and its path in the deflected state increases;
- with an increase in the speed of movement from 1,6 to 2,46 m/s, the root-mean-square values of the angular deviations from the steady motion increase, and the path traversed in the deflected state decreases;
- fixation of self-aligning support wheels allow to increase stability of movement a little.

Table 5 - Mean-square values of the deviation angle ($\pm\sigma_{\Delta\varphi}$) in the horizontal plane of the installation variants from the steady-state motion and the path (S) traversed by them in the deflected state
(v = 1,6m/s).

Working width of the device, m	Accommodation of working bodies	Connecting the unit to the tractor	Supporting wheels: self-aligning $\pm\sigma_{\Delta\varphi}$, grad.	self-aligning S, m	fixed $\pm\sigma_{\Delta\varphi}$, grad.	fixed S, m
13,6	wedged	trailed	3,61	31,20	3,49	30,20
		suspended	3,90	28,00	3,82	27,40
	frontaled	trailed	4,87	17,30	4,46	17,00
		suspended	5,67	16,50	5,53	14,50
8,2	wedged	trailed	0,93	18,20	0,84	14,60
		suspended	0,99	10,70	0,99	10,00
	frontaled	trailed	1,33	8,20	1,23	8,00
		suspended	1,36	7,60	1,38	7,10
3,6	wedged	trailed	0,52	'4.30	0,40	3,40
		suspended	0,64	3,10	0,59	3,10
	frontaled	trailed	-	-	-	-
		suspended	-	-	-	-

Table 6 - Mean-square values of the deviation angle ($\pm\sigma_{\Delta\varphi}$) in the horizontal plane of the installation variants from the steady-state motion and the path (S) traversed by them in the deflected state, at different speeds (the capture width of the installation is 8,2 m)

Speed of movement, m/s	Accommodationof working-tools	Connecting the unit to the tractor	Supporting wheels: self-aligning $\pm\sigma_{\Delta\varphi}$, grad	self-aligning S, m	fixed $\pm\sigma_{\Delta\varphi}$, grad	fixed S, m
1,60	wedged	trailed	0,93	18,20	0,84	14,60
		suspended	0,99	10,70	0,99	10,00
	frontaled	trailed	1,33	8,20	1,23	8,00
		suspended	1,36	7,60	1,38	7,10
2,00	wedged	trailed	1,46	21,00	1,39	15,10
		suspended	1,62	11,40	1,54	10,70
	frontaled	trailed	1,95	8,90	1,67	8,60
		suspended	2,02	8,00	2,00	7,20
2,46	wedged	trailed	2,53	25,00	2,16	16,30
		suspended	2,78	12,50	2,79	12,40
	frontaled	trailed	3,10	10,70	3,98	9,80
		suspended	3,30	9,80	3,05	8,30

Thus, to improve the stability of the movement of wide-cutting flat-top cutting machines in the horizontal plane, it is advisable to use the wedge arrangement of the working elements on the frame of the machine and the hinged (semi-mounted) way of connecting it to the tractor, to provide the possibility of fixing the self-aligning support wheels during operation. Most of these provisions are implemented in the designs of wide-cutting flat-cutting machines - KPSh-9, KPSh-11, PG-3-5, KTS-10-2 and others.

The stability of the movement of wide-spread soil-cultivating machines, in which the working bodies are disks, are examined in sufficient detail in the works of G.N. Sineokov and V.F. Strelbitsky [14,15] as with a symmetrical arrangement of discs (stubble cleaners, harrows), and asymmetrical (raking-seeder). The dependences obtained in these studies, obtained when considering the equilibrium conditions of disk machines, allow us to choose the valuesof the parameters of the circuits of these machines and their operating modes, which provide the necessary stability of motion in the horizontal plane.

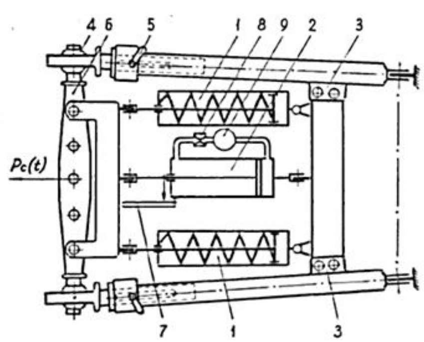
2.6. On the unevenness of traction resistance of cultivators-flat cutters

The process of traction resistance of soil-cultivating machines is uneven and depends on the type and parameters of the working organs, the physical and mechanical properties of the soil and the operating modes of the unit. It is established that each percent increase in the unevenness of the traction resistance of the machine causes a drop in the effective engine power by an average of 0,8% with a similar increase in fuel consumption [41]. Large fluctuations in load on the hook of the tractor lead to an unstable operation of the engine and a decrease in effective power to 20 - 30%. To reduce the unevenness of the traction resistance between the tractor and the technological machine, it is expedient to introduce an elastic element with a damper. In this connection, it became necessary to carry out the following works:
- explore the dynamic characteristics of the tractor-planar system - the soil and the unevenness of the tractive resistance with a rigid connection between the tractor and the flat-cutter;
- select the parameters of the elastic element with a damper;
- evaluate the efficiency of the introduction between the tractor and the flat-cutter of an elastic element with a damper and its effect on the dynamic characteristics of the system and the unevenness of the traction resistance.

The investigations were carried out on the autumn soil cultivation with the help of a trailing laboratory-field installation made on the basis of the KPSh-9 flat-top cultivator [31]. The unit was aggregated with a tractor K-700A at a speed of$v$ = 2,18 m/s. The background of the field is an untreated stubble of barley. The soil is sandy loamy. Its moisture content in horizons 0-5 and 0-10 sm, respectively, was equal to 5 and 9% of PPV, and hardness – 2,4 and 4,5 MPa.

Traction resistance $P_c(t)$ was recorded using a tensometric dynamometer installed between the tractor and the movable link of the parallelogram frame to which the tractor attachment mechanism was connected. Simultaneously with the traction resistance, changes in the hardness of the soils in the working area $p_c(t)$and in the loosening depth$h(t)$ were recorded with a tensometric hardness meter and a rheochord sensor attached to the frame of the installation. When recording and analyzing these processes, the method of magnetographic tensometry was used, which made it possible directly during the experiments to monitor the received realizations for stationarity and approximation of systems to linearity conditions.

To investigate the effect of elastic and damping elements and their parameters on the dynamic characteristics of the system and the nonuniformity of the $P_c(t)$process, a special device was developed, the scheme of which is shown in Fig. 63. The processes of displacement of the damper piston and springs were fixed by the rheochord sensor 7, and the pressure in the hydraulic cylinder behind the throttle 8 - with the help of strain gauge pressure sensor 9.



1 - elastic elements; 2 - hydraulic cylinder CS-75; 3 - the tides of the lower links of the tractor hitch; 4 - tips of the lower links; 5 - retainer for traction tips; 6 - trailer hitch bracket; 7 - rheochord sensor; 8 - throttling hole; 9 - pressure sensor

Fig. 63. Scheme of the device for studying the effect of the parameters of the elastic and damping elements of the tractor-planar-soil dynamic system on the nature of the traction resistance

Dynamic characteristics of the tractor-planar-soil system: mutual spectral planes $S_{P_{cp}}(\omega)$, $S_{P_{ch}}(h)$, mutual correlation functions - $R_{P_{cp}}(\tau)$, $R_{P_{ch}}(\tau)$ amplitude-frequency characteristics $W(j\omega)$; the coherence functions $K^2(\omega)$) were determined from two channels: in one case, the unevenness of the hardness of the treated soil layer p (t) and the output process $P_c(t)$), i.e. $[p(t) \rightarrow P_c(t)]$, and in the other -$[h(t) \rightarrow P_c(t)]$, where $h(t)$is the process of changing the depth of loosening.
In studying the transient processes of the system, a stepwise change in the output quantity was taken as the disturbing effect

$$у_{p_c}(t) = \begin{cases} 0 & t \leq 1 \\ у_{p_c}(t) & t > 0. \end{cases} \tag{91}$$

Such a perturbation most fully characterizes possible cases of maximum rise and fall of the load $P_c(t)$ on the hook of the tractor. For some idealization (91) such a perturbation was achieved as follows:
- the installation was buried at a predetermined depth (h = 10 sm), which was monitored on the 15-20 m path;
- after the penetration, the installation was displaced backward by 0,1-0,15 m;
- with the help of a tractor, all gaps in the joint of the sample were selected with a slight tension;
- the rated frequency of the tractor engine was set;
- the beginning of the movement was carried out by a sharp touch, then the unit moved on the road 15-20 m.

When recording transients, an oscillographic method was used to record the traction resistance $P_c(t)$move the piston of the hydraulic cylinder H (t), and the oil pressure $P_c(t)$From the obtained oscillograms, the duration of the transient process (t_c) and the overshoot f_n

$$f_n = \frac{A_1}{A} \cdot 100\% \tag{92}$$

where: A_1 - exceedance of the maximum value of $P_c(t)$over the mean value at the time of motion:A is the average value of $P_c(t)$.

When choosing the parameters of the elastic and damping elements, the criterion was taken to be the dynamism factor of the system

$$\rho_d = T_1/T_2, \tag{93}$$

where:T_1andT_2 are the time coefficients characterizing respectively the damping and inertial properties of the system.

In this case, in the case ρ_d<1, the system will have vibrational properties, and if ρ_d> 1 - the character of the oscillation will be aperiodic.
The time constant$T_2$is found from formula

$$T_2 = \sqrt{\frac{m}{C_2}}\ , \tag{94}$$

where: C_2- coefficient of elasticity (stiffness); m - is the reduced mass of the aggregate, which with some assumptions is

$$m = \frac{m_{\text{т}} + m_{\text{м}}}{m_{\text{т}} \cdot m_{\text{м}}} \tag{95}$$

where: $m_{\text{т}}$и$m_{\text{м}}$ - are the mass of the tractor and the machine, respectively.

T_1 is found as the ratio of the damping coefficient C_1 to the stiffness coefficient of the elastic element c_2.

$$T_1 = \frac{C_1}{C_2}. \tag{96}$$

The coefficient C_1depends on the load $P_{\text{ц}}$ (the pressure in the cylinder before the throttle) and the speed υ_1 of displacement of the piston of the cylinder and is found from the relation

$$C_1 = \frac{P_{\text{ц}}}{v_1}. \tag{97}$$

At the same time, $v_1$is determined from the graphic display (oscillogram) of the transient process of piston displacement $H(t)$. For this purpose, a tangent to the inflection point of this curve is carried out and, taking into account the scale factors,

$$v_1 = H_n/t, \tag{98}$$

where:H_n— the amount of displacement of the piston beyond t is the time of this displacement.

Evaluation of the effectiveness of the introduction of elastic coupling between the flat-cultivator and thetractor, as well as the identification of the main perturbing factors on the character of the formation of$P_c(t)$, were carried out by the identification method using magnetographic tensometry and computation.

Graphs $S_{P_{cp}}(\omega)$, $W(j\omega)$, $R_{P_{cp}}(\tau)$and$K^2(\omega)$ dynamic system tractor-planar-soil with a rigid connection between the tractor and the plane cutter through the channel $p(t) \rightarrow P_c(t)$.) are shown in Fig. 64.

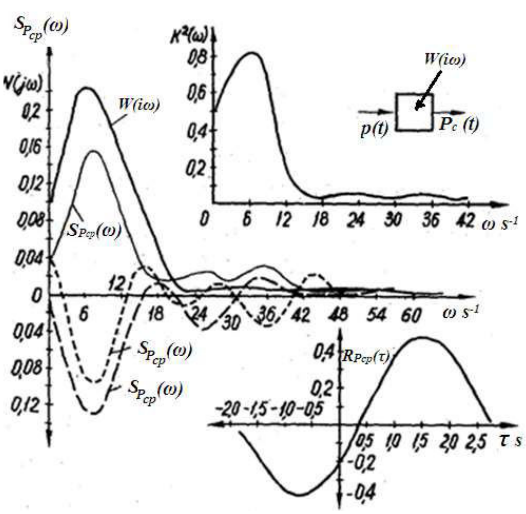


Fig. 64. Graphs $S_{P_{cp}}(\omega)$; $W(j\omega)$; $R_{P_{cp}}(\tau)$and$K^2(\omega)$ dynamic systemtractor-planar-soil with a rigid connection between the tractor and the flat-cutter through the channel$p\ (t)\ \rightarrow\ P_c\ (t)$

These graphs clearly reveal the periodicity (oscillations) of the changes in traction resistance and the relatively high mutual interrelation of these oscillations with the hardness of the soil. In this case, at a frequency ω = 6s^{-1}, the effect of soil hardness on the character of traction resistance approaches a linear one, as $K^2(\omega)$≈0,8. At a frequency ω≈15s^{-1}, this connection practically disappears, which indicates that the given channel of transmission of the disturbing effect is narrow-band.

At the maximum interconnection of soil hardness $p(t)$ and traction resistance $P_c(t)$in the range ω = 0 - 15s^{-1}, the real$s^d_{P_{cp}}(\omega)$and imaginary $s^m_{P_{cp}}(\omega)$components of the energy spectrum of the frequencies have negative values. This indicates that the module of mutual spectral density$S_{P_{cp}}(\omega)$ lags behind the effects on the hardness plane of the soil by 90 - 180° and is in the third quadrant. Such a delay angle is highly undesirable.

The graph of the mutual correlation function $S_{P_{cp}}(t)$ indicates the presence of a latent periodic component of the mutual bonds$p(t)$ and $P_c(t)$. The maximum of positive interrelation is observed at τ = 1,5s, and negative - at τ = -0,75s. The complete loss of the connection between these processes occurs at τ = 0,35s, which corresponds to the path L = 0,76 m passed (at v = 2,18 m/s).

Analogous graphs are obtained for the dynamical system under consideration by the channel$h(t) \rightarrow P_c(t)$. With a rigid connection, the dependence of $P_c(t)$ on $h(t)$ is negligible. The maximum value of the coherence function $K^2(\omega)$ does not exceed 0,16, and the energy spectrum $S_{P_{ch}}(\omega)$modules are located in a wider range ω = 0 - 24 s^{-1}.

Thus, for these conditions and operating conditions, the unevenness of the traction resistance is largely due to the process of changing the hardness of the treated soil layer than the change in the depth of processing. On the channel $p(t) \rightarrow P_c(t)$the dynamical system under consideration, with a rigid connection between the tractor and the flat-top cultivator, processes the input disturbances unsatisfactorily. The output process $P_c(t)$ has a large nonuniformity, since the system itself has oscillatory properties in the transmission of input disturbances. In order for this system to operate in an aperiodic mode, it is advisable to introduce an elastic element with a de-shaping device between the tractor and the flat-cutter.

The choice of the rigidity of the elastic element and the diameter of the throttling damper hole was based on the obtained values of the mean value of the process $P_c(t)$) M_{P_c} = 24 kN), the dispersion (D_{P_c}= 4 kN^2), the energy spectrum $S_{P_{cp}}(\omega)(\omega$ = 0-12 s^{-1}), and also the permissible sizes from the point of view of constructive design of the device. For these conditions, the total stiffness of the elastic elements was assumed to be c_2 = 500 kN/m, and the cross-sectional area (S_d) of the throttling opening of the hydraulic damper (hydraulic cylinder CS-75) should be within S_d = 10-80 mm^2.

With the known masses of the tractor m = 13,500 kg, the cultivator-plane cutter m = 2,800 kg and the accepted value of C_2 according to (94) and (95), we find that
T_2≈0,068s. The results of processing the oscillograms, as well as the calculation of the coefficients T_1 and ρ_d according to the formulas (93, 96, 97, 98) for various areas of the cross section of the throttling holes are given in Table 7.

Table 7 - Dynamic variation of the system under consideration when an elastic element with a damping device is inserted between the tractor and the flat-top cultivator plane cutter, depending on the cross-sectional area ofthe throttling device

Name of indicators	The values of the exponents for the areas of the cross-section of the throttling hole (mm)^2		
	78,50	20,00	10,00
Piston movement - N, m	0,0286	0,0467	0,0411
Travel time - t, s	0,060	0,110	0,093
Piston travel speed - υ, m/s	0,476	0,592	0,411
Traction resistance in time t - P_c, kN	42,75	51,40	46,60
Coefficient of damping - C_1, kN s/m	90,28	101,10	106,00
Coefficient of elasticity - C_2, kN/m	500	500	500
Time coefficients, s			
T_1	0,180	0,242	0,212
T_2	0,068	0,068	0,068
Dynamical coefficient - ρ_d	1,33	1,78	1,56

An analysis of the data of this table shows that within the limits of the investigated areas of the cross section of the throttling damper hole, the process of oscillations of the system under consideration is aperiodic ($\rho_a > 1$). However, with a cross-sectional area of the throttle opening of 20 mm^2, the system is closer to the properties of a first-order aperiodic link. Such a change in P_c (t) as an output process of a dynamical system is clearly confirmed by transient processes along the channel p (t)→P_c (t)(Fig. 65).

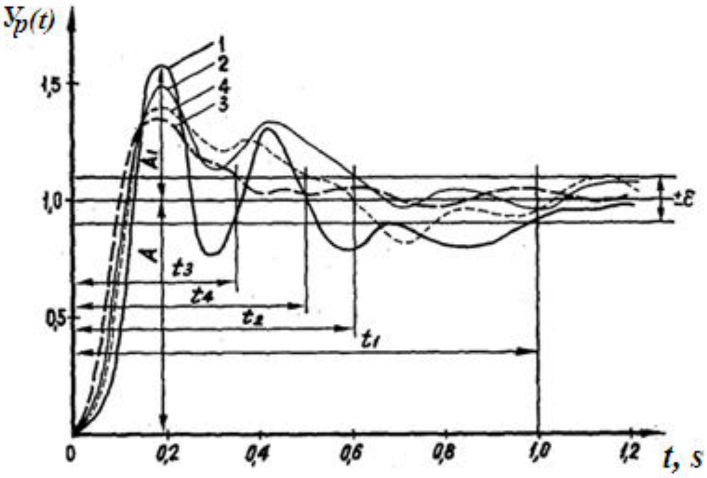


Fig. 65. Transient functions of the dynamic system with rigid and elastic couplings with a damper between the tractor and the plane cutter through the channel p (t)→P_c (t);1 - rigid connection; 2,3,4 - elastic connection with the damper, respectively, with the cross-sectional area of the throttle 78,5; 20 and 10 mm^2

With a rigid connection between the tractor and the flat-top cultivator, the amount of overshooting of the system reaches $f_{n_1} \approx 58\%$, and the duration of the transient process is $t_1 \approx 1{,}0$ s. The introduction into the system of an elastic element with a damper with a cross-sectional area of a throttling opening of 78,5 mm^2 leads to the appearance of the properties of aperiodicity (a gentler decline with a decrease in the amplitude of the oscillations). Although the overshoot value varies insignificantly $f_{n_2} \approx 48\%$, however, the transient time decreases to $t_2 \approx 0{,}6$s. With a decrease in the cross-sectional area of the throttle to 20 mm^2, the overshoot is reduced to $f_{n_3} \approx 32\%$, and the transient time to $t_3 \approx 0{,}35$s. At the same time, the aperiodicity of the process decline is more pronounced. Further reduction of the diameter of the throttling hole to 10 mm^2 leads to an increase of $f_{n_4} \approx 40\%$ and t_4 to 0,5s. In this case, the rigidity of the coupling of the plane to the tractor increases and the attenuation coefficient relative to the current average value ofP_c (t); decreases, and the oscillatory properties of the output process begin to appear again. The introduction of an elastic licking hole (20 mm^2) between the tractor and flat-nose significantly and positively affects the magnitude and nature of the drag process. This follows from the comparison of the graphs $S_{P_{cp}}(\omega)$; $W(j\omega)$and $K^2(\omega)$of the dynamical system with the rigid (see Fig. 64) and elastic with the dampener (Fig. 66) by the connections between the tractor and the machine.

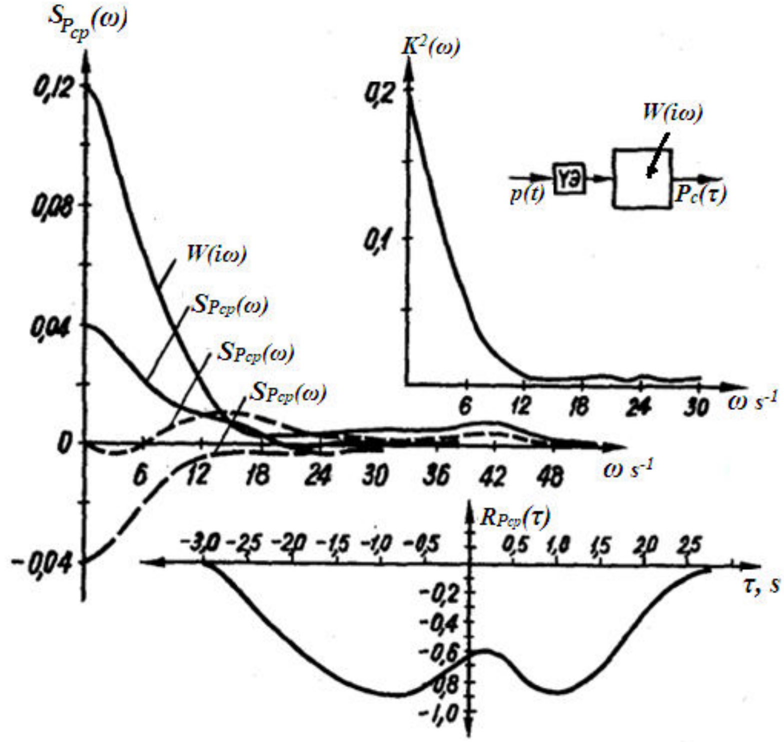


Fig. 66. The graphs $S_{P_{cp}}(\omega)$, $W(j\omega)$, $R_{P_{cp}}(\tau)$, and $K^2(\omega)$of the tractor-planar-soil dynamic system, when an elastic coupling with a damper is introduced between the tractor and the machine with the cross-sectional area of the throttling hole 20 mm^2 along the channel $p\ (t)\rightarrow P_c\ (t)$

The values of the mutual spectral density module $S_{P_{cp}}(\omega)$ are shifted to the region of infra-low frequencies. If, in the case of a rigid coupling, the maximum value of$S_{P_{cp}}(\omega)$is 0,156 at ω = 6 s^{-1} (see Fig. 64), then the introduction of an elastic coupling with the damper reduces $S_{P_{cp}}(\omega)$at the same frequency to 0,02, and its maximum value reaches only 0,04. Similarly, the value of the transfer function decreases from $W(j\omega)$ = 0,226 for a rigid coupling (see Fig. 64) to $W(j\omega)$,= 0,12 when an elastic coupling with a damper is introduced into the system (Fig. 66). This is explained by the fact that the mutual connection between the processes $P_c(t)$and $p\ (t)$practically disappears, since the coherence function $K^2(\omega)$decreases at a frequency ω = 6 s^{-1} from 0,8 (see Fig. 64) to 0,06 (Fig. 66). The maximum of the coherence function between the processes $P_c\ (t)$ and $p\ (t)$ does not exceed $K^2(\omega)$ = 0,2.

The connection of the input perturbation h (t) and the output process $P_c\ (t)$ with an elastic coupling is practically absent $K^2(\omega)\leq$ 0,1.

The resultant coherence function$\sum K_{ph}^2\ (\omega)$ between the output process $P_c\ (t)$, the input processes $P(t)$ and $h(t)$(Fig. 67) shows that in the case of a rigid coupling in the system, the nature of the $P_c(t)$ process is mainly due to changes in these perturbations $\sum K_{ph}^2\ (\omega)_{max}\approx$ 0,92. The influence of these perturbations on the character of$P_c\ (t)$, is mainly observed in the frequency range ω = 0-21 s^{-1}.

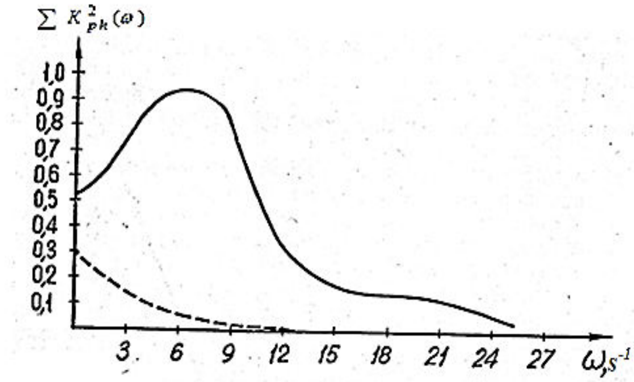


Fig. 67. The resulting coherence function$\sum K_{ph}^2(\omega)$ of the output process $P_c(t)$,of the system from external perturbations $p(t)$and h (t) in the connection between the tractor and the machine is rigid (-) and elastic with a damper (- - -)

In the region of frequencies ω = 3 - 9s^{-1}, they practically completely form the character of the drag process of the cultivator-plane, since the total coherence function in this case is $\sum K_{ph}^2(\omega) > 0.8$. The introduction into the system between the tractor and the flat-top cultivator of the elastic coupling with the damper narrows the frequency range to ω = 0 - 6s^{-1}. Moreover, at a frequency ω = 0 - 6s^{-1} ($f \cong 1$ Hz), where practically linear dependence of $P_c(t)$,on $[p(t) + h(t)]$was observed, the coherence function$\sum K_{ph}^2(\omega) \leq 0{,}1$, and at infra-low frequencies$\omega \leq 0{,}1 s^{-1}$, the interrelation of these processes does not exceed $\sum K_{ph}^2(\omega) \leq 0{,}3$.
Thus, the introduction between the tractor and the installation of an elastic connection with the damper should, in comparison with the rigid coupling, reduce the unevenness of the traction resistance. The results of the experimental verification confirmed this position. With approximately identical operation modes of aggregates, the introduction of an elastic coupling with the damper between the tractor and the flat-top cultivator made it possible to reduce the standard deviation of the drag process by 22,2 - 35,7%, and the coefficient of variation - by 14,4 - 34,1%. At the same time, there has been a tendency to reduce the average value of traction resistance by 2,5 - 9,2%.

2.7. Choosing the optimal combination of working width and speed of soil-cultivating machines and soil conservation tools

To obtain the best operational and economic performance of soil-cultivating machines and tools of soil-protecting agriculture, when assembling them with a tractor of a certain class of traction, it is necessary to choose the optimal gripping width and speed of the machine. It is known that in order to choose such an optimal combination, it is necessary to establish not only the laws governing the variation of the traction resistance of the machine and the required power when operating at specified speeds, but also the effect of the working width and speed on the unit's productivity and direct operating costs when performing a particular technological operation.
When considering the questions of aggregating tractors with agricultural machines, it is necessary to distinguish, according to the degree of importance, the agrotechnological and operational-economic indicators by which comparative evaluation of aggregates can be made. Undoubtedly, any aggregate must correspond to agrotechnological (initial) requirements. Operational and economic indicators should be considered together, highlighting the main and auxiliary ones [32,34,41,42]. The main operational and economic indicators include unit capacity and labor costs, fuel consumption per unit of work performed, direct operating costs, unit investment and reduced costs, determined taking into account the return on investment. The traction efficiency of the tractor, the utilization factor of the shift time, the specific fuel consumption, the tractor skidding, etc., are included in the auxiliary ones.
There are several criteria for choosing the optimal grasp width of the unit: for minimum labor costs or reduced costs. When developing and implementing new technology, when selecting the most

cost-effective options, the main indicator of economic efficiency is the cost that includes direct operating costs and capital investments, taking into account the industry standard efficiency factor. The indicator of productivity growth, although it refers to the main indicators of economic efficiency but does not serve as a criterion for choosing the most cost-effective options but is mainly used in planning to determine to what extent the introduction of measures for the development of technology affects the performance of the plan for productivity growth, the number of workers and their wage fund.

The productivity of the machine-tractor unit exerts a significant influence on the costs of living labor, the cost of production, specific investment and reduced costs. But when choosing the most cost-effective options for new technology, there are cases when one option has the best performance indicators (labor costs) and the worst for the given costs, and the second - on the contrary. In this case, preference is given to the second option, since the main and the main economic indicator of choice are the reduced costs [34].

On the basis of the foregoing, it is advisable to take the minimally reduced costs per unit of work done as a criterion in choosing the optimal combination of the width of capture and the speed of movement of soil-cultivating machines and implements of soil-protecting agriculture U_{np}[8]. At the same time, it is necessary to determine and analyze indicators such as productivity per hour of shifting time - W, living labor costs - $З_т$, fuel consumption - Q_g and material consumption per unit of work performed - M_e.

The process of soil cultivation by machines and implements of soil-protecting agriculture is characterized by the following indicators that affect the values of the criterial indicators

$$\begin{aligned} &P_{op} = f_1(B_0;\ v_n); \\ &\delta_0 = f_2(P_{op};\ G_m); \\ &P_f = f_3; (f_m;\ G_m); \\ &N_{dv} = f_4(P_{op};\ P_f; \delta_0; v_n); \\ &W = f_5(B_0;\ v_n; L_*; \tau_{sm}), \end{aligned} \tag{99}$$

where: P_{op}- pulling resistance of the machine; δ_0- coefficient of slipping of the tractor; P_f- the effort to roll the tractor; N_{dv}- engine power required to overcome the resistance of the soil to the machine and the rolling of the tractor, taking into account the skidding ratio; $B_0$is the grip width of the machine; $G_т$ - is the weight of the tractor; $f_т$ - is the drag coefficient of the tractor rolling; L_* -the length of the rut; τ_{sm} - coefficient of working time use of a shift.

Criteria (goal functions) are generally equal

$$U_{np} = f_6(W);\ B_0;\ B_t;\ t_{gt};\ q_e;\ N_{dv}; B_{ud.m};\ t_{gm});$$

$$\begin{aligned} &З_т = f_7(W);\ \Lambda_{тм}; \\ &Q_r = f_8(q_e;\ N_{dv}W); \\ &M_e = f_9(B_0; W) \end{aligned} \tag{100}$$

where:B_t- is the book value of the tractor; $B_{ud,m}$ - the price of 1 m of the capture machine; t_{gt} and t_{gm} - annual load, respectively of the tractor and the machine; $\Lambda_{тм}$- number of personnel serving the unit.

The change in the traction resistance of the machine, depending on the width of the grip and the speed of its movement, is expedient to determine by the rational formula of Academician V.P. Goryachkin, represented in the form

$$P_{op} = B_0(f_0)\gamma_m + k_1 a + \varepsilon_0 a v_m^2 \tag{101}$$

where: f_o - is the sum (reduced) coefficient of friction; k_1 - coefficient characterizing the resistance of the cross-sectional area of the deformation soil layer;γ_m - weight of the flat-cutting machine, per 1 m of grip; ε_0 - is the coefficient characterizing the resistance resulting from the transfer of kinetic energy to the soil layer during its transfer.

The pull resistance of the machine (P_{or}) is determined depending on the type of work, the depth of processing and the speed of movement, based on the results of field tests. Thus, for example, analysis and systematization of such data made it possible to reveal the dependence of the average resistivity of flat-crop cultivators (per 1 m of capture) on the speed of movement for three types of technological operations (Fig. 68). The obtained dependences were approximated by the rational formula (101). The coefficients k_1 and ε_0 were determined by the method of least squares and are given in Table 8.

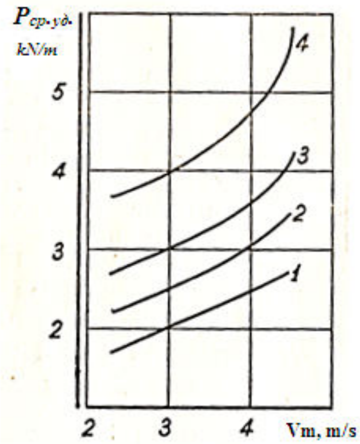


Fig. 68 Average specific (per 1 m of grip) the resistance of the cultivator-plane from the speed of movement in the performance of various technological operations:
1 - pre-sown loosening a = 7 - 8 m;2 - processing of steam a = 10 sm;
3 - processing of steam and autumn plow a = 12 cm;
4- processing of autumn plow a = 16 sm.

Table 8 - Values of the coefficients k_1 and ε_0

Operation	Depth of loosening, sm	k_1, N / m^2	ε_0, N·s^2/m^4
Presowing tillage	7	9340,01	790,45
	10	10750,39	800,01
Fallow land cultivation	12	11710,12	800,22
Basic autumn cultivation of soil	12	11710	800,22
	16	13840,18	740,28

The coefficient of slip depends mainly on the load on the hook of the tractor, the type of its propulsors, the physical and mechanical properties of the soil, its state, etc. Therefore, the values of the tractors slipping coefficients should be determined by the formula [34]

$$\delta_0 = a'(P_{op}/G_{\text{т}}) + b'(P_{op}/G_{\text{т}})^{c'}, \tag{102}$$

where:a';b'; $c'$are the empirical coefficients characterizing the design features of tractors' tractors and the conditions of their operation.

The values of the coefficients of skidding are taken from the traction characteristics of tractors constructed from experimental data on various backgrounds. The experimental data $\delta_0 = f(P_{cr})$are approximated by an expression of the form (102).

To do this, according to these dependences, the coefficients δ_{01}; δ_{02}; $\delta_{03}$for the corresponding forces on the hook of the tractor $P_{\text{кр}_1}$; $P_{\text{кр}_2}$; $P_{\text{кр}_3}$and a system of equations

$$\begin{gathered}
\delta_{01} = a'(P_{\text{кр}_1}/G_{\text{т}}) + b'(P_{\text{кр}_1}/G_{\text{т}})^{c'}; \\
\delta_{02} = a'(P_{\text{кр}_2}/G_{\text{т}}) + b'(P_{\text{кр}_2}/G_{\text{т}})^{c'}; \\
\delta_{03} = a'(P_{\text{кр}_3}/G_{\text{т}}) + b'(P_{\text{кр}_3}/G_{\text{т}})^{c'};
\end{gathered} \tag{103}$$

The solution of this system with respect to c' has the form

$$P_{\text{кр}_1}^{c'}=(\delta_{02}P_{\text{кр}_3}-\delta_{03}P_{\text{кр}_2})+P_{\text{кр}_2}^{c'}(\delta_{03}P_{\text{кр}_1}-\delta_{01}P_{\text{кр}_3})+$$
$$+P_{\text{кр}_3}^{c'}(\delta_{01}P_{\text{кр}_2}-\delta_{02}P_{\text{кр}_1})=0. \quad (104)$$

To simplify the solution of (104), the values $P_{\text{кр}_1}$; $P_{\text{кр}_2}$; $P_{\text{кр}_3}$ are represented as power numbers with one base, but with different exponents $P_{\text{кр}1}=A_1^{\text{К}_1 c'}$; $P_{\text{кр}2}=A_1^{\text{К}_2 c'}$; $P_{\text{кр}3}=A_1^{\text{К}_3 c'}$;, where $K_1<K_2<K_3$ by the number 2, 4, 6, etc. For the known C' coefficients a' and b' are found from the expressions

$$b'=(\delta_{02}-\delta_{01}\left(\frac{P_{\text{кр}2}}{P_{\text{кр}1}}\right)\left[\left(\frac{P_{\text{кр}2}}{G_{\text{т}}}\right)^{c'}-\frac{P_{\text{кр}2}}{P_{\text{кр}1}}\left(\frac{P_{\text{кр}2}}{G_{\text{т}}}\right)^{c'}\right]^{-1};$$

$$a^1=\frac{G_{\text{т}}}{P_{\text{кр}1}}\left[\delta_{01}-b^1\left(\frac{P_{\text{кр}1}}{G_{\text{т}}}\right)^{c^1}\right].$$

The results of calculating the coefficients a', b' and c' for equation (102) from the experimental data $\delta_0=f(P_{\text{кр}})$ are given in Table 9.

Table 9 - Values of the coefficients *a', b'* and *c'*

Operation	Background	Tractor class	Type of propulsor	a'	b'	c'
1. Presowing tillage	Processed field	3	crawler	0,017	3,20	5,30
		5	wheeled	0,228	11,80	5,914
2. Fallow land cultivation	Processed field	3	crawler	0,017	3,20	5,30
		5	wheeled	0,228	11,80	5,914
3. Basic (autumn) cultivation of soil	Untreated field (stubble)	3	crawler	0,040	21,00	14,20
		5	wheeled	0,213	23,95	6,065

The engine power required to overcome the resistance of the tiller and the self-movement of the tractor, taking into account slippage, is determined when the efficiency of the propellers and transmission is 0,88 by the formula

$$N_{\text{дв}}=3{,}1\cdot v_n(P_{op}+P_f)/(1-\delta_0), \quad (105)$$

where: v_n − travel speed (km/h), and $P_f=f_{\text{т}}\cdot G_{\text{т}}$.

The capacity of the unit (ha/h) of the shift time was determined from equation

$$W=0{,}36\cdot B_p\cdot v_n\cdot\tau_{\text{см}},$$

where: B_p - is the working width of the tool.
Since $B_p=B_0-\Delta B_0$, a $\tau_{\text{см}}=\rho'(m'-n'v_n)^{B_0}$, then

$$W=0{,}36(B_0-\Delta B_0)v_n\cdot\rho'(m'-n'v_n)^{B_0}. \quad (106)$$

The factors ρ', m', n' take into account the natural conditions, the length of the gong, the size of the field, the technological and technical requirements and parameters of the unit. The average length of the gon is assumed to be 1500 m as the most common in the steppe regions of Kazakhstan.
Therefore, according to the developments of the All-Union Mechanization Institute (VIM), the following values of the coefficients entering into the formula (106) for the determination of W are adopted:

$$\rho'=0{,}925;\quad m'=1{,}001;\quad n'=0{,}00703.$$

The resulted expenses for processing of 1 hectare of area are defined by expression

$$U_{пр} = З_0 + А_0 + R_з + C_г + X_x + K_a, \quad (107)$$

where: $З_0$ - payment of labor of machine operators; $А_0$ - depreciation charges; $R_з$ - costs for capital and current repairs and technical care; $C_г$ - expenses for fuel and lubricants; X_x - storage costs; K_a-norm efficiency investment.

To find the quantities that determine the reduced costs, use the known formulas, and the values of the regulatory ratios included in them are taken according to normative documents.

For cultivators-flat cutters, for example, formula (107) takes the form

$$U_{пр} = \frac{1}{W}\left[1{,}124 + 0{,}3736B_0 + \frac{B_т}{t_{2т}}\left(\frac{a'_r + r_т + 25}{100}\right) + 0{,}67 \cdot 10^{-4} q_c\, N_{дв}\right]. \quad (108)$$

where: $r_т$ - is the normative percentage of the deduction for the repair of the tractor;
a'_r - the normative percentage of deductions for the tractor's amortization;
q_c - is the specific fuel consumption per unit of power.

Additional criterial indicators - living labor costs, fuel consumption and material consumption were calculated by the ratios

$$З_т = \frac{1}{W}; \; Q_g = 10^{-3} q_e \cdot \frac{N_{дв}}{W}; \; M_e = 0{,}8 \cdot B_0/W. \quad (109)$$

To solve the problem numerically, a mathematical model is constructed on the computer for choosing the combinations $B_0 v_n$, which represents a structural diagram expressing in the enlarged form the relationship between the variables of the functions being solved (99). The structural scheme for selecting $B_0 v_n$, flat-top cultivators is shown in Fig. 69.

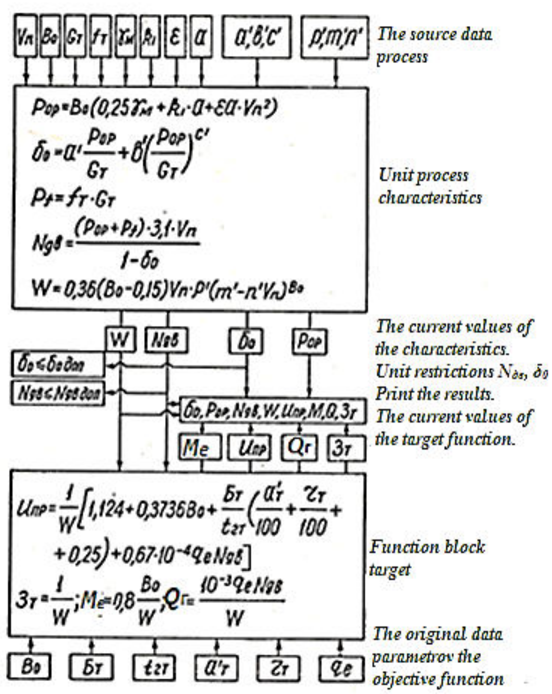


Fig. 69. Structural diagram of mathematical model for investigation and choice of working width and speed of motion of flat cutting units.

The resulted expenses are defined at admissible loading of engines of tractors on capacity, $N_{дв.доп} = (0{,}85\text{-}0{,}95) \cdot N_{дв.ном}$.

On the basis of the obtained results of the solutions for each tractor traction class, type of work and depth of loosening, the dependences are constructed; $N_{дв} = f(B_0)$; $W = f(B_0)$; and $U_{пр} = f(B_0)$for all given values of the argument v_n. For example, in Fig. 70 shows these dependences for the K-700 tractor when plowing is processed to a depth of 12 sm and a length of the race is 1500 m. At appropriate scales on the graphs of the dependencies $N_{дв}= f(B_0)$; ; we build zones of permissible capacities for tractors, for example,$N_{дв}$= 148 and 222 kW. The intersection points of straight lines and the curves of the dependences $N_{дв} = f(B_0)$; give possible combinations of the capture width and the speed of movement of the aggregates ($B_0 v_n$) that optimally load the tractor. Below the allowable zone (line), any combination of ($B_0 v_n$) will be underloaded by the tractor, and above - overload. Having lowered the intersection points on the abscissa axis (B_0) we obtain for each value of v_n the corresponding capture width of the aggregate. Thus, we get optimum combinations of $B_0 v_n$ for loading the engine for each tractor, type of work and depth of loosening. Knowing the value and the corresponding optimum speed from the graphs of the dependences $W = f(B_0)$and $U_{пр} = f(B_0)$, we determine the productivity and the resulted costs corresponding to this capture width of the unit. Using equations (109), we calculate for optimal combinations the values of the additional exponents - $З_т$,Q,andM_e. According to the results of these data, for optimal combinations, dependences $U_{пр} = f(v_n)$; $W = f(v_n)$; $Q_g = f(v_n)$; $З_т= f(v_n)$; and $M_e = f(v_n)$, for each class of tractor, type of work and depth of loosening.

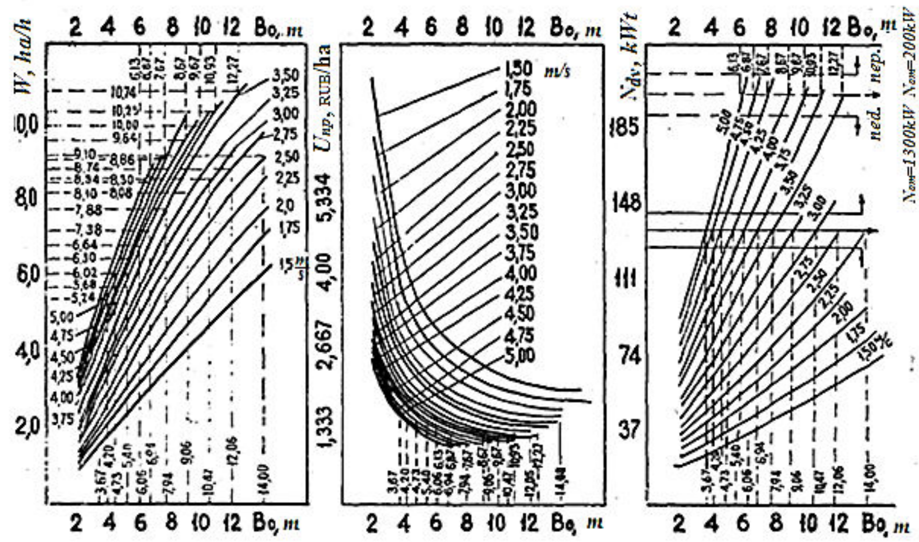
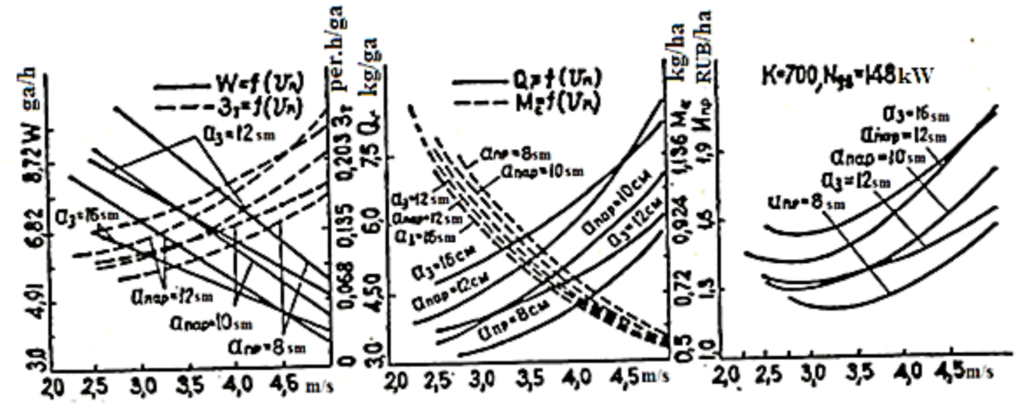


Fig. 70. Productivity (W), reduced costs ($U_{пр}$) and engine power (N_{dv}), depending on the width of the gripper (K-700, plowing - a = 12 sm, L_*= 1500 m)

As an example, Fig. 71 shows these dependences for the tractor "Kirovets" (N_{dv} = 148 kW).The nature of the change in the parameters $W, Q_g, З_т, M_e$, which mainly determine the amount of the reduced costs, will determine the minimum for each class of tractor, type of work and the depth of loosening of the reduced costs, which corresponds to a certain optimal engine load combination B_o, v_n(for cultivators flat-cutters, see Table 10).



$a_{пар}$– steamtreatment; $a_{пр}$ – pre-treatment; $a_з$ - autumntreatment

Fig. 71. Productivity (W), fuel consumption (Q_t), labor input ($З_т$) and material consumption (M_e), the reduced costs ($U_{пр}$) of the flat-topping unit as a function of υ_n, with the optimal ratio of $B_0 v_n$(K-700, $N_{дв}$ = 148 kW)

Table 10 - Minimum reduced costs (RUR / ha) of flat-topped units with tractors of Class 3 and 5, having different engine capacities

Class of tractor and engine power, kW	Preseeding processing, a = 7 - 8 sm	Fallow land treatment		Summer plowing	
		a= 8 sm	a= 10 sm	a= 12 sm	a= 16 sm
3 55,5	1,071	1,200	1,306	1,225	1,447
3 66,6	0,958	1,078	1,183	1,107	1,312
3 111,0	0,887	0,968	1,041	0,988	1,156
5 148,0	1,2290	1,3325	1,4473	1,3473	1,5760
5 222,0	1,0896	1,1802	1,2446	1,2110	1,3871

In the investigated range of engine power and the speed of movement of flat-topped aggregates, the minimum reduced costs for tractors of one class of traction, depending on the type of work and the depth of loosening, decrease with increasing (forcing) the engine power.
The points of inflection of the functions $U_{пр} = f(v_n)$ for the aggregates under study with an increase in the energy capacity of the planar loosening are shifted at the optimal combination B_o, v_n toward lower velocities (see Fig. 71). Moreover, for tractors with higher power the magnitude of this displacement increases. So, for flat-topped units with tractors K-700 and T-150, the displacement of the optimum speed lies in the range of 18 - 23%, and for DT-75 it is only 9%.
According to the method described, optimal combinations of the width of capture and the speed of movement of such soil-cultivating machines and implements of soil-protecting agriculture as flat-top cultivators, flat cutters and deep loosers to tractors of various classes are determined.
The width of the cultivation of flat-top cultivators to tractors DT-75, T-150 and Kirovets should be respectively within the limits of: 4,6-5,00; 6,4-7,00 and 8,50-10,50 m. Optimum speeds of movement of such units are equal: for tractors of DT-75 type with $N_{дв}$ = 55,5 kW - 7-9 km/h, and at $N_{дв}$= 66,6 kW 8-11 km/h; for the tractor T-150 -10,0 - 13,5 km/h; for tractors of the "Kirovets" type with the engine$N_{дв}$= 148 kW -13,6 km/h, and for $N_{дв}$ = 222 kW -10,0 - 13,6 km/h. At the same time tractive efficiency of tractors in different types of work is: for tractors of the type DT-75 with $N_{дв}$ = 55,5 kW – 0,40 - 0,67, and with $N_{дв}$ = 66,6 kW – 0,42 - 0, 68; for T-150 – 0,50-0,71; for the "Kirovets" with $N_{дв}$ = 148 kW and $N_{дв}$ = 222 kW – 0,43-0,62.When processing steam and autumn plow at a depth of 17,5-27,5 sm, the width of the take-up of flat-cutting plows for Class 3 tractors, which have a power of 55,5 kW and 66,6 kW, lies within 2,6-3,5 m at speed movements 5,4 - 7,2 km/h; for tractors of Class 5, which have a power of 148 and 222 kW engines, the width of

the gripper of the machine should be 4-5 m at a speed of 7,2-9,9 km/h. At the same time, tractive efficiency of tractors of class 3 and 5 lie, respectively, in the range 0,5-0,7 and 0,42-0,65.

3. Seeding machinery for soil conservation agriculture

3.1. Operating conditions and basic agrotechnological (initial) requirements for seeding machines for soil conservation agriculture

Soil-protective agriculture is widespread in many countries of the world, in huge areas and, as a rule, in such arid conditions where the leguminous crops are cultivated. Therefore, let us consider the results of theoretical and experimental studies, which are the basis for creation of machines for sowing of the leguminous crops.

With the use of soil-free tillage methods, up to 70-80% of crop residues remain on the surface of the field, about 20-30% of them were been sealed by the working bodies of tillage machines to the depth of seeding. Under such conditions, the two-disc coulters and packing devices, which are used in the dumping system of agriculture, become practically inoperative. Small distances between the two-disc working elements in the row and between the rows, between the field surface and the frame of the machine, between the rings of the rollers, contribute to clogging the machines with crop residues and deformed soil-such seeders lose "patency" and performance on stubble backgrounds. Consequently, when choosing the type of sowing machines for soil-protecting agriculture, and the parameters of the working elements and their arrangement on the frame, the working conditions should be taken into account.

For each zone and type of culture, the optimal timing of sowing was determined. In practice, they vary slightly in terms of years and soil conditions. Therefore, seeders in the soil protection system of agriculture work either on dense dry soils, or in conditions of excessive moistening. In the first case, the working organs of the sowing machines do not penetrate the soil, but in the second - the openers and the rollers will be stuck in the soil. Therefore, when creating sowing machines for soil conservation agriculture, it is necessary to strive to ensure that they are to some extent less "sensitive" to sharp fluctuations in indicators characterizing the physico-mechanical properties of the soil, i.e. the parameters of working elements and the design diagrams of the seeders must ensure the depth of the coulters and efficiency of the seeding machines in general.

Anti-erosion seeding machines are used, as a rule, in steppe arid regions. Therefore, they must not only protect the soil from wind erosion before emergence of sprouts, but also must perform the technological process of sowing in such a way that moisture is rationally used. This necessitates a careful choice of the rational method of sowing, the depth of seeding, the determination of permissible unevenness of feeding them into the openers and between the openers, the method of post-sowing of crops and the degree of compaction of soil. All these issues have been considered by many authors [4,5,6,28,36,48,49,50,51,52,53,54] and are presented in the form of recommendations. Proceeding from the principles and tasks of soil protection agriculture, working conditions and requirements for the technological process of sowing seeds of leguminous crops, the main agrotechnological (initial) requirements for seeding machines are as follows:

- seeding of cereals and legumes on stubble backgrounds should be carried out in a rowwise manner simultaneously with pre-plant cultivation, or without it, row seeding of the sowing and application of granular mineral fertilizers, and with re-equipping of seeders with special openers and rinks - use of strip sowing with continuous packing;
- work should be performed on soils with different moisture content at a moisture content of 35-75% of the maximum field water capacity;
- the sowing apparatus should provide a seeding rate set for each soil-climatic zone with a deviation of not more than ± 3%;
- unevenness of sowing along the seed drill tubes and the instability of seeding should not exceed ± (3-6) %;
- crushing of the sown seeds of grain crops can not exceed 0,3%, and legumes - 1%;
- to seed the seeds to a depth of 4-8sm, with 90% of them should be in the horizon, corresponding to the average specified depth, and two adjacent horizons 0,5 sm;
- to ensure the rate of sowing of granular mineral fertilizers within 50-200 kg/ha with the general instability and unevenness of their sowing by the seed drill tubes not more than ± 10%;

- after passing the seed drills, at least 70% of the crop residues from the original quantity should remain on the surface of the field, and the content of erosion-hazardous fractions (<1 mm) of soil in the 0 - 5 sm layer should not increase against the original one;
- to create the optimal soil density when it is rolled up after sowing in the seeding zone.

The diverse and severe soil and climatic conditions and insufficient supply of moisture in the main grain-producing regions of the United States, Canada, Australia, Russia and Kazakhstan predetermine the need for cultivation of cereals by various technologies, for having numerous models of sowing machines with various types of working organs. Moreover, the development and production of sowing machines in these countries are concentrated in numerous firms, which, as a rule, specialize in certain types of leguminous seeders (Table 11).

For example, in the US, 10 firms are engaged in the development and production of grain seeders, which produce 38 models, respectively in Canada 16 and 63, in Australia 8 and 26, in the CIS countries 5 and 29. At the same time, the main working bodies of grain seeders in these countries are lancet paws and disks. Pointed paws with large spacing allow sowing on stubble backgrounds, and discs - on backgrounds that have no crop residues on the surface of the field. For cultivation of leguminous crops in the conditions of Northern Kazakhstan, seeders-cultivators of the type SZS-6 (SZS-12) were developed and mastered for sowing on stubble backgrounds, for seeding in pairs - a seed drill SZP-3,6, and on wetlands - tiller-seeder machine LDS-6.

In Germany, France and Great Britain, soil and climatic conditions and moisture supply make it possible to obtain a higher yield of wheat 62,5-81,7 s/ha, and sowing is done practically only by seeders with disk working organs.

Table 11 - The main technical characteristics of grain seeders produced by the leading companies of the world's grain-producing countries

Country	The total number of firms that produce seeders, incl. the key ones	Number of produced models of sowing machines	Type of working body and the number of models produced with it	Space between rows, sm	Width of grip, m
USA	10 John Deere, Concord, International Harvester et al.	38	Disk – 19 Narator - 9 Arrow-foot - 5 Chisel - 2 Anchor - 2 Cutter - 1	16,0-30,5 15,2-17,8 15,0-30,5 7,7 17,8-20,3 20,3	2,2-12,6 2,2-15,3 10,7-27,4 7,0-14,0 13,7 2,4
Canada	16 Massseu Ferguson, Morris, Flecxi-Coil, Case et al.	63	Disk - 16 Narator– 8 Arrow-foot - 36 Anchor - 3	15,0-30,5 20,8-30,5 15,0-36,0 18,0-25,0	2,1-14,4 4,1-16,4 3,0-19,6 2,1-3,2
Australia	8 Connor-Shia, John Sherer, Symonds et al.	26	Disk – 8 Narator– 1 Arrow-foot - 16 Blade - 1	15,2-33,0 18,5 12,6-18,0 15,2	4,0-14,2 5,9 3,1-19,5 2,3
France	3 Roger, Huard, HodetGougis	12	Disc – 11 Foot - 1	10,0-22,0 18,0	3,0-8,0 6,0

Germany	3 Amazone, H Weiste, Fortschritt	7	Disc - 6 Blade - 1	11,5-20,0 15,4	3,0-8,0 6,0
United Kingdom	4 Bamlett, Mas- seu-Ferguson, Moore, Hist Stanhay	6	Disc - 6	12,0-17,5	2,2-5,2
CIS countries, incl. Russia and Kazakhstan	5 Red Star, Sibselmash, Belinskelmash, Tselinograd selmash, Kazakhselmash	29	Disk – 14 Narator - 5 Arrow-foot - 8 Cfltnder – 1 Combined - 1	15,0-22,8 12,0-48,0 22,8 15,0 22,8	3,6-18,0 6,0-12,0 2,1-12,3 3,6 14,3

In addition to these requirements, modern technical and operational requirements are also imposed on seeding machines of soil conservation: load the tractor within specified limits; on the preparation of sowing units for long-distance transportation and bringing it back into working condition, the tractor driver must spend a minimum of time and labor; have a low material consumption and high productivity; should be equipped with seeding and seed level control systems in the containers.

One of the ways to increase the productivity of sowing units is the use of a centralized sowing system (CSS). In it, the predetermined seeding rate is provided by centrallized dispenser, mounted on a large-capacity tanks, and distribution to thr coulters is carried out by the airflow. Implementation of this system will simplify the design of seeders, reduce the material consumption and traction resistance, reduce the time for seeding. In the future application CSS will allow to create combined units in which are sealed portion, for example, made of type trailer cultivator, may be used for performing other technological operations agrotechnical deadlines of which do not coincide with the timing of crop.

In cultivation of cereals and legumes on light soils by the soil composition of the soil protection system of agriculture, it is envisaged to use strips of small width. On them in a certain sequence, perennial herbs are placed, for example, grit, steam and grain crops. For sowing on stubble backgrounds of seeds sown with very small norms and prone to the formation of vaults in the vessel, conventional grain-seeding seeders are not adapted. Therefore, there was a need to create a special grass seeder to work on stubble backgrounds.

To implement the above requirements for seeding machines, even having developed all the modifications on one basic model, is now practically impossible. Therefore, the working conditions for soil protection agriculture of seeding machines, the requirements for the technological process of sowing, the seeds of cereals and legumes, practical recommendations on the use of seeders in different soil and climatic zones, etc., have necessitated the creation and use of three basic types of seeders.

1 For the sowing of leguminous crops in the system of soil-protecting agriculture on a lea field clean from weeds and on the surface of which, after a few mechanical treatments, there remains a small amount of crop residues, and in the soil a sufficient amount of moisture is accumulated, it is advisable to use two-disk press drills with a row spacing of 15 sm.

2 To sow the second, third, etc. leguminous crops after lea is needed by seeders-cultivators, performing in a single pass the pre-sowing cultivation of soil, row seeding, introduction of mineral fertilizers and after-sowing pressing of the soil.

Replacing the lancet paws on the tubular racks on the handkerchiefs, pre-sowing treatment can be excluded. Sowing machines for strip sowing were developed and created by replacing the working parts and packing devices, as well as seeders for sowing grasses on stubble backgrounds.

3 Seed-cultivators are used and have effect, as a rule, in drought years. For sowing of cereal crops on wetland soils it is better to use a seed drill, combining pre-sowing tillage, sowing and post-sowing leveling of the field surface.
The technological scheme of sowing seeds by any sowing machine provides for three structural-technological systems, each of which performs in a certain sequence the following functions:
- container for continuous seeding with the seeds with a device for their individual dosing in the openers or one or several dispensers with seed spreaders, as well as seed pipes for supplying seed under the opener;
- plowshare for formation on the necessary (given) depth of the bottom of the furrow and laying seeds on it and sealing them with soil;
- device for seeding seeding or an adaptation for leveling after sowing the field surface.

Considering the working conditions of seeding machines for soil protection agriculture and the requirements for the technological process of sowing seeds by them, let us consider the main provisions for choosing the type and parameters specified above for constructive-technological systems.
The capacities of seeders for seeds and fertilizers should have the optimum capacity, and evenly and continuously supply seeds to dosing devices, regardless of the direction of travel and the inclination of the seeder.

3.2. Determination of the optimum capacity of seeding machines' containers

To determine the capacity of the tank, it is usually recommended [55,56] to use formula

$$E = \frac{L \cdot H_{max} \cdot B_p}{10^4 \cdot \gamma \cdot \eta_c},$$

where: L - the length of furrow carried by the drill from one filling station to another, m; H_{max} - maximum seeding rate, kg/ha; B_p is the working width of the drill, m; γ - density of sown material, kg/m^3; η_c - is the coefficient of filling the capacity of the sown material (≈0,9).
The length of the container for individual dosing of the sowing material is determined by the formula

$$l = a(n+1),$$

where: a - is the width of the aisle; n - is the number of coulters, and the cross-sectional area of the capacitance is $S = E \cdot l$.
In this case, the width of the upper neck and the height of the container is recommended to be chosen depending on its capacity, length, cross-section shape and ease of maintenance. To ensure continuous supply of seeds to the outlets, the front and rear walls of the container are installed at an angle to the base equal to twice the friction angle (φ) of the seed material over their surfaces.
However, these recommended formulas do not allow you to choose the optimum capacity of the container, since it is not clear what length of the gong L should be taken as optimal. If we take it to the maximum, then, naturally, there will be more mass of the material to be sown in the container. This will require more effort to move the machine, there will be less speed of the machine with a given reserve of tractor power, and, consequently, the productivity of the unit will decrease. For small L, the mass of the material in the vessel will be insignificant and it will have to be filled more often, losing more time for stopping the unit, which will also adversely affect its productivity and fuel consumption. Therefore, in recent years, more attention has been paid to optimizing the capacity of seed drills, taking into account its influence not only on the operational performance of the machine, but also on rational use when cutting sheet material.
The paper [55,56] convincingly shows the significant influence of the capacity of the sowing container on the productivity of the seeding unit (W_c) and provides the following formula for its determination

$$W_c = \frac{t_c \tau}{\frac{37(G+\psi E\gamma)\cdot f}{B_p \cdot N_T} + \frac{H t_3}{E\gamma}}, \qquad (110)$$

where: t_c - time of change,h; τ - coefficient of working time use shift; G - weight of seeder, kg; ψ - coefficient of capacity filling, which varies from 1 to 0,05 during operation (0,5 is recommended);

f - coefficient of rolling of the seeding machine under the given conditions (for seeders f ≈ 0,23); $N_т$ - traction power of the tractor at sowing, hp.; t_3 - duration of a single refueling of seeds, h.
However, the sowing of leguminous crops is a low-energy operation, in which the tractor is loaded at 60-75% [57,58]. Therefore, a change in the capacity factor of the tank or a slight increase in its capacity should not significantly affect the performance of the unit. In addition, according to the formula (110), it is difficult to optimize the capacity of combined machines that combine the operations of sowing seeds and applying fertilizers. Therefore, it is recommended to use the following expressions:
for seeds

$$W_c = \left\{\frac{1}{B_р}\left(\frac{10}{\upsilon_р} + \frac{10^4 \cdot t_n}{L}\right) + H_c\left[\frac{1}{E_c \cdot \gamma_c} \times \left(\frac{t_1}{\eta_я n_C} + \frac{E_у \gamma_у}{Q'_3 K}\right) + \frac{1}{Q_3}\right]\right\}^{-1}, \qquad (111)$$

for fertilizers

$$W_т = \left[\frac{10}{B_р}\left(\frac{1}{\upsilon_р} + \frac{10^3 \cdot t_n}{L}\right) + H_у\left(\frac{t_1}{E_у n_c \gamma_у \eta_я} + \frac{1}{Q'_3}\right) + \frac{H_c}{Q_3}\right]^{-1}, \qquad (112)$$

where: t_n - is the time taken for one turn, h; L - is the length of a run, m; H_c, $H_у$ - norm of sowing seeds and fertilizers, kg/ha; $E_c, E_у$ - capacity of containers for seeds and fertilizers, dm^3; γ_c, $\gamma_у$ - density of seeds and fertilizers, kg/dm^3; n_c is the number of seeders in the unit; $\eta_я$ - capacity utilization factor of grain ocontainers; t_1 - preparatory refueling time (tanker approach, maneuver, etc.), h; Q'_3, Q_3 - prductivity of the loader when loading seeds and fertilizers, kg/h; $K = \frac{E_у \gamma_у H_c}{E_c \gamma_c H_у}$ - the ratio of areas sown with seeds and fertilizers after simultaneous refueling.
Dependence of the productivity of a three-reed aggregate (seeder SZP-3,6 or SZ-3,6) on the capacity of the seed container and the speed of motion, found by [57] according to the formula (111), is shown in Fig. 72.

The analysis shows that with the increase in the capacity of the seed compartment, the productivity of the seeding machine increases, and is most noticeable when working at higher speeds, and at low - the effect of the capacity of the grain capacity of the serial seeders is negligible. Thus, if doubling the capacity of cereal capacity in the range of 450-900 dm^3, this will increase the productivity of the unit only from 4,4 to 4,6 ha/h, or by 4%, while operating at speeds of 12 - 15 km/h - by 10 - 11%.
Taking into account the unevenness of the curve W_c, it is advisable to determine the optimal capacity of seed capacity on the basis of those values at which a significant increase in productivity is observed. For conditions where small initial fertilizer rates (0,050-0,150 t/ha) are introduced simultaneously with sowing, the capacity of the standard grain sowing drill is advantageously increased from 450 to 900 dm^3 (up to 230 dm^3 per 1 m of the seeding machine width).

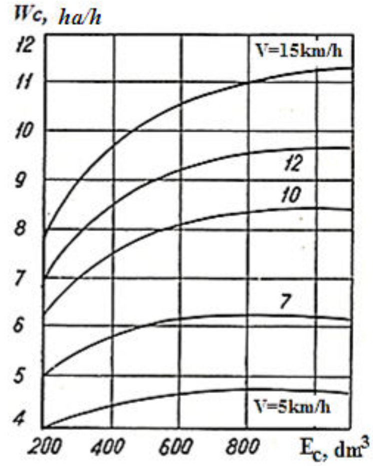


Fig. 72. Productivity of the three-reed unit in dependence
from the speed of its movement and the capacity of the container for seeds

The change in the productivity of the seeding machine, depending on the capacity and the rate of application of fertilizers, found by formula (112), is shown in Fig. 73.
The introduction of fertilizers with large norms simultaneously with the sowing of cereal crops also affects the capacity of the seed tank.
With the increase in the fertilizer application rate, the optimal capacity of the fertilizer hopper also increases, while the capacity of the seed container decreases [57,58]. However, the total capacity of the drill capacities when applying fertilizers with a norm of 0,05-1,25 t/ha is 300-365 dm^3 per 1 m of the seed machine's gripper (Fig. 74).
It is important not only to determine the optimum capacity of the tanks for seeding machines, but also to choose their shape and dimensions in order to use the sheet material rationally for cutting, since sometimes the condition of their minimal surface is not observed [55].Using the method of extremal values of functions, formulas are given in this paper for determining the basic dimensions for the most common forms of capacitances: cylindrical; cylindrical with a truncated cone at the bottom; rectangular, passing in the lower part to the trapezoidal and triangular (Fig. 75).

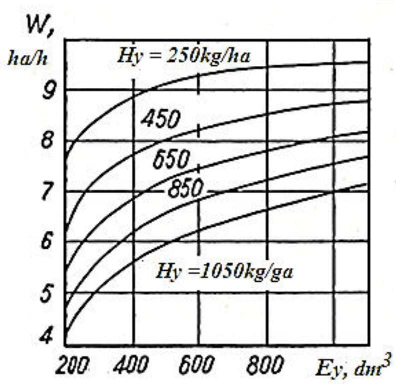


Fig. 73. The capacityof the unit, depending on the capacity of the tank and the rate of application offertilizers

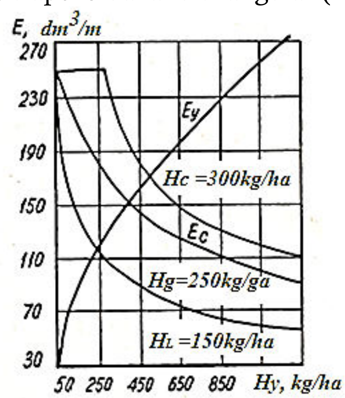


Fig. 74. Optimal capacity of seed and fat capacity, depending on the application rate of fertilizers

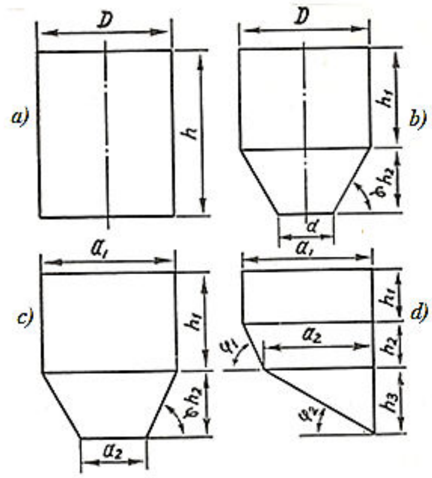


Fig. 75. Forms of seed or fertilizer capacities of seeders:
a) - cylindrical; *b)* - cylindrical with a truncated cone; *c)* - rectangular, passing at the bottom to the trapezoidal;*d)* - a rectangular, passing in the lower part into a triangular

To determine the basic dimensions of these tank shapes, formulas were used for their volume (V) and an expression relating the capacity of the tank (E) to the operational and technological performance of the seeder:

$$E = \frac{WTH}{\eta_я \gamma}\,, \tag{113}$$

where: W - productivity of the seeding machine, ha/h; T- the duration of the machine until the next filling of the material, h.
Equating the expression (113) to the formulas for the volumes of each of the above container shapes, and taking into account the expressions of the areas of their walls, the following analytical dependences were obtained for determining the width, diameter, height and other sizes (see Fig. 75) of tanks [55]:

- of cylindrical shape

$$D = \left(\frac{4WTH}{\pi\eta_я\gamma}\right)^{\frac{1}{3}}, \quad h = D; \tag{114}$$

- of cylindrical shape with a truncated cone

$$D = \left\{4WTH\Big/\left[\pi\eta_я\gamma\left(\frac{1}{2} + \frac{1}{2\cos\varphi} - \frac{1}{3\mathrm{tg}\varphi} + \frac{B_1^3}{6\mathrm{tg}^2\varphi}\right)\right]\right\}^{\frac{1}{2}}, \tag{115}$$

$$d = DB_1\Big/\mathrm{tg}\varphi;\ B_1 = (1-\cos\varphi)\Big/\cos\varphi;$$

- of rectangular shape, passing at the bottom into a trapezoidal one

$$a_1 = \left\{2WTH\Big/\left[\eta_я\gamma l\left(1 + \frac{1}{\cos\varphi} - \frac{\mathrm{tg}\varphi}{2} - \frac{B_3^2}{2\mathrm{tg}\varphi}\right)\right]\right\}^{\frac{1}{2}}, \tag{116}$$

$$a_2 = B_2 a_1\Big/\mathrm{tg}\varphi,\ B_2 = B_1 = (1-\cos\varphi)\Big/\cos\varphi,$$

$$h_1 = (WTH\Big/\eta_я l a_1) - \left[(a_1^2 - a_2^2)\mathrm{tg}\varphi\Big/4a_1\right];$$

- of rectangular shape, passing at the bottom into a triangular one

$$a_1 = \left\{2WTH \Big/ \left[\eta_я \gamma l \left(\frac{1}{\cos\varphi_1} - \frac{B_3^2}{4(\mathrm{tg}\varphi_1 - \mathrm{tg}\varphi_2)}\right)\right]\right\}^{\frac{1}{2}}, \quad (117)$$

$$a_2 = a_1 B_3 \Big/ 2(\mathrm{tg}\varphi_1 - \mathrm{tg}\varphi_2),\;\; B_3 = \frac{1}{\cos\varphi_1} - \frac{1}{\cos\varphi_2} + (\mathrm{tg}\varphi_1 - \mathrm{tg}\varphi_2)\,,$$

$$h_1 = \frac{WTH}{\eta_я \gamma l a_1} - \frac{a_1 - a_2^2}{2a_1} tg\varphi_1 - \frac{a_2^2}{2a_1} tg\varphi_2,$$

where: l - is the length of the container of a rectangular shape, and the designations of the remaining dimensions included in the formulas (114,115,116,117) are clear from Figure 75.
These expressions quite fully define the main sizes of containers, as they include not only the parameters of capacity, but also the main operational and technological indicators. The dimensions of the containers found are also rational from the point of view of the best use of sheet material, since they are obtained from the condition of a minimum of the total surface area oftheir walls.
The above methods of optimizing the capacity, shape and size of containers for seeds and fertilizers can be used both for seed tanks placed on the frames of seeding machines with individual dosing of seeds in each colter, and for centralized tanks with a common metering device for the whole unit or part thereof. The centralization of tanks, the loading of seeds and fat into a single seeder of the same width, without increasing its capacity, can increase productivity by 10,5% and reduce labor costs by 9% [59].

3.3 Metering, distributing and transporting systems of sowing machines

The seeds or fertilizers coming from the vessel through the exit windows must be fed into the colters in a strictly prescribed amount, determined by the seeding rate. These functions carry out metering, conveying and distribution systems.
In soil conservation agriculture on wide-seated seeding machines, two systems are used for dosing, distributing and transporting seeds and corn into the openers: mechanical, using individual coil feeders for each colter; a centralized sowing system (CSS) with a coil-type group feeder and one or two-stage distribution by means of a different type of dividing heads and pneumatic transport to the coulters [43.60].
When choosing the geometric parameters and operation modes of the coil type dispenser, the working volume of the coil is determined for each opener first, providing the seed rate set for 1 hectare, taking into account the physic-mechanical properties of the seeds and the method of seeding them.
The working volume F_0 of the coil is the volume of seeds sown by it in one revolution. The calculated values of the working volume of the coil are determined by the given agrotechnological (initial) requirements by the norm of sowing H (kg/ha), the row spacing a (sm), and also by the ratio $i = n_в \,/\, n_к$ from the axis of the support and drive wheels of the seed drill to the shaft of the sowing units ($n_в и n_к$ - the rotational speed of the shafts of the apparatus and of the drive wheels, respectively) [13,45,46,47]. For one turn of the drive wheel, the planter at a given rate of H should sow (without taking into account the sliding of the wheels)

$$M = \pi \cdot D_k \cdot H \cdot B \cdot 10^{-3}, \text{ g/1 wheel revolution,} \quad (118)$$

where: $B = n \cdot a$- seed drill width, sm; n-number of colters; D_k -diameter of drive wheel, m.
Each device sows the volume of seeds at a known density γ (g/sm³)

$$F_k = \pi \cdot D_k \cdot H \cdot a \cdot 10^{-3}/\gamma, sm^3/1 \text{ wheel revolution,} \quad (119)$$

and taking into account the coefficient of sliding of the drive wheel ε

$$F_k = \pi \cdot D_k \cdot H \cdot a \cdot 10^{-3}/\gamma(1-\varepsilon). \quad (120)$$

For one-wheel revolution, the coil will sow iF_0sm^3 of seeds, i.e. $iF_0 = F$, there where

$$F_0 = \pi \cdot D_k \cdot H \cdot a \cdot 10^{-3}/\gamma(1-\varepsilon). \quad (121)$$

If the seed drill is designed for sowing a number of crops, the seeding rate of which varies from$H_{min}$to H_{max}, then, as follows from formula (121), these norms can be provided either by changing the working volume of the coil at$i$ - const (by changing the working part length of the coil), or by changing the gear ratio i at F_0 - const, or by changing F_0 and i. Since the length of the coil is limited by the arrangement of individual dispensers under the bottom of the seed container, the size of its exit windows and the accepted width of the rows, the possibility of obtaining several gear ratios by rearranging or changing gears and asterisks is provided to increase the number of sown crops in the machine transmission system.
When calculating the size of the coil that provides seed sowing in an amount of F_0, it should be borne in mind that the choice of the value of the transfer ratios i is limited by the linear speed of rotation v_{kat} of the coil, referred to its diameter d. Increasing the density v_{kat} beyond certain limits leads to a deterioration in the quality of sowing - the uniformity of the grain jet is impaired. Practically suitable values for i for grain seeders can be considered$i$= 0,12-1,6. Since $i = n_B/n_k = D_k \cdot v_{kat}/d \cdot v_m$, then for grain seeders

$$v_{kat}/d = (0{,}12 \ldots 1{,}6)v_m/D_k, \quad (122)$$

where: v_m - forward speed of the seeder.
The principle of operation of the coil dispenser has some features. When the coil rotates, a flow occurs that consists of seeds that have fallen into the grooves and seeds located in the space between the coil and the lower or upper walls of the box (active layer) (Figure 76). An active layer can sometimes be located and a fixed layer.

The volume of seeds fed by the coil in one turn is composed of the volume of seeds trapped in the grooves ($F_ж$) and the volume of seeds ejected from the active layer (F_{ak})

$$F_o = F_ж + F_{ak}. \quad (123)$$

If the value of $F_ж$ for a given coil size can be found by calculation (assuming that $F_ж$ is approximately equal to the groove volume), then the definition of the F_{ak} value is complicated by a number of circumstances, and primarily by the fact that the velocity of the seed movement is not constant in the thickness of the layer c, and the form of the function $v_x = t(x)$ can only be determined experimentally for specific conditions. The regularity of the change in the rate of movement of seeds is found from the dependence

$$v_x = v_{kat}\left(1 - \frac{x}{c}\right)^m \quad (124)$$

where:$v_{kat} = \pi d n_в/60$ - exponent, determined by experience.

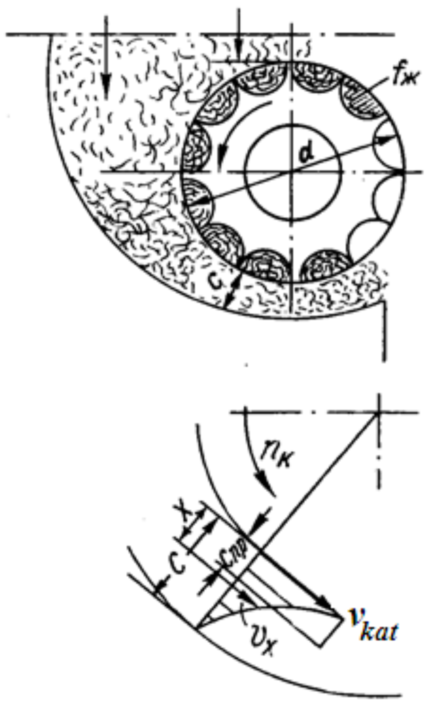


Fig. 76 Diagram of operation of the coil metering device (lower seeding)

Calculation of the working process of the coil-type metering device is usually carried out, guided by the reduced thickness of c_{np} the active layer, which is also determined experimentally. The relation between c and c_{np} is expressed by the following equality

$$c = c_{np}(m + 1). \tag{125}$$

According to experimental data, the values of the indicator m for wheat are 2,6, oats – 2,5, barley – 2,6, millet -1,4. The value of c_{np} varies insignificantly with the change in the length of the working part of the coil and the speed of its rotation: for rye, it varies between 2,2 and 2,5 mm with a change in the working length of the coil from 20 to 30 mm, and for wheat with an increase in the working length of the coil from 5 up to 25 mm decreased from 5,0 to 3,2 mm [13,45].
The volume of seeds caught in the grooves will be

$$F_ж = \beta \cdot f_ж \cdot z \cdot l_p \ , \tag{126}$$

where: $f_ж$ – cross-sectional area of the groove; z— number of grooves;
l_p - working coil length; β - coefficient of filling of grooves.
With the selected coil diameter, the cross-sectional area of the groove $f_ж$ and the number of grooves z are determined by the shape of the groove profile. The most common are the groove profiles (Fig. 77a, b).

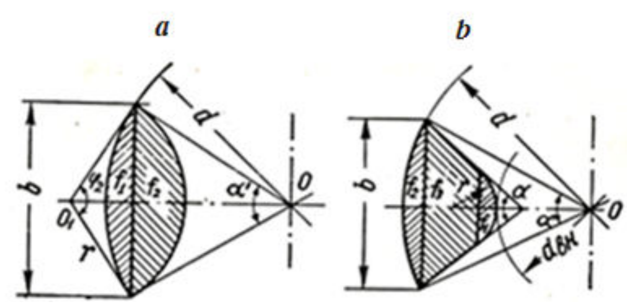


Fig. 77. To calculate the cross-sectional area of the grooves

In the first case

$$f_{ж} = f_1 + f_2 = \frac{d^2}{8}(\alpha' - \sin\alpha') + \frac{r^2}{2}(\varphi_2 - \sin\varphi_2), \quad (127)$$

where: $\alpha' = arcsin\frac{b}{d}$; $\varphi_2 = 2arcsin\frac{b}{2r}$; $b = d\sin\frac{\pi}{z} - \Delta b$;
$\Delta\ b$- thickness of jumper between grooves.

In the second case

$$f_{ж} = f_1 + f_2 + f_3 = \frac{d^2}{8}(\alpha' - \sin\alpha') + \frac{r^2}{2}[\pi - \alpha - \sin(\pi - \alpha)] + \\ + \frac{b^2 - 4r\cos^2\frac{\alpha}{2}}{4\text{tg}\frac{\alpha}{2}}. \quad (128)$$

The grooves must be so large as to provide a uniform grain flow without a noticeable pulsation (usually z = 10-12). Sometimes, to reduce pulsations, the grooves are placed at an angle (usually up to 15°) to the longitudinal axis of the coil.
The filling factor of the grooves depends on the shape and size of the seeds, the profile of the groove, the area of its cross-section. Thus, for a coil with z = 12, with vertical feeding of seeds, β changed from 0,7 (l_p = 8 mm) to 0,85 (l_p = 30 mm) for seeds of wheat, barley, oats, and for seeds of clover and alfalfa β ≅ 0, 9 [13].
The seed volume supplied from the active layer with one coil rotation is found by the formula

$$F_{ak} = \pi \cdot d_k \cdot c_{np} \cdot a \cdot l_p. \quad (129)$$

Consequently, taking into account the $F_{ж}$and F_{ak} values, the volume of seeds emitted by the coil in one revolution is calculated as

$$F_0 = (\beta \cdot f_{ж} \cdot z + \pi \cdot d \cdot c_{пр}) l_{р}. \quad (130)$$

The main geometric parameters of the seed box are determined by the size of the hole in the bottom of the seed container through which the seeds enter the box and the dimensions of the working coil [61]. In Fig. 78 shows the scheme of the box and the letter symbols used in the subsequent exposition: the longitudinal dimension of the input window is c, the height is S, the width at the entrance is K, the width in the working part is p, the dimensions determining the location of the coil are d and e.

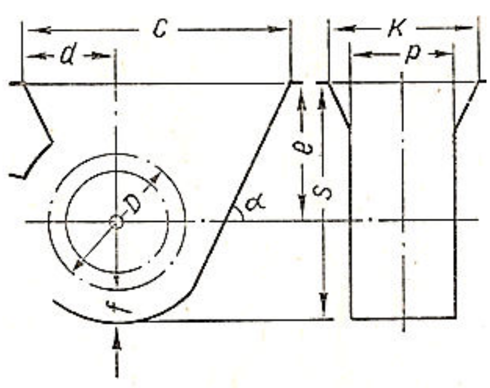


Fig. 78. Seed Box Scheme

The dimensions of the inlet must not be less than the dimensions of the opening in the bottom of the container. Therefore, the size of K should be taken equal to or slightly larger than the diameter of the window in the bottom of the container. The size c is taken considerably larger to create conditions conducive to stable sowing: the seeds filling the box slide along its front inclined wall towards the coil, fill the grooves when feeding them under some pressure. As a result of this feeding, not only the filling of the grooves is ensured, but also the seeds that are outside the direct

influence of the coil's ribs are attracted by the forces of friction. In grain seeders, the size c is usually taken to be approximately equal to twice the diameter of the coil. The size p should be equal to the design length of the working coil. The axis of the coil is located at a distance e from the top edge of the box, approximately equal to the diameter of the coil.
The free space "f" between the bottom of the box and the ribs of the coil for the passage of the seeds carried at the bottom seeding takes more than the thickness of the active layer. At the exit, this gap is somewhat reduced. The distance d of the axis of the coil from the rear edge of the box at the seed drills is 0,60 - 0,65 of the diameter of the coil, and the total height of the box is assumed to be approximately equal to twice the diameter of the coil.
In wide-grained seeders with a centralized sowing system, the sown materials from the tank are fed to a common coil type dispenser and fed into the receiving chamber, picked up by the air flow and fed through the distribution and transport systems to the coulters.
To calculate the coil of a centralized batcher of the coil type (Fig. 79), the above formulas and procedure are applicable to seeders with a centralized sowing system. However, data on the thickness of the reduced active layer, the coefficient of filling of the grooves, the frequency of rotation of the coil, included in these formulas, are obtained for a small diameter of the coil (≈ 59 mm) and an insignificant length of its working part (≈30 mm). Feeding the same large volume of seeds with a single centralized batcher predetermines the need to increase several times the diameter and length of the working part of the coil, the cross-sectional area of its grooves and the speed of rotation. In this regard, data on the filling ratio of the grooves, permissible coil rotation frequency, the degree of seed damage, etc., require experimental refinement. The analysis of the data (Table 12) of the influence of the rotational speed of the coil of the centralized dispenser and the degree of opening of the input window (the ratio of the area of the inlet window over the metering device to the cross-sectional area of the coil along the axis of rotation) by the duty factor indicates that for any degree of opening of the input window the filling factor decreases an increase in the frequency of rotation of the coil [62]. Regardless of the frequency of the coil rotation, the filling factor increases with the opening degree of the input window.

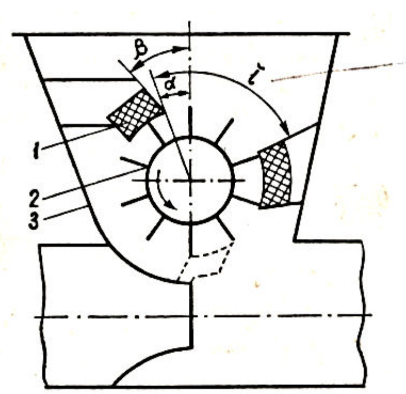


Fig. 79. Arrangement of sealing elements at the coil type dispenser of seeders with centralized sowing system (CSS) 1 - sealing elements(a, β - angles of installation of elements);
2 - coil; 3 - body.

Table 12 - Changing the filling factor of grooves with seeds, depending on the degree of opening of the input window and the frequency of rotation of the coil

Culture	Degree of opening the entrance window	Coefficients of filling at the frequency of coil rotation, s^{-1}				
		0,17	0,30	0,44	0,66	0,81
Wheat	0,62	0,48	0,44	0,43	0,31	0,26
	0,76	0,63	0,57	0,59	0,44	0,41
	0,92	0,83	0,73	0,75	0,63	0,53
	1,10	0,95	0,92	0,87	0,67	0,57
Oats	0,62	0,42	0,32	0,20	0,19	0,17
	0,76	0,51	0,46	0,37	0,27	0,25
	0,92	0,65	0,60	0,51	0,36	0,35
	1,10	0,85	0,80	0,72	0,52	0,45

Studies [63, 64] on the determination of the limiting angular velocity of rotation of the dispenser coil, at which the consistency of seeding per revolution is maintained, with different lengths and filling zones (l = 0,06; 0,08; 0,10 and 0,12 m) show the following. With increasing filling zone length from 0,06 to0,08 m in all cases, the limiting angular velocity of the coil increases. Further increase in the filling zone length to 0,10 m results in an insignificant increase in the limiting angular velocity of the coil only when the pea and barley are sown, and when the wheat and oats are sown, the rate decreases.
An increase in the length of the filling zone to 0,12 m in three crops leads to a decrease in the limiting angular velocity of the coil, and only on pea - to an increase. For all values of the length of the filling zone, the smallest value of the limiting angular velocity of the coil was obtained by sowing oats, with its largest value being 6,28 rad/s at l = 0,08 m. Such a change in the duty cycle with an increase in the rotational speed of the coil of a centralized dispenser can should also affect the stability of the seeding rate, which is confirmed by experimental studies [40,62,63,64]. With the same frequency of rotation, the coil sows different cultures with different norms; when the speed of the coil of the dispenser is increased, the proportionality of the increase in the seeding rate is observed up to a certain limit. For wheat, barley, peas and oats, this limit sets in at a coil rotation frequency of 0,5-0,6 s^{-1} (Fig. 80). With an increase in the translational speed of the seeder, and, consequently, of the frequency of rotation of the coil, the seeding rate will change with a constant length of its working part (Fig. 81).

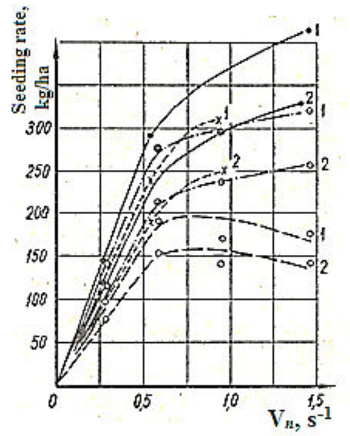


Fig. 80. The norms for sowing wheat seeds
(-), barley (- - -),oats (– –) and peas (- · -) depending on the rotational speed of the coil of the centralized dispenser (coil diameter 115 mm, working length 145 mm, total groove volume 870 sm^3)
1 - υ = 8 km/h; 2 - υ = 10 km/h

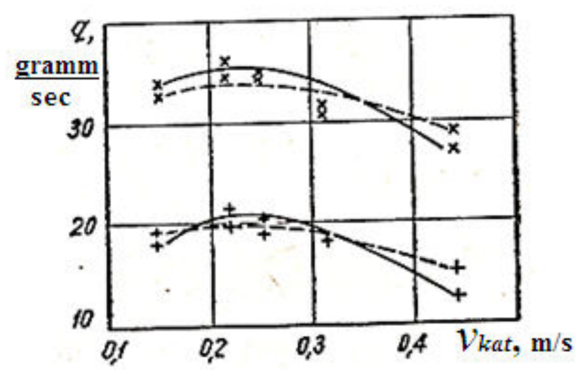


Fig. 81. The capacity of the dispenser, depending on the linear speed of rotation of the coil (diameter 100 mm, working length 30 mm, static pressure in the output window of the dispenser 0,1 kPa): -x- - the barley; - + - -case;– without the use of air flow; - - - - using air flow.

For each sowing culture, there is an optimal rotational speed (angular velocity) of the coil of the centralized batcher (Figure 82). In this case, oats are a culture that limits the limiting angular velocity of the coil. It should not exceed 6,28 rad/s with a coil diameter of 110 mm, a number of grooves 8-10, a working coil length of 100 mm and a filling zone width of 80 mm [63,64].

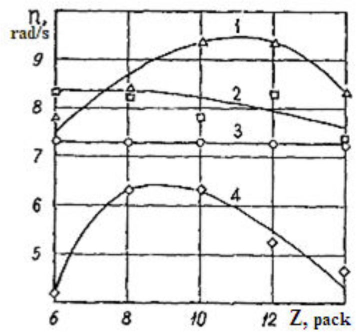


Fig. 82. The limiting angular velocity of the coil of the dispenser, depending on the number of its grooves in the sowing of wheat (1), barley (2), pea (3), oats (4).

A centralized coil type dispenser feeds seeds into the receiving chamber with an ejector device. In the latter, the seeds are captured and transported by the same air flow to the switchgear. If the parameters of the ejector device and the air flow created by the fan have optimal values, then at a minimum energy cost they ensure a high-quality performance of this process. Otherwise, a static head, which sharply reduces the ejection effect, can form in the receiving chamber.
According to [63, 64], the optimum airflow velocity in front of the receiving chamber of the dispenser should be within 23-25 m/s, and the diameter of the ejector confuser is 60-65 mm.
In addition to these data, practically no information is available in the literature on the results of the work to justify the optimum parameters of the ejector device and the air flow for wide-grained grain drills with centralized sowing system. Only in work [64] if the width of the rectangular confuser should be equal to the maximum length of the working part of the dispenser coil, the theoretical height of the window height sufficient for seizing and transporting seeds by the air flow is theoretically justified for such a confuser. For this purpose, dependencies have been proposed for finding the path traversed by seeds in the initial section in the direction of pneumatic transport

$$X = v_n \cdot t - \frac{v_{\text{в}}^2}{2}\left\{ln\left[1 + \frac{(v_n - v_{\text{н}})gt}{v_{\text{в}}^2}\right]\right\}, \tag{131}$$

and the action of gravitational forces

$$Y = \frac{v_{\text{в}}^2}{2}\left[ln\left(\frac{1 + \frac{v_{\text{в}}+v_{\text{ну}}}{v_n - v_{\text{ну}}} + \frac{2g}{v_n}\cdot t}{1 + \frac{v_{\text{в}}+v_{\text{ну}}}{v_n - v_{\text{ну}}}}\right) - \frac{2gt}{v_{\text{в}}}\right] \tag{132}$$

where: v_n - speed of air flow; $v_{\text{в}}$ - speed of seed wandering; $v_{\text{н}}$- seed speed in the direction of transportation; g - acceleration of gravity; $v_{\text{ну}}$ - initial speed of seeds in the direction of the action of gravitational forces.
The dependences $У = f(X)$, constructed according to (131) and (132), represent the trajectories of the seed movement in the air flow in the initial section. Taking into account the length of the receiving chamber -*x*and the height of the confluence window - *y* (Fig. 83), it is possible to determine the airflow velocity flowing from it, sufficient to capture the seeds without dropping them to the bottom of the receiving chamber.

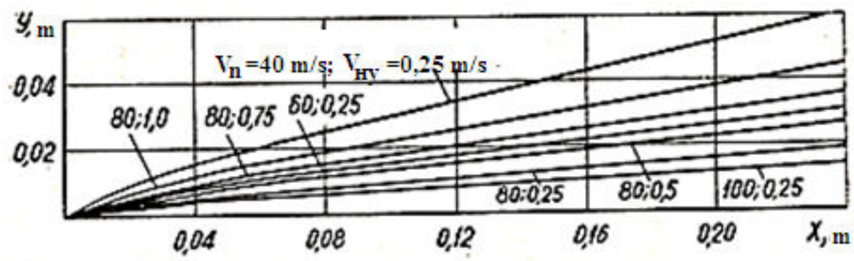


Fig. 83. The trajectory of the seeds in the initial section of pneumatic conveying, depending on the vertical component, the initial velocity of the seeds $\upsilon_{ну}$, and the speed of the air flow $\upsilon_{п}$

A large vertical component of the initial velocity of the seeds entering the airflow requires a larger height of the cross-section of the confuser. Consequently, the seeds need to be introduced into the air stream with a minimum vertical component of the initial velocity, which is ensured when the dispenser operates with a preliminary seed flow in the gap between the coil and the body, which is assumed to be 0,018 m [64]. At the same time, the productivity of the sowing system without additional pressure losses for pneumatic transport increases by 12,5-19,7%.

To determine the parameters of the sowing coil, depending on the parameters of the technological process

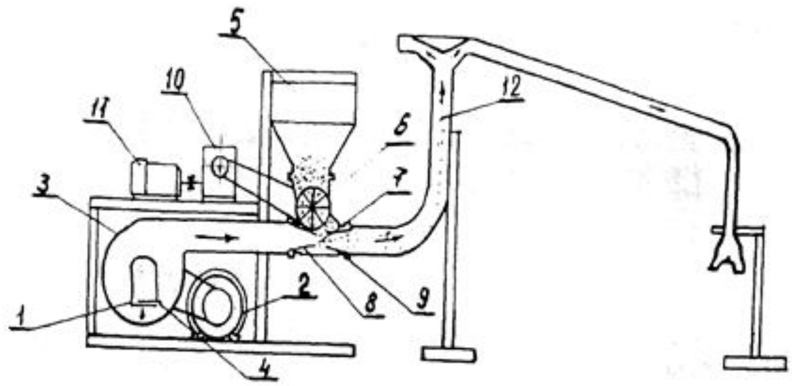
$$[\pi(r^2 - r_0^2) - (r - r_0)B \cdot z]L_k \frac{\omega}{2\pi} = \frac{N\upsilon_c \cdot l \cdot n}{10^4 \cdot \gamma\eta} \qquad (133)$$

where: r - is the radius of the coil; r_0 - radius of hollows of coil grooves; B - is the thickness of the ribs; z - number of grooves; L_k - is the length of the working part of the coil; ω - is the angular velocity of rotation of the coil; N - is the seeding rate; $v_c - \gamma$ - working speed of the seeder; l - row spacing; n - is the number of rows to be seeded; γ — bulk density of seeds; η - is the filling factor of the grooves.

The foregoing has necessitated the experimental refinement of the values of the parameters and operating modes (OKD), the ejector device, and the air flow parameters of the seeding system of seeders with CSS [43]. The studies were carried out on a special installation (Fig. 84).

The fan at the speed of rotation of the impeller $75^{s^{-1}}$ provided an excess air pressure (6,8-7,12) 10^3 Pa, and the air supply - 1277-2282 m^3/h.

The pressure in the receiving chamber was measured at three points: to the right and left of the airflow between the confuser and the diffuser near the lower part of the coil that emerges into the receiving chamber of the dispenser and was registered with a specially developed ten-membrane pressure transducer.



1 - inlet nozzle of the fan; 2 - the electric motor of the fan drive; 3 - fan; 4 - valve;
5 - bunker; 6 - batcher; 7 - the receiving chamber;8 - confuser; 9 - diffuser; 10 - reducer;
11 - drive; 12 - pneumatic conveyingsystem
Fig. 84 Installation for the investigation of the OKD and the receiver of the ejector type

At the first stage, experiments were carried out to select the optimal sizes and operating modes for the OKD - diameter (X_1), the length of the working part (X_2), the number of grooves (X_3), the coil speed (X_4) and the airflow velocity (X_5), and the second stage for the ejector receiver - the diameter of the confuser (Z_1), the diameter of the diffuser (Z_2), the distance between the diffuser and the confuser (Z_3), the length of the diffuser (Z_4), and the length of the confuser (Z_5).
To study the effect of these factors on the operation of the OKD and the receiver of the ejector type, the experimental design method was used [65, 66]. Eight variants of OKD and eight variants of an ejector receiver were made. Numerical values of the lower and upper levels of their parameters and operating modes are taken from the analysis of published sources (Table 13).

Table 13. Conventional designations, name and level of variation of parameters and modes of operation of the OKD and the receiver of the ejector type

Phase of re-search	Symbol of factors	Name of factor	Dimen sion	Variation levels	
				lower	upper
I	$X,$	Coil diameter	mm	100	150
	x_2	Coil length	mm	120	200
	X_3	Number of coil grooves	pc	6	12
	x_4	Coil frequency	s^{-1}	0,35	1,5
	x_5	Air speed	m/s	12,0	41,0
II	$Z,$	Diameter of confuser	mm	36	90
	Z_2	Diameter of diffuser	mm	40	95
	Z_3	Distance between diffuser and confuser	mm	40	80
	Z_4	Length of diffuser	mm	95	130
	Z_5	Length of confuser	mm	150	180

The used half-replicas $2^{5-1}_{y(x)}$и$2^{5-1}_{y(z)}$ were completed to an orthogonal order plan [66, 67], and the investigated processes were modeled by second-order regression equations of the form

$$\hat{y} = b_0 + \sum_{i=1}^{5} b_i\, x_i + \sum_{i>j}^{5} b_{ij}\, x_i x_j + \sum_{i=1}^{5} b_{ii}\, x_i^2 \tag{134}$$

where: x_i , x_j - are the values of the factors;

b_0 - is the free term;
b_i - coefficients of regression of the relevant factors showing the level of influence of one or another of them on the object under study;
b_{iJ} - regression coefficient of the corresponding factors of pair interaction;
b_{ii} - are the coefficients of quadratic effects.

After implementing the experiments on the matrix of the second-order orthogonal plan, the coefficients of the quadratic regression model were determined, and their significance was estimated. Using the F-test Fisher checked and confirmed the adequacy of the equations obtained for real processes [65,66,67]. To study the response surface, the obtained quadratic regression equations for the OKD were reduced to the canonical form and had the following appearance during sowing:

wheat $$\hat{y} - y_s = 0{,}249X_1^2 + 0{,}54X_2^2 + 0{,}443X_3^2 + 1{,}277X_4^2 - 1{,}126X_5^2; \tag{135}$$

barley $$\hat{y} - y_s = 1{,}58X_1^2 - 0{,}18X_2^2 - 0{,}7X_3^2 + 0{,}79X_4^2 + 0{,}65X_5^2; \tag{136}$$

peas $$\hat{y} - y_s = 0{,}197X_1^2 - 0{,}918X_2^2 + 0{,}31X_3^2 - 1{,}25X_4^2 + 0{,}861X_5^2; \tag{137}$$

oats $$\hat{y} - y_s = 0{,}552X_1^2 + 0{,}247X_2^2 + 0{,}302X_3^2 - 0{,}823X_4^2 + 1{,}849X_5^2. \tag{138}$$

In these equations, the eigenvalues of the characteristic polynomials have different signs. Therefore, the response surfaces studied are surfaces of the "minimax" type and have the form of two-band hyperboloids. An investigation of the response surface of this type was carried out using two-dimensional cross sections [66].At the same time, limitations of the a priori information were superimposed on the operating modes of the metering system - the frequency of rotation of the coil should not be more than 0,8 s^{-1}, the air speed should be at least 34-41 m/s. As a result of these studies, the following optimal values of coil parameters were chosen:
- diameter of the coil should be at least 115 mm for sowing seeds of peas, wheat and barley, and for sowing oats - lie within 135-150 mm;
- working length of the coil should vary from 200 to 280 mm;
- the number of coil grooves is 10, and their total volume should lie in the range from $0{,}908\cdot10^{-3}$ to $2{,}77\cdot10^{-3}$ m^3.

Very often, the known methods of searching for an extremum lead outside the permissible region (D), and the most universal method of two-dimensional sections requires, as experience has shown, the optimization of the OKD parameters, a large amount of calculation and graphical work. Therefore, in finding the optimal receiver parameters, the ejector type, we used the optimization algorithm of sliding tolerance [67]. This allowed us to recommend optimum values for the parameters of the intake chamber of the ejector type for each kind of seeds and application rates (Table 14).

Table 14. Optimal values of the parameters of the intake chamber of the ejector type

Parameters, their conditional designation and dimension	Maximum and minimum rates of seed sowing, kg / ha					
	wheat		oats		peas	
	250	100	250	100	300	100
Z_1- diameter of confuser, mm	66	66	90	90	66	66
Z_2- diameter of diffuser, mm	98	98	98	98	98	98
Z_3 - distance between confuser and diffuser, mm	68	68	68	68	68	68
Z_4- length of diffuser, mm	130	130	118	118	130	130
Z_5 - length of confuser, mm	180	180	168	168	180	180

From Table. 14 it follows that for the same parameters of the ejector type receiver, the maximum and minimum rates of sowing of wheat and pea are ensured. To ensure the same rates of sowing oats, due to its physico-mechanical and aerodynamic properties, the parameters of the ejector receiver must be different. In this case, there are two possible solutions: the receiver of the ejector type on the tank must be removable, replaceable; change the shape of the cross-sections of the confuser and diffuser, which affects the degree of ejection [68].

In seed drills with a pneumatic sowing system, the air-seed mixture of a certain concentration flows from the intake chamber through pipelines to distribution and transport devices.

The velocity of the air carrier (V), corresponding to the beginning of the transportation of the grain in the suspended state, is usually called the minimum speed of sustainable transportation (U). To maintain a steady flow of a two-component flow in a pipe of a certain length, it is necessary to create a differential pressure (ΔP), which is equal to the sum of the pressure losses due to the friction of the moving air about the pipeline wall ($\Delta P_в$) and the presence of solid particles (ΔP_m)

$$\Delta P = \Delta P_в + \Delta P_m . \qquad (139)$$

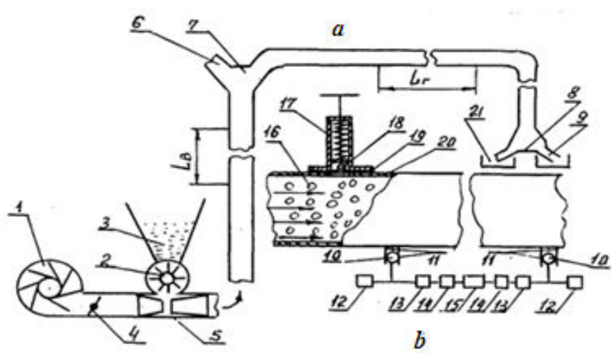
Knowing these components and the speed of sustainable transportation, you can not only calculate the required power for the fan drive, but also the minimum (critical) velocity of the carrier of the medium [69]

$$V = \Delta P / \mu \rho U, \qquad (140)$$

where: μ - mass flow rate concentration of the material in the flow (concentration coefficient); ρ – density of air.

It is generally accepted [69] that it is impossible to establish a general regularity of the boundary of the speed of sustainable transportation, regardless of the properties of the seeds, the concentration of the airsense stream, the direction of transportation, the number of outflowing streams from the distribution devices, their orientation and branching. In this regard, for the experimental studies to determine the steady speed of transportation and pressure losses in pneumatic lines, an installation was developed using a two-stage distribution system according to the 1x8x9 scheme, i.e. on 72 openers (Fig. 85a).

The air-seed flow along the vertical pipeline Ø120 mm was supplied to the switchgear (7) of the first stage, divided into four outlet nozzles (6). Subsequently, flows through horizontal pipelines Ø67 mm came to distributors (8) of the second stage, and then, after dividing by 9 streams, the seeds through the Ø32 mm (9) vasicle ducts entered the collection tanks (21). Variations in the speed of the air carrier were made by changing the position of the flap (4).



a - diagram of the installation; b - diagram of the device for measuring the velocity of grain flow in material pipes

Fig. 85. The installation for the investigation of the process of seed pneumotransportation

For each selected position of the shutter, a dynamic pressure (H_d) was measured at several points along the vertical cross-sectional area along the vertical ($L_в$) and horizontal ($L_г$) sections of the pipeline. Then the formula$V = 1{,}29\sqrt{H_d}$ determined the speed of the air flow (the carrier medium). Pressure measurement in mm of alcohol column was made by cup micromanometer with the help of Pitot tube at distances of more than 7- 8 pipe diameters from the beginning of the vertical or horizontal section, i.e. at steady speed of an air stream.

To measure the speed of movement of seeds in the air flow, a radiometric method was used (Fig. 85*b*). From the polymer material, pea and wheat models (18) were made that were close in shape, mass, and speed to natural seed samples. They inserted a source of ionizing radiation of the necessary activity and intensity. The velocity of the seeds was determined from the distance traveled and the time of flight of the charged seed, which was introduced into the air-seed stream by a special device (17) during the steady-state regime.

At a distance of 7-8 pipe diameters from the hole intended for inserting the grain model into the flow, the first curtain is mounted on the outer wall of the pipeline, and at a certain distance from it the second shutter (11) for the screening of the detectors (10), powered by high-voltage units (12). The distance between the detectors was taken at 1,29m for the vertical pipeline Ø120 mm and 2,7m for the horizontal pipeline Ø 67mm.

When passing by the first detector (10), the radioactive radiation from the model falls on it and is converted into electric pulses arriving at the input of the first integral discriminator (13) and further into the first intensimeter (14). It produces one signal from which an electronic time meter is triggered (15). When the model passes by the second detector (10) and the second intensifier (14), a signal is also generated, stopping the time meter. Knowing the distance between the detectors and the time of movement of the labeled grain particle, its mean velocity in the air-seed flow is determined.

The loss of pressure of the carrier stream during the transportation of the seeds on the vertical (1,0 m) and horizontal (2,5 m) pipeline sections was determined by measuring the static pressure drop when passing a clean air stream first, followed by an air-seed mixture. The pressure was measured with a cup micromanometer. The studies were carried out using pea seeds at seeding rates of 100, 200 and 300 kg/ha and wheat at seeding rates of 110, 180 and 250 kg/ha. In the air duct leading to the metering device on the vertical section of the pipeline, the speed of the air-bearing medium varied from 15,9 to 31 m/s, and in each of the four horizontal sections - from 8,8 to 22 m/s. At the same time, the air flow varied in the vertical section from 528 to 1180 m^3/h, and on horizontal level - from 114 to 291 m^3/h.

With known seconds of supply of air mass, seeding rates and the speed of movement of the unit (V_a = 7,2 km/h), the values of the concentration coefficients in the vertical section of the pipeline at the velocity of the carrier medium V = 15,9-31,0 m/s varied in limits 0,32-1,68 at sowing of peas, and at sowing of wheat - 0,36-1,32. On horizontal sections of pipelines, the concentration coefficients for sowing peas at the velocity of the carrier medium V = 10,9-22,0 m/s varied from 0,36-1,95, and when the wheat was sown – 0,40-1,47 (see Table 15).

Table 15. The speed of transportation of pea seeds ($V_г$) and wheat (V_n), concentration coefficients ($\mu_г$ and μ_n) at different seeding rates and flow velocities (V)

Pipe line	Carrier speed flow (V), m/s	Peas			Wheat		
		Transport speed and concentration coefficients ($V_r/\mu_г$) at seeding rates (kg/ha)			Transport speeds and concentration factors (V_n/μ_n) at seeding rates (kg/ha)		
		100	200	300	110	180	250
Vertical 120 mm	31,0	9,56/0,32	8,10/0,64	7,84/0,96	13,68/0,3	12,25/0,5	11,90/0,
	29,0	9,00/0,34	7,59/0,68	7,27/1,02	13,20/0,3	11,37/0,6	11,06/0,
	27,0	8,70/0,37	6,93/0,72	6,65/1,09	12,7/0,41	10,20/0,6	9,74/0,9
	25,0	7,06/0,40	6,50/0,80	6,00/1,20	11,7/0,44	8,90/0,72	8,64/1,0
	21,5	5,58/0,43	4,80/0,86	4,50/1,29	9,80/0,51	8,40/0,84	8,18/1,1
	20,0	4,81/0,48	4,26/0,96	4,03/1,44	8,40/0,55	7,60/0,90	7,50/1,2
	19,0	-	-	3,09/1,68	7,0/0,58	6,50/0,95	6,26/1,3
	18,0	3,94/0,58	3,46/1,16	*)	-	5,56/1,00	*)
	16,4	2,59/0,61	2,98/1,22	*)	5,20/0,67	*)	*)
	15,9	2,10/0,63	*)	*)	4,70/0,69	*)	*)

Horizon-tal 67 mm	22,0	8,53/0,36	7,50/0,72	6,85/1,08	11,22/0,4	9,60/0,66	9,05/0,9
	20,2	8,20/0,40	7,15/0,80	6,42/1,20	10,70/0,4	9,0/0,72	8,92/1,0
	18,6	7,06/0,43	6,0/0,86	5,46/1,29	9,87/0,48	8,26/0,78	8,06/1,0
	17,1	6,43/0,47	5,71/0,94	5,10/1,41	9,30/0,52	7,45/0,85	7,46/1,1
	14,7	5,30/0,51	4,98/1,02	3,96/1,53	8,70/0,56	6,45/0,92	6,58/1,2
	13,6	4,80/0,59	4,10/1,18	3,29/1,77	7,60/0,65	4,92/1,06	2,77/1,4
	12,3	-	-	2,23/1,95	5,80/0,72	3,36/1,18	*)
	11,7	4,02/0,69	3,31/1,38	*)	-	2,66/1,24	*)
	11,2	3,44/0,72	2,32/1,44	*)	4,50/0,79	*)	*)
	10,9	2,98/0,74	*)	*)	3,40/0,81	*)	*)

*) - sustainable transportation stops.

Analysis of the data of Table 15 shows that in the vertical section of the pipeline the transportation of peas at the maximum seeding rate ceases at a velocity of the carrier medium V <19 m/s and $\mu_г$> 1,68, and when sowing the average norm V <16,4 m/s and $\mu_г$> 1, 22. In both cases, the speed of the pea in the flow during steady transportation was approximately $V_г$> 3,0 m/s. The transportation of wheat at the maximum seeding rate ceases at a velocity of the carrier medium V <19 m/s and μ_n> 1,32, and when sowing the average norm V <16,4 m/s, and μ_n> 1.0. The speed of movement of wheat seeds in a stream with stable transportation for these rates of sowing of wheat lies within V_n> 5,56-6,26 m/s.

On horizontal sections of pipelines, the transportation of peas at the maximum seeding rate ceases at V <12,3 m/s and $\mu_г$> 1,95, and when sowing the average at V <11,2 m/s and $\mu_г$> 1,44. The speed of movement of peas in the flow with stable transportation for these rates of sowing $V_г$> 2,3 m/s. The transportation of wheat at the maximum seeding rate ceases at V <13,6 m/s and μ_n> 1,47, and at an average seeding rate, respectively, V <11,7 m/s and μ_n> 1,24. The speed of movement of wheat in the flow with stable transportation for these seeding rates V_n> 2,7 m/s.

On the vertical section of the pipeline, with an increase in the velocity of the carrier flow from 15,9 to 31 m/s and the seeding rates of pea and wheat seeds, the total specific pressure loss (ΔP) increases by 1,5-2 times (per 1 m of path) (Table 16). In the event that the pressure losses $\Delta P_г$ and ΔP_n make up 40 - 50% of the total pressure loss ΔP, the stable transportation of seeds is violated. With an increase in the velocity of the carrier medium from 15,9 to 31 m/s, the pressure losses $\Delta P_г$ and ΔP_n, regardless of seed seeding rates, have a minimum value at V = 21,5-27,0 m/s, i.e. in this case, sustainable transportation of seeds is carried out with minimal energy costs.

Table 16. The total pressure loss in the grain-air flow (ΔP) and due to the presence of pea seeds ($\Delta P_г$) and wheat (ΔP_n) at different speeds of the carrier medium and seeding rates

Pipe line	The speed of the carrier medium, *(V)*, m/s	Peas			Wheat		
		The specific pressure loss of $\Delta P/\Delta P_г$(Pa) at seeding rates (kg/ha)			The specific pressure loss of $\Delta P/\Delta P_г$(Pa) at seeding rates (kg/ha)		
		100	200	300	110	180	250
Vertical, 120 mm L = 1,0 m	31,0	248,6/30,	288/58,6	303/81,3	252,6/13,8	276/42.	288,6/60,
	29,0	230/21,7	263/48,3	286/70,5	243,5/10,4	257,4/3	271,4/46,
	21,0	204,5/16,	243/40,0	267/58	212/6,2	230,3/2	248,8/35,
	25,0	182,6/15,	221/37,4	247,3/49,	187/3,6	213/29,	228/33,6
	21,5	145/16	183,2/46,	216/73,2	167,8/6,8	192/31	205,2/39,
	20,0	125,4/19,	170,4/53,	204,7/83,	137/6,0	163,5/4	179,3/54,
	19,0	120/26,7	–	212,3/97,	126/10,8	149/47,	188/74,9
	18,0	–	162,4/68,	*)	–	152,3/5	*)
	16,4	116,3/35,	166/82,7	*)	*)	*)	*)
	15,9	117,2/39,	*)	*)	*)	*)	*)
Hori-zon-tal 67 mm L = 2,5 m	22,0	178,6/1,6	182,4/4,3	201,4/12,	171,5/1,0	175,9/2,	183,6/6,4
	20,2	158,7/2,7	164/6,0	187/16	152,5/1,5	155/3,7	162/7,2
	18,6	143/4,4	148/8,0	172,8/19	135/1,5	140,1/4,	161,2/9,1
	17,1	127,1/3,1	135,1/12,	161,4/21,	119,5/1,9	125,4/5,	139/10,4
	14,7	98,5/4,2	107,4/14,	137,4/28,	105,9/2,6	113/6,3	122,1/13,
	13,6	88/6,7	101,1/18,	127,8/33,	83,8/4,8	89,3/9,3	99/17
	12,3	·	·	132/40,6	71,4/6,9	79/14	104/23,4
	11,7	74,7/7,0	87,7/20,4	*)	-	81,5/18,	*)
	11,2	72/16,4	89/35,7	*)	66,5/11,0	*)	*)
	10,9	72,8/20,8	*)	*)	68/14	*)	*)

*) - sustainable transportation stops.

With an increase in the velocity of the carrier medium from 10,9 to 22 m/s and the rates of seed sowing in the horizontal section of the pipeline (see Table 16), the losses of the total specific pressure (at 2,5 m) increase by 1,5-2,5 times and most intensively at lower seeding rates. In the event that the fraction of pressure loss $\Delta P_г$ and ΔP_n is 22-40% of the total pressure loss, the process of seed transportation ceases. Such ratios $\Delta P_г/\Delta P$and $\Delta P_n/\Delta P$ regardless of the seeding rates were obtained at a carrier flow velocity of less than 13,6 m/s.

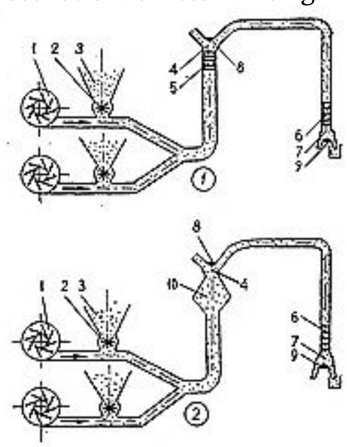
Thus, with the accepted most common seed distribution scheme for a seeder with a 1x8x9 CSS and pipeline parameters, the values of the critical velocities of the carrier medium (U) at which seed transportation ceases, in a vertical section of 120 mm will be within the range of 16-19 m/s, and on horizontal sections 67 mm within 11-12 m/s.

To ensure the specified rates of seeding of peas and wheat with minimal energy inputs for the distribution and transportation of seeds to the plowshares, the optimal velocity of the carrier medium in the vertical section of the pipeline should lie within the limits of 21,5-27,0 m/s, and in horizontal sections of pipelines 17-19 m/s.

From the ejector, a seed flow of a certain concentration flows through pipelines to distribution heads, in which the seeds are divided by their outlet nozzles and through connecting pipelines are fed either directly to the coulters (single-stage distribution) or to the distribution heads of the second stage (two-stage distribution). By experiments and practice it is established that at 30 - 36 openers in a seeder unit, a single-stage seed distribution is usually used, and with a larger number of openers it is advisable to use a two-stage distribution [70].

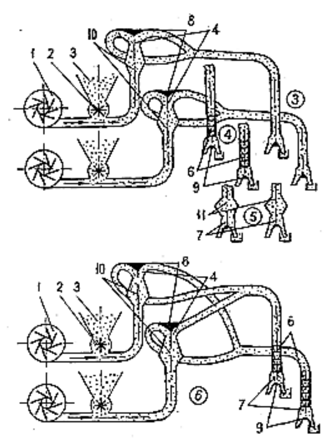
The two-stage seed distribution can be performed according to two schemes: 1x n x m (Fig. 86), or $2\text{x}\frac{n}{2}xm$(Fig. 87), where the first number means the number of heads of the first stage of distribution; n - is the number of output nozzles of the heads of the first stage of distribution, equal to the number of the dividing heads of the second stage; m - is the number of outlet nozzles at the heads of the second stage of distribution.

For the distribution of seeds according to the scheme 1x7x9 (n = 7, m = 9), studies were carried out to select the shape of the distribution head of the CSS and to justify the parameters of the pneumatic network [71]. As a result of the analysis of the flow patterns in the pilot versions of the distribution heads, the most rational model was chosen, for which a steady flow of the stream, the absence of vortices and stagnant zones are characteristic.



1 - fan; 2 - batcher; 3 - container for seeds; 4, 7 – distributive heads of the 1st and 2nd stage of distribution; 5, 6 - conical rings in front of heads of the 1st and 2nd stage of distribution; 8, 9 - divisible cone heads; 10 - expansion chamber

Fig. 86. Variants of seed distribution according to the scheme 1x8x9,



1 - fan; 2 - batcher; 3 - container for seeds; 4,7 - distributive heads of 1st and the 2nd stage of distribution; 5, 6 - conical rings before the heads of the 2nd stage of distribution; 8, 9 - separating cones of heads; 10, 11 –expan sion chambersbefore the heads of the 1st and 2nd stage of distribution

Fig. 87. Variants of seed distribution according to the scheme 2x8/2x9

As a criterion for optimizing the parameters of the pneumatic system of the CSS, the specific energy consumption of seeding and the actual second seed supply were taken [72,73]. The specific energy intensity of the process of seed transportation and distribution depends on the pressure in the P_{sv} system and the air-seed mixture flow concentration, which is characterized by a coefficient μ equal to the ratio of the second air supply to the second seed supply. The pressure loss in the pneumatic network of the centralized sowing systemP_{sv} (Fig. 88) increases with increasing μ, and the specific energy intensity decreases. Minimum power consumption N_{yd} corresponds to μ = 2,0-2,5. With a further increase in μ, precipitation of the sown material in the connecting pipelines is observed.

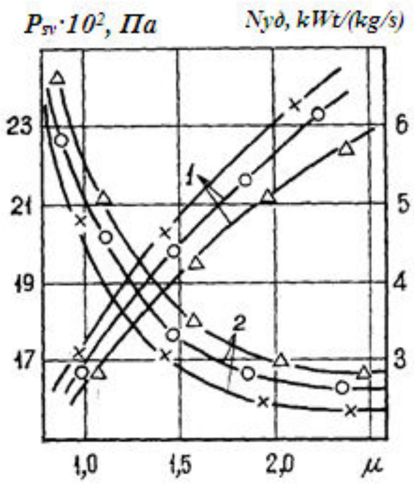


Fig. 88. Pressure loss P_{sv} (1) and specific energy consumption of seeding N_{yd}(2) depending on the air-seed mixture concentration:
– Δ – – oats; – o – – peas; –x– – wheat

Therefore, the following values of the centralized sowing system pneumatic network parameters are recommended:
- the diameter of the air ducts from the fan to the receiving chamber of the dispenser should lie within the limits of 100 - 125 mm;
- the diameter of the pipeline of the distributor of the first stage according to the scheme 1 x n x m should be equal to 180 mm;
- diameter of pipelines between the heads of the first and second stages of distribution - 65 - 75 mm;
- the diameter of the ovules to the openers - 32 mm.

In order to increase the uniformity of seed distribution in the cross section of the pipelines, 5-6 conical rings with an internal diameter of 160 mm with a spacing of 25-30 mm are set before the pneumatic head of the first stage of distribution, and rings with an internal diameter of 55-60 mm and before the heads of the second stage of distribution 4 the distance between them is 15-20 mm. The slope of the generatrix of the conical surface of the rings is recommended to be equal to π/4.

It should be noted that the pressure loss and velocity in the pneumatic system of the CSS are determined not only by the concentration of the air-seed flow, but also by the adopted distribution scheme and its parameters. Thus, for example, according to the data of [40], when the seeds are distributed according to the 1x8x9 scheme (see Fig. 86, variant 2), significant losses of airflow pressure and a drop in its velocity are noted (Table 17) in the heads of the first and second stages of distribution. Consequently, the adopted parameters of the dividing heads can not be fully considered optimal, although they are provided with an allowable unevenness in dividing the seeds and transporting them to the openers.

Using the recommended parameters of the distributor heads and the pneumatic network, experimental studies were carried out to compare the two patterns of seed distribution (see Figure 86,87) for 72 openers (1x8x9, 2x8 / 2x9) and various devices for improving seed distribution with pneumatic heads [40].

Table 17. Changing the pressure and speed of the airflow of the centralized sowing system pneumatic system (distribution according to the scheme 1x8x9)

Place of measurement	Total cross-sectional area of the pipeline at the measuring points, sm^2	Dynamic airflow pressure, Pa	Air speed, m/s
At the output of the fan	175,5	2070,0	58,20
Before the expansion chambers of the heads of the 1st stage of distribution	190,0	735,4	34,66
At the output of the expansion chamber	190,0	723,3	34,34
At the outlet of the nozzles of the heads of the 1st stage of distribution	312,6	257,2	20,28
Before the expansion chambers of the heads of the 2nd stage of distribution	308,0	247,6	20,16

At the output of the expansion chambers of the heads of the 2nd stage of distribution	308,0	220,5	18,99
At the outlet of the nozzles of the heads of the 2nd stage of distribution	410,4	149,9	15,63

The installation options, assembled according to the scheme 1x8x9, are as follows (see Fig.86):
1 - before the heads of the 1 st and 2 nd stage of distribution there are conical rings;
2 - before the head of the 1st stage of distribution there is an expansion chamber, and before the heads of the second stage of distribution - conical rings.
The variants of the installation, assembled according to the scheme 2x8/2x9 are as follows (see Fig. 87);
3-before the heads of the 1st stage of distribution, expansion chambers are installed, and in front of the heads of the second stage straight tubes, the length of which is equal to the height of the expansion chambers;
4-differs from 3 by installing the heads of the 2nd stage of distribution of conical rings in the supply pipelines; bends from one head of the first stage of distribution;
5 - before the heads of the second stage of distribution, expansion chambers are installed;
6-differs from 4 in that the doubling of the bends from the heads of the second stage of distribution was carried out not from one head, but from two.
The main results of a comparative evaluation of various devices for leveling the air-seed flow density in the cross sections of pipelines in front of the dividing heads are given in Table 18.

Table 18. The results of the evaluation of the effectiveness of the installation of rings in the supply pipeline and the radial movement of the distribution cone in the head housing

Culture	Sowing rate, t/ha	The coefficient of variation, %			
		conduit with conical rings		piping without conical rings	
		the cone is centered	optimal cone position	the cone is centered	optimal cone position
Wheat	0,110	6,50	2,70	6,70	1,80
	0,180	4,40	2,10	4,40	1,90
	0,250	4,00	3,20	3,70	3,20
Barley	0,110	5,50	2,00	5,50	1,80
	0,180	3,80	2,20	3,50	1,80
	0,250	2,90	1,90	2,60	2,40
Oats	0,110	11,80	4,60	9,60	1,30
	0,180	13,40	5,70	13,50	4,50
	0,250	13,20	5,30	12,90	7,80
Peas	0,100	10,00	7,50	7,00	5,40
	0,200	9,50	5,00	10,20	4,00
	0,300	7,50	3,30	7,50	3,90
Millet	0,010	12,90	6,40	9,30	10,00
	0,030	10,10	9,60	8,50	8,10

An analysis of the data of this table shows that the installation of conical rings in the supply line practically does not improve the distribution of seeds along the head outlets. More effective in these experiments was the radial displacement of the distributive cone inside the head housing, which enabled in all experiments to reduce the coefficients of variation. However, this adjustment is very laborious and practically impossible in production conditions.
Another method of improving the uniformity of seed distribution by the head of the first stage of distribution is the doubling of its branches. The effectiveness of this method was tested when the seeds were distributed according to the 2×8/2×9 scheme (variants 4, 6). The doubling of the outlet bends was carried out through 90°, 135° and 180° only for each head of the first stage of the distribution separately (variant 4). Doubling the output branch pipes from two adjacent heads did not have a significant effect (variant 6).

To equalize the density of seed distribution over the cross-sectional area ofthe pipelines, it was proposed to install expansion chambers in front of the heads of the first and second distribution stages. Studies [40] have found that the rational form of the expansion chamber is two truncated cones, connected by large bases. The smaller diameters of the truncated cones should be equal to the diameters of the supply and discharge pipelines. For heads of the first stage of distribution, the rational form of the expansion chamber is conical asymmetric, and its cone with a higher height should be installed in the direction of the distribution head. The rational form of the expansion chamber for pipelines of the second stage of distribution is a conical symmetrical one. The recommended values for the parameters of the expansion chambers are shown in Fig. 89.

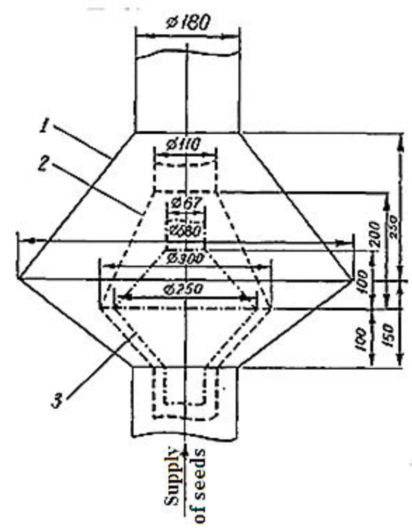


Fig. 89. Recommended parameters of expansion chambers for different diametersof material pipes and seed distribution schemes:
1 - for a pipe with a diameterof 180 mm (1st stage of distribution according to the scheme 1x8x9);
2 - for a pipeline with a diameter of 110 mm (1st stage of distribution accordingto scheme2x8/2x9);
3 - for pipelines with a diameter of 67 mm (for heads of the 2nd stage of distribution)

The doubling of the taps of the heads of the first stage of distribution among themselves, which is possible with the distribution of seeds according to the scheme 1x*n*x*m*, and the installation of expansion chambers with the recommended parameters before the heads of the 1st and 2nd stages makes it possible to reduce in the laboratory conditions the uneven distribution of seeds across 72 openers of the wide-spread seeder (Table 19).

Table 19. Uneven seeding in the openers with tools to improve seed distribution under different distribution schemes

Culture	Sowing rate, t/ha	The coefficient of variation, %			
		Scheme 1x8x9		Scheme 2x8 / 2x9	
		variant 1	variant 3	variant 4	variant 5
Wheat	0,110	1,52	2,53	5,43	1,67
	0,180	3,46	2,19	5,23	1,73
	0,250	3,01	2,58	6,38	1,50
Barley	0,110	3,88	2,52	5,74	3,67
	0,180	4,04	2,83	7,24	2,72
	0,250	3,98	4,34	7,74	2,36
Peas	0,100	6,24	7,13	3,86	4,18
	0,200	4,00	7,34	1,99	1,67
	0,300	4,20	6,15	2,76	1,99

Oats	0,100	9,03	9,90	10,04	1,66
	0,180	10,83	15,43	7,24	2,63
	0,250	9,26	17,67	12,54	4,22

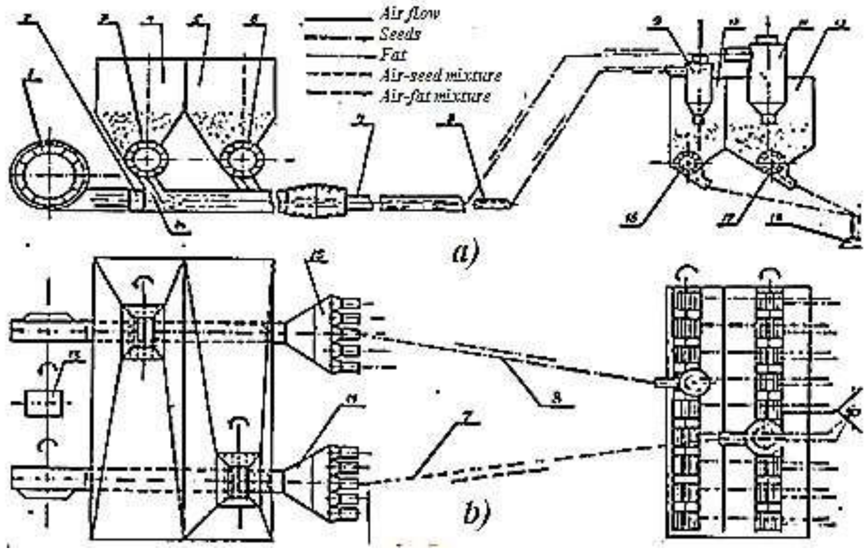
The length of the pipelines from the distribution heads of the 2nd stage to the openers, for understandable design reasons, will be different - than the colter farther from the distributor head, the longer the pipeline. This increases the uneven distribution of the seed material across the openers, since more will enter the openers with shorter pipelines, since the pressure loss in them of the supporting medium is less.
During the movement of the unit along the unevenness of the field surface, the frames of the seeders oscillate in different planes. The pipes leading multicomponent mixture to distributing heads, having internal diameters from 70 to 180 mm and rigidly connected to the seeder frame, deviate from the vertical position, increasing the unevenness of the flow distribution over the area of their cross-section. In order to reduce the negative effect of the deviation of the central pipe from the vertical on the uniformity of the distribution of the sown material over the branches of the chord heads, it is recommended [68] to use a horizontal high-loaded distributor in which the sown material supplied by the airflow is divided by the edge of the partition into two parts. In the first part, performed on the principle of the diffuser, a preliminary distribution of the flow takes place.
Reflecting from the front plates, the air-seed-fat mixture enters the upper and lower plates of the second part, made on the principle of confusor, and, coming off them, the flows are mixed and fed to the outflow pipelines.
The rates of sowing of cereals and leguminous crops and fats lie in the range of 50-300 kg/ha. Pneumatic transport of such quantity of material is carried out by an air stream with speed not less than 15-30 m/s. With such a quantity of feed material and air flow velocity values, the concentration coefficient of even a two-component mixture flow is in the range 0,32-1,95. In this case, in a two-component flow in a horizontal material conduit of a horizontal divider, the solid particles move along its lower part of the cross-sectional area [43, 69]. Therefore, the edge of the partition, which is in the center of the supply pipe of the horizontal divider, can not divide the flow of the mixture evenly, which adversely affects the uniformity of the material distribution at the outlet nozzles. At the same time, as mentioned above, its share in the uneven distribution of the sown materials between the openers will make a significant difference 1,5-2,0 times the length of the pneumatic lines due to the position of the seed coulters with respect to each of the heads of the second stage of distribution, as well as the different values of the losses in their pressure.
It is practically impossible to eliminate the above-mentioned drawbacks, especially when the seeders are working with the centralized sowing system in the field, since it is necessary, depending on the velocity of the carrier stream, the type of seeds and the rates of their seeding, etc. constantly changing the length and diameters of the air lines during operation.
It is for these reasons that, in the field, wide-seated seeders with centralized sowing system, seemingly with valid parameters and operation modes of the sowing systems, do not provide an allowable (± 6%) uneven seed distribution between the openers (Table 20).

Table 20. Uneven distribution of seeds in open areas in field conditions produced by different firms sowing machines

Indicators	Sowing machines and crops								
	"Connor Shia"			"Flexi-Coll"			C3C-14		
	wheat	barley	oats	wheat	barley	oats	wheat	barley	oats
1	2	3	4	5	6	7	8	9	10
Minimum seeding rate, kg/ha installation rate-actual	50,0 48,8	- -	80,0 79,2	50,0 23,1	50,0 51,0	50,0 49,3	50,0 52,7	50,0 50,4	50,0 47,7
Uneven distribution of seed between openers, %	7,4		7,6	10,91	10,37	19,13	10,0	12,7	14,0
Average seeding rate, kg / ha nstallation rate - actual	130,0 134,4	- -	130,0 138,0	130,0 132,6	150,0 151,0	130,0 132,0	130,0 125,0	150,0 142,0	130,0 123,2
Uneven distribution of seed between openers, %	4,7		7,8	10,45	11,78	21,29	11,3	10,0	13,5
Maximum seeding rate, kg/ha: Installation rate - actual	250,0 200,0	- -	250,0 160,0	250,0 252,1	250,0 248,5	250,0 173,0	250,0 245,0	250,0 236,8	250,0 202,0
Uneven distribution of seed between openers, %	5,8	-	7,2	12,3	12,1	16,5	8,1	11,3	15,3

Since the wide-seated seeding machines with CSS eliminate the drawbacks for the above reasons, in principle, it is not possible to use a combined pneumo-mechanical sowing system on such seeders (Fig. 90), which allows using the positive qualities of the sowing systems of seeders with individual seed dosing and seeders with a CSS [60].

The proposed combined pneumo-mechanical sowing. The system contains: centralized tanks for seeds (4) and fats (5), each of which has common coil type metering devices (3) and (6); receiving chambers (2) consisting of a confuser and a diffuser (14); two high-pressure fans (1), driven by the PTO of the tractor and gearbox (13); two horizontal high-loaded seed distributors (15) and fats (19) supplying the material to the second distribution stage with material pipes (7) and (8). Horizontal distributors of seeds and fats have the number of outlet nozzles equal to the number of narrow-cut soil-cultivating-sowing modules-sections.



a) - side view; b) - top view
Fig. 90. Combined pneumo-mechanical system

As the second stage of distribution, conventional grain containers (compensators) for seeds (12) and fats (10) with individual coil metering devices (16) and (17) are used, the number of which is equal to the number of coulters of the module-section.
On the grain tanks (compensators) of the second distribution stage, the discharge cyclones (9) and (11) of the corresponding capacity are installed to divide the air flow and the materials to be sown. Individual coil dispensers of grain containers supply seeds and fats to the vanes, which, under the action of gravity, enter the openers (18).
To reduce the uneven distribution of the mixture between the outlet nozzles of the horizontal high-loaded seed distributor and the fat, changes were made to it, compared with the proposed [68], based on the following considerations [74].
The rates of sowing of cereals and leguminous crops and fats lie in the range of 50-300 kg/ha. Pneumatic transport of such quantity of material is carried out by an air stream with speed not less than 10-30 m/s. With this quantity of feed material and air velocity values, the concentration coefficient of the two-component mixture flow is in the range 0,75-2,5. In this case, in a two-component flow in a horizontal material conduit, solid particles move along its lower part of the cross-sectional area [69]. Therefore, most of the sown material will flow into the lower part of the horizontal divider (Fig. 91), which will not allow evenly distributing them between the outlet nozzles (3).
The proposed horizontal distributor [74] (see Fig. 91) consists of an inlet branch pipe 1 and outlet nozzles 3, a housing 2, a rhombic partition 6 installed at a height of 0,5 of the diameter of the inlet pipe and dividing the inner cavity into the upper and lower parts. The distribution plates 4 and 5 with spherical reflectors 9 are installed on the air-seed-fat mixture facing the flow 7 of the shell walls 2 and the partition 6. The inlet pneumatic duct 8 is connected to the inlet branch pipe 1.

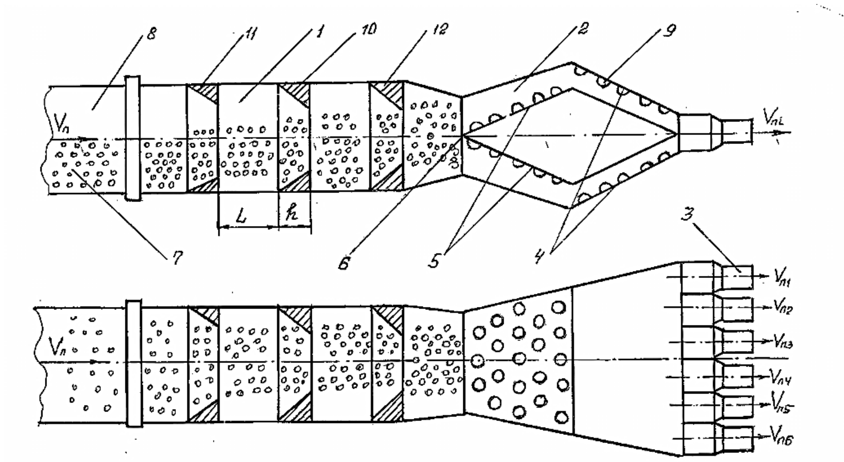


Fig. 91. Scheme of a horizontal distributor of loose materials with turbulators to pneumatic or combined pneumo-mechanical sowing systems of wide-spread seeding machines.

The inlet branch pipe 1 is provided with annular turbulators 10, made in the form of a truncated hollow cone and installed at certain intervals with each other. At the same time, the turbulators face the flow of a mixture of large diameters of the truncated cone rings, which, in order to reduce the trauma of the seeds, is equal to the internal diameter of the inlet branch pipe. Moreover, at the first input side of the flow direction of the turbulizer mixture stream 11, the smaller diameter of its truncated cone has a greater value than the smaller diameter of the output turbulator 12.

The air-seed-flow-stream 7, flowing along the pneumo-material-duct 8, whose solid particles move along the bottom of the pneumatic duct, enters the inlet pipe 1 with turbulators 10, which increase the turbulence of the flow. Since the smaller diameter of the truncated cone at the inlet ring turbulizer 11 is larger than that of the output turbulator 12, the solid particles of the multicomponent flow 7 are centered and more evenly distributed over the cross-sectional area of the outlet opening of the inlet pipe 1. Therefore, the rhombic baffle 6 of the proposed horizontal bulk material distributor which will unlike existed ones divide the flow of the mixture into practically equal parts between the lower and upper cavities of the distributor, creating in them approximately the same conditions for the movement of seeds and fats, which makes it possible to reduce the uneven distribution of the flow between outlet nozzles 3.

The proposed sowing system [60] will function normally only if the number of seeds and fat (Q_n, kg) fed by the central coil metering devices is equal to the consumed amount (Q_p, kg) of seeds and fats by individual coil units of technological modules [75], i.e. $Q_n = Q_p$.

The necessary amount of seeds and fodders supplied and consumed is regulated by the agrotechnical rate of their application per 1 hectare (kg/ha).

Practice has shown that the individual coil-type feeders used in seeding machines provide predetermined rates of seed sowing with an overall seeding ± 2%, which practically does not negatively affect the fulfillment of the condition $Q_n = Q_p$.

In the combined pneumo-mechanical sowing system, the uneven distribution of seed and fat between the output channels of the horizontal dividing head (the first stage of distribution) can reach 10-15%. In this case, the condition $Q_n = Q_p$, will not be fulfilled for some capacities of technological modules. In order to reduce the probability of failure to fulfill the condition $Q_n = Q_p$, the capacities of process modules are designed to play the role of compensators, i.e. before starting work, they should be filled with seeds and fat by about 2/3 of their total volume. However, if the unit is operated for a long time, the difference in the number of seeds and fats supplied and consumed can lead to an overflow of the compensating grain containers of some modules or to their complete emptying.

In this connection, it becomes necessary to determine the minimum time period for which an overflow (T_n) or emptying (T_0) of some of the compensation capacitances of process modules

$$T_n = \frac{E}{Q^i_{nmax} - Q^i_{nmin}} \ , \tag{141}$$

$$T_0 = \frac{E}{Q^j_{pmin} - Q^j_{pmax}} \ , \tag{142}$$

where: Q^i_{nmax}, Q^i_{nmin} - the maximum, minimum number of seeds (tukov) fed by one of the output channels of the horizontal dividing head into the capacity of the technological module;
Q^j_{pmax}, Q^j_{pmin} - the maximum, minimum quantity of seeds (tukov) consumed by individual sowing units of the coil type with the capacity of the technological module;
E - working number of seeds in the compensatory capacity of the technological module, i.e. E = 0,67 of the total number of seeds (tukov) in the module's capacity.
In general, the number of seeds fed and consumed, for example, is determined by a given seed rate per hectare (N, kg/ha). Therefore, taking into account the instability of the general rate of seeding by the centralized coil metering device, the maximum, minimum number of seeds entering into one module will be equal to

$$Q^i_{nmax} = (N + 0{,}15N): 6,$$
$$Q^i_{nmin} = (N - 0{,}15N): 6. \tag{143}$$

Individual coil apparatuses of the tanks of technological modules provide an instability of the general seed sowing rate within ± 2%, i.e. the maximum and minimum number of consumed seeds will beequal to

$$Q^j_{pmax} = N + 0{,}02N \ ,$$
(144)
$$Q^j_{pmin} = N - 0{,}02N \ .$$

Define T_n and T_0 for the case of the maximum seeding rate of wheat seeds (250 kg/ha) and the speed of the aggregate v = 10 km/h with a capture width (B) equal to 12 m.
The capacity of the unit in this case per hour of net working time will be equal to

$$W = 0{,}1 \cdot B \cdot v = 0{,}1 \cdot 12 \cdot 10 = 12 \text{ ha/h}.$$

Then, in the ideal case, the quantity of seeds Q must be supplied and expended in 1 hour of operation of the unit

$$Q = W \cdot N = 12 \cdot 250 = 3000 \text{ kg/h}.$$

At the same time, 3000:6 = 500 kg/h of wheat seeds should flow into each of the six branch channels of the horizontal divider. However, taking into account the general instability of the norm of sowing by the centralized coil metering device
(± 2%) and the uneven distribution of seeds between the horizontal divider channels [± (10-15%)], the amount of seeds supplied to each capacity of the technological module, according to (143) equals to

$$Q^i_{n.max} = 500 + 0{,}15 \cdot 500 = 575 \text{ kg/h},$$

$$Q^i_{n.min} = 500 - 0{,}15 \cdot 500 = 425 \text{ kg/h},$$

i.e. the difference from the established rate of its sowing will be 75 kg/h.
The amount of seeds Q^j_p, consumed per hour by a single technological module at $B_м$ = 2 m and V_n = 10 km/h and a rate of application of 250 kg/ha will be:

$$W_{м} = 0{,}1 - B_{м} \cdot v_n = 0{,}1 \cdot 2 \cdot 10 = 2\text{ha/h},$$
$$Q_p^j = W_{м} \cdot N = 2 \cdot 250 = 500\text{ kg/h}.$$

Then

$$Q_{n.max}^j = 500 + 0{,}02 \cdot 500 = 510\text{ kg/h},$$
$$Q_{n.min}^j = 500 - 0{,}02 \cdot 500 = 490\text{ kg/h}.$$

Thus, in the case of

$$Q_{n.max}^i - Q_{p.min}^j = 575 - 490 = 85\text{kg/h},$$
$$Q_{n.min}^i - Q_{p.max}^j = 425 - 510 = -85\text{ kg/h}.$$

With a grain capacity capacity of the technological module 276 dm^3 and density of wheat 800,0 kg/m^3, the number of seeds in the box will be

$$800{,}0 \cdot 0{,}276 = 220{,}8\text{ kg}.$$

In the event that one-third of this quantity of wheat seeds (73 kg) is used as compensation, the "working" number of seeds will be E = 148 kg. This number of seeds according to (142) is sufficient for the operating time of the unit at

$$Q_{nmax}^i - Q_{pmin}^j,\ T_n = 148:575 = 0{,}26\text{ h},$$

$$\text{thus, when}\quad Q_{nmin}^i - Q_{pmax}^j,\ T_0 = 148:425 = 0{,}35\text{ h}.$$

In the first case, the seeds accumulate 22,10 kg (85·0,26 = 22,1 kg), which may eventually lead to overflow of the module capacity, and in the second case, their quantity will decrease by 29,7 kg (-85·0, 35 = - 29,7 kg), which predetermines the possibility of complete emptying of the module capacity. Since the "working" amount of seeds in the module's capacity is 148 kg, the grain capacity will overflow after 6,7 hours of operation (148:22,1 = 6,7), and emptying after 5,0 hours (148:29.7 = 5,0).

Overflow of capacity after 6,7 hours and emptying after 5,0 hours of net work time of the unit will occur only if:

- the general instability of the norm of sowing by centralized coil metering devices will constantly increase (+2%) or decrease (-2%);
- the unevenness of the distribution between the output channels of the horizontal divider will be 15%, with each of the output channels receiving more seeds (+ 15%) or less (-15%);
- the general instability in the rate of sowing of individual coil units of process modules will consistently consume more seeds (+ 2%) or less (-2%).

The complete coincidence of these events during the operation of the unit is virtually eliminated, since the general instability of the norm of seeding by the central coil metering device, the uneven distribution of the horizontal dividing head along the output channels, the general instability in the rate of seeding of the technological modules by individual coil apparatuses are random processes in which the numerical values of it constantly vary greater or lesser than the average.

Consequently, it is been assumed that overflow or complete emptying of grain containers of process modules does not occur during the entire time of the shift (10 hours). In case of a change, it is necessary to check the level of seed filling and fertilization of the module's capacities and approximately to align them manually. Installed in the capacity of the modules, the level sensors of their filling allow controlling the time of overflow or emptying in case of clogging (malfunctioning) of the pneumatic distribution system.

The foregoing results of studies on the choice of the type of a basic constructive scheme of combined tools for shallow soil treatment [24] as well as the type and parameters of the combined pneumatic sowing system [43,49,60,74,65] (see Fig. 90) form the basis creation of a seeder-cultivator of block-modular SKBM-12 [76] is shown in Fig. 92.

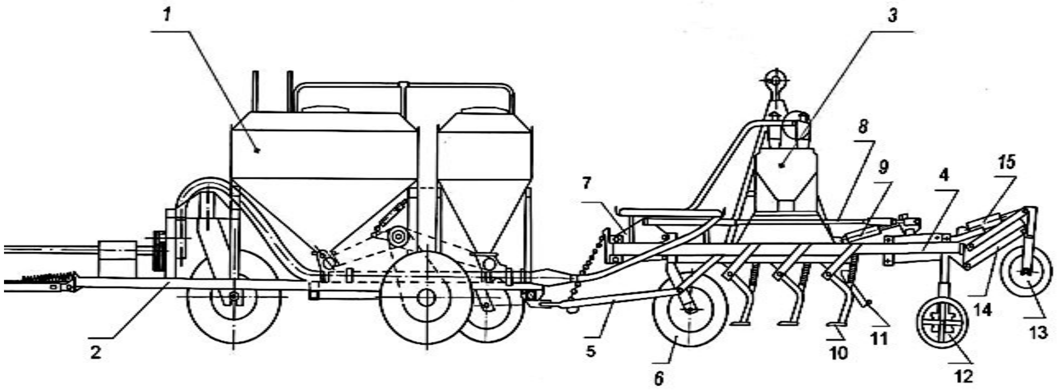


Fig. 92. The scheme of the seeder-cultivator of the block-modular

SKBM-12 consists of the centralized system of the centralized system (1), the coupling (2) and the technological module (3). The basis of the module is a flat frame (4), on which are mounted:
- a hitch frame (5) with a trailer height adjustable;
- the front support and adjustment wheel (6), moved in height relative to the frame by means of a parallelogram mechanism (7), adjustable along the length of the rod (8), the hydraulic cylinder (9);
- loosening (10) and leveling (11) working elements;
- the support roller (12), which is pivotally connected to the frame of the module, whose height position is adjusted by means of a pull rod (8) and a hydraulic cylinder (9) in synchronism with the front wheel-adjustment wheel (6).

A frame with transport self-aligning wheels (13) is pivotally connected to the suspension axis of the roller frame (12). These wheels with the help of the mechanism (14) and the hydraulic cylinder (15) allow to transfer the wide-grip unit to the position of long-distance transport.

The use of two transport wheels instead of one provides a stable position of the module, since its center of gravity will be within three support points. This allows to exclude from the construction of the unit locking devices that connect the rigid frames of the extreme and adjacent modules.

The results of the acceptance tests of SKBM-12, conducted by the accredited testing center of the KB LLP "KAZNIIMESH", (protocol No. 228-428-2011 of October 2011), showed the following: Grain and fat dispensers of the tanks of the Central seed system (CSS) block ensure the seeding rates of wheat seeds (grade "Omskaya 29") and granulated mineral fertilizer (superphosphate). The maximum sowing ability of wheat was 256,5, the minimum 48,6 kg/ha, superphosphate, respectively, 206,3 and 4,48 kg/ha, which is within the permissible agricultural demand limits of deviations from the set seeding rates (± 5%).

At the economic rate of sowing superphosphate (80 kg/ha), 81,4 kg/ha (deviation from the target ± 1,8%) was sown. The unevenness of the sowing of fertilizers between the apparatus of the compensator tanks was ± 5,4% with an allowable ± 10%. In this case, the instability of the total seeding reached ± 4,8% with an allowable ± 10%.

Thus, the proposed combined pneumo-mechanical system for wide-seated seeding machines (see Figure 90) made it possible to eliminate the main disadvantage of pneumatic sowing systems - to reduce the uneven distribution of seeds between openers from ± (10-30)% to ± 2,5% ± 6%.

At the same time, SKBM-12 satisfies all the specific agrotechnological (initial) requirements imposed by soil conservation agriculture on seeding machines.

With the required depth of coulters 4,0-8,0 sm SKBM-12 provides a minimum of 4,21 and a maximum of 8,34 sm depth of their embedding. At an established depth of colter stroke of 7,0 sm, the average depth of seeding was 7,5 sm with a root-mean-square deviation of a depth of ±0,7 sm. The number of seeds embedded in the horizon corresponding to the average depth and two adjacent one-centimeter layers was 81,8%. Those have not been seeded into the soil after passage of SKBM-12 are absent.

After passing the packing rollers of technological modules, the average height of the unevenness of the surface of the field was 4,8 sm with an allowable 5,0 sm. Ring-shoe rollers provide soil strip compaction to 1,18 g/sm^3 in the seed arrangement, which corresponds to a given soil type 1,1-1,2 g/sm^3.

After passage of SKBM-12, the content of erosion-hazardous particles in the upper soil layer 0-5 sm is reduced by 6.9%, which corresponds to the requirements - the number of erosion-hazardous particles should not increase.

SKBM-12 provided 99,1% of weed cropping at the required - not less than 97%.

In the process of testing, premature emptying or overflow of capacitor-compensators of technological modules was not observed.

The block-modular principle used in the SKBM-12 allows, unlike foreign seeding complexes, to form the factory to the manufacturer at the request of consumers aggregates of different width of capture to tractors of class 2,3,4 and 5.

According to the applications of the consumers, SKBM can be completed by the manufacturer with opener paws and rolling compactors for row or belt sowing.

SKBM-12 are equipped with:

- sensors of the level of seeds and fat in the tanks of the CSS and in each container-compensator of technological modules;
- sensors controlling the operation of centralized coil dispenser'sCSS and supplying seeds and fat in the tank-compensators of technological modules;
- sensors controlling the operation of the sowing apparatus of capacitances-compensators of technological modules.

The indicators of all sensors are output to the alarm panel installed in the tractor driver's cab. Cost SKBM in comparison with similar in the width of seizure sowing complexes far abroad is in 1,4-1,6 times less. The use of SKBM-12 in comparison with the sowing unit SZTS-12 allows to reduce operating costs by 14,1% and to receive an annual economic effect of 416,9 thousand Tenge.

3.4 Selection and justification of parameters of constructive-technological scheme of a seeder for cultivationof agricultural crops by comb technology

Under certain soil and climatic conditions, in order to create optimal conditions for the growth and development of crops, to preserve soil and its fertility, and to reduce material and financial costs, the comb technology of their cultivation is applied.The main methods of resource saving with the comb method of sowing are carrying out sowing with simultaneous cutting of irrigation furrows, fertilization and formation of ridges, increasing the uniformity of irrigation, reducing the cost of irrigation water; carrying out of call-up water-charging and early vegetative watering; reduction of unproductive discharges and erosion of soils; ensuring the uniformity of irrigation techniques for all crops of field crop rotation and reduction of the seeding rate up to two times [77].

A combination of comb-like technology and zero tillage, that is, cultivation of crops along "permanent ridges and furrows", is particularly effective. The main principle of the new technology is the use of permanent ridges for cultivating crops under irrigation conditions [77]. The essence of the technology is that the crests formed during the cultivation of winter wheat are used for 5 or more years for direct sowing of subsequent crops, thereby ensuring the preservation of soil fertility, rational use of land and water resources, saving energy and resource costs, protecting the environment environment.

Especially important is the role of permanent ridges in preventing water erosion and lodging of plants. The new technology is fundamentally different from the existing technologies for cultivating crops under irrigation that is based on the following principles [77]:

- cropping on permanent crests for 5 years or more;
- exclusion of basic and pre-sowing soil treatments (plowing, harrowing, leveling and compaction of soil layer) by direct sowing into the ridges along the stubble of the previous crop;

- watering of plants on permanent furrows, providing a sharp reduction in irrigation water consumption, preventing irrigation soil erosion;
- rational use of fertilizers by local introduction of them into ridges of furrows;
- preservation and increase of fertility of irrigated arable land, environmental protection.

For formation of soil ridges and sowing of seeds of agricultural crops, special combination machines are used, for example, a seeder-cultivator for comb seeding (Murzaev F.F., et al. // Agriculture, 1989, No. 5, p. 67-68); combined seeder from WAZONER (India); a seeder combined for sowing crops and forming a ridge field surface (Patent 2110903, Russia IPC.5 A01B 49/04, Far Eastern Agricultural Research Institute., 27.02.96, published on May 20, 1998).

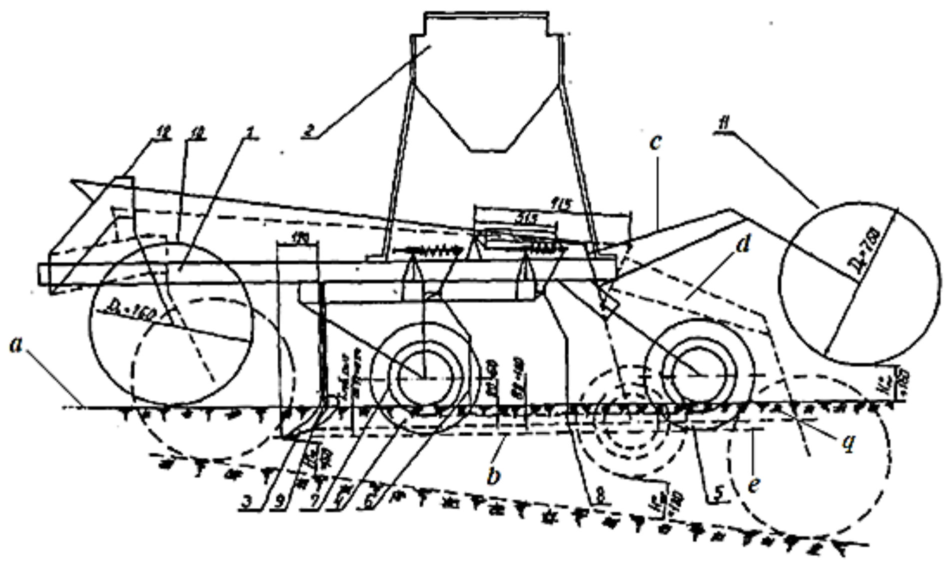
The main disadvantages of these machines are:
- it is generally known that the average height of the unevenness of the fields is ± (2 - 4) sm, so that the paw hills can not in principle operate at a depth of 1-2 sm and therefore it is not possible to provide an even seed depth by forming over the seeds an even layer of soil;
- in the case of coulters sowing in uncompacted soil ridges, the seeds are also unevenly embedded in depth, the specified profile of the cross section of the ridge is violated, and its resistance to degradation decreases with time;
- in both cases, the light weight of the squeezing crest-comber does not allow, despite the presence of a spring mechanism, to create the necessary pressure on the soil and form crest-resistant crests.

The last drawback was eliminated in the crest - the zero seeder СГНММ 2,8-НИИВХ [77]. In it, instead of the spring mechanism, the comb-forming rollers use the part of the seed mass that comes with them.

To eliminate these drawbacks, a sowing comber was proposed (preliminary patent No. 17400 of the Republic of Kazakhstan, SPA "Kazselkhozmechanization", filed on 22.11.2004, published on June 15.2006) containing a frame, seed and fat capacity, paw hills, coulters for co-location of seeds and two rows of ridge-forming rollers, the first of which is set behind the hillers, and the second behind the openers, a mechanism for adjusting the depth of the path of the hoe-pushers and transferring the seeder to the transport position. The coulters place the seeds together with the fat in the ridges, preformed by the hillsides and the front row of the rink. In this case, the coulters quite significantly loosen and destroy the profile of the ridge. The open crests of the coulters are preformed and compacted by the second row of the ridge roller as they have the same shape and cross-sectional profile with the rollers of the front row and are located on one longitudinal-horizontal axis. At the same time, the gravity from 2/3 of the total mass of the seeder and the seeds and fats located in its tanks is divided into two rows of ridge rollers, and 1/3 to the front support wheel, which allows, without additional mechanisms, to form, compact and create stable to the destruction of soil ridges.

However, this seeder-comber also has several drawbacks:
- seeds and fats are co-located in the soil crest;
- regardless of the type and suspension of the opener, they are not able to provide a precisely defined depth of seeding, since they operate under conditions of a loose and insufficiently compacted crest formed only by the front roller-comb;
- in the transport position, the planter-comber is supported by the front support wheel and the rear roller-comb forming machine, which does not exclude the possibility of its deformation and breakage, especially when turning the unit at the end of the pen and during long transportation.

To eliminate the shortcomings of seeder-combers mentioned above, a comb-forming machine with separate placement in the crests of seeds and fat was proposed (Innovative patent No. 24793 Republic / Kazakhstan by KazNIIMESH LLP, declared 08.02.2011, published on November 15.11.2011). The comb forming machine (Figure 93) consists of a frame 1, a seed and a fertilizer tank 2, lap-hillers 3, a front 4 with deformers 7 and a rear row 5 of ridge rollers, coulters with vanes and otkartachami 8, tubular racks with handrails 9, front the self-aligning support wheel 10, the rear support wheels 11 and the adjustment mechanism 12 of the loosening depth of the pawl carriers and the transfer of the machine to the transport position.



a- the upper surface of the ridge. *b* - the bottom of the furrows *c* - working position. *d*- transportnoe position.*e* - the cultivation depth of the mouldboard. *q* -the height of the ridge.
Fig. 93. - Ridge-forming machine with separate placement in the crests of seeds and fats

During operation, tubular racks with drums (Figure 94), installed in the same row and between the paw pens 3, feed the fat to the surface of the untreated field. The soil that descends from two adjacent paw-hills covers the fat and forms a trapezoidal crest above them with arbitrary parameters. Following the first roller 4 forms the given profile of the ridge and the deformers 7 form two open grooves, the distance between them and their depth exactly correspond to the row and the depth of sowing, stipulated by the agrotechnical requirements of the sown crop. The colter struts 6, located above the open grooves, guide the flow of seeds into them. The openers 6 of the openers 8 fixed on the stand seal the furrow with seeds, and the second roller 5 finally forms and compacts the crest-resistant crests, the profile of which is shown in Fig. 95.
When transferring the machine to the transport position, the rear wheels 11 fall to the surface of the field and the machine will have three support points. This makes it possible not only to withdraw the ridge rollers from contact with the soil, but also to automatically switch off the sowing seeds and fat devices as they have been driven from the rear row of the rink 5.
In order to unify the units and parts, reduce the labor costs and the means to manufacture such a combined comb-forming machine, it is advisable to develop on the basis of the technological modules of the existing seed drill unit SZC-6/12 or the block-modular SKBM-12 cultivator-cultivator created in recent years. This predetermines the need to maintain the kinematic scheme and the diameter of the rollers (550 mm) in the developed comb forming machine.

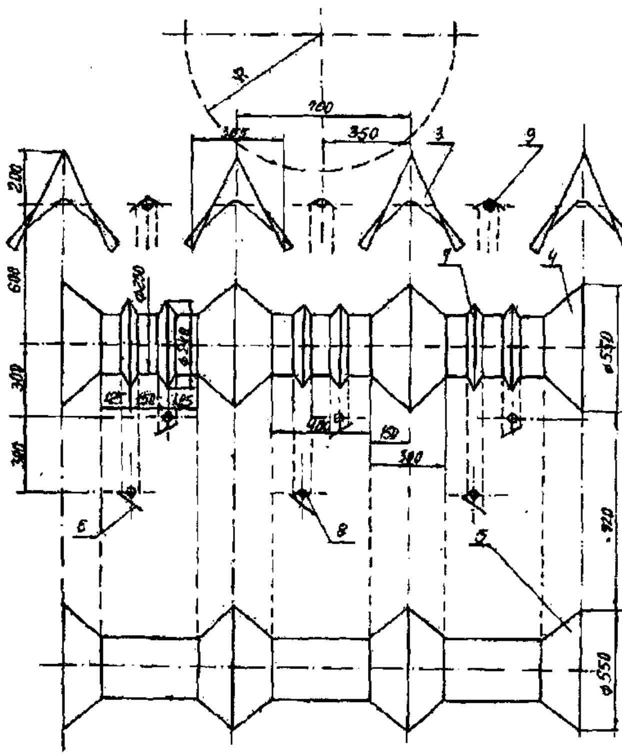


Fig. 94. Scheme of the arrangement of the working elements of the comb-forming machine in plan

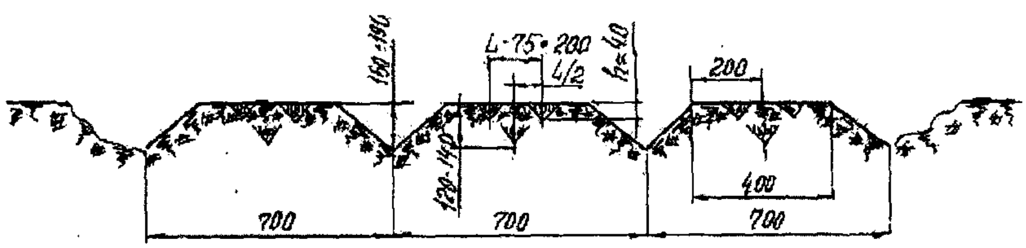


Fig. 95. The cross-section of the profile of the formed crests

With the adopted kinematic scheme and the diameter of the ridge-forming roll 550 mm, the diameter of the smooth roller forming the horizontal surface of the ridge should be 230 mm to form the maximum height of the crest 160 mm provided by the agricultural requirements.
Mounted at a depth of loosening of 80-160 mm, the hind legs ahead of the front rink with an interval of 700 mm spill the soil onto the surface of the field, forming a certain height of the trapezoidal mound in the transversely vertical plane. Tapered rollers with a diameter of 550 mm roll over the bottom of the groove formed by the hillsides and form the lateral inclined sides of the trapezium of the ridge, and smooth cylindrical rollers with a diameter of 230 mm should compact the mound to a certain height, i.e. 160 mm. It should be noted that the height of the ridge may be more than 160 mm due to the relaxation of the deformed soil.
To determine the condition (criterion) of compacting the soil with a smooth cylindrical roller without soil unloading, let us consider the forces arising during their interaction (Fig. 96) [79,80].

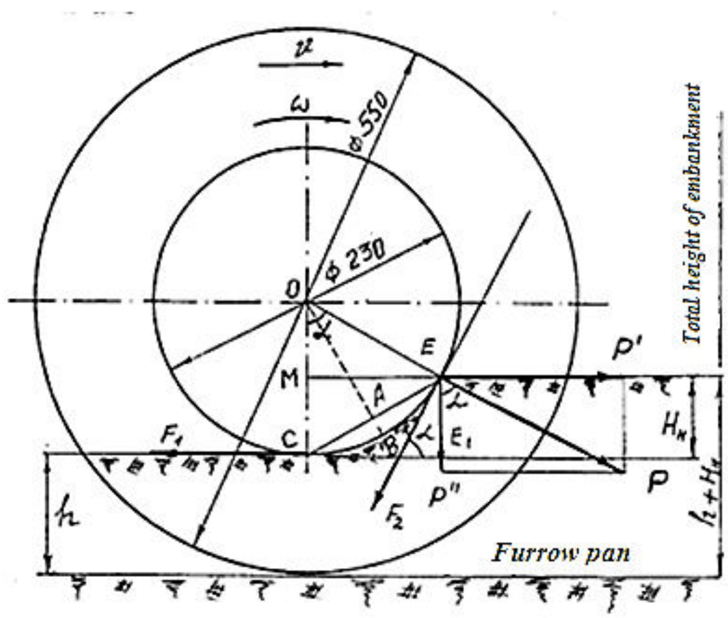


Fig. 96. Forces arising from the interaction of a smooth cylindrical roller with soil

From the action of the force P there is a frictional force F_2 between the rim of the rink and the obstacle from the soil, and also the force F_1 between the soil obstacle and the soil surface directed in the direction opposite to the direction of the roller's motion.
The pinching of the soil obstacle between the soil surface and the surface of the roller takes place if

$$F_1 + F_2 \cdot cos\alpha \geq P', \quad (145)$$

where: P' - is the horizontal component of the force P equal to the obstacle; α - is the angle between the horizon and the tangent to the circumference of the rink held at point E of its contact with the obstacle.
Wherein,

$$P' = P \cdot sin\alpha;\ F_2 = P \cdot \text{tg}\alpha_{2;}\, F_1 = N \cdot \text{tg}\alpha_1, \quad (146)$$

where: α_1, - is the angle of internal friction of the soil; $\alpha_{2;}$ is the angle of friction of the soil over the metal;
N - is the normal pressure of the roller on the soil, which is equal to

$$N = P'' + F_2 \cdot sin\alpha - P \cdot cos\alpha + P \cdot \text{tg}\alpha_2 \cdot sin\alpha \quad (147)$$

When substitute the values (146) and (147) into (145), we obtain

$$P \cdot cos\alpha \cdot \text{tg}\alpha_1 + P \cdot tg\alpha_2 \cdot sin\alpha \cdot \text{tg}\alpha_1 + P \cdot tg\alpha_2 \cdot cos\alpha \geq Psin\alpha. \quad (148)$$

Reducing (148) by P and dividing by *cos a*, we obtain

$$\text{tg}\alpha \leq \frac{\text{tg}\alpha_1 + \text{tg}\alpha_2}{1 - \text{tg}\alpha_1 \cdot \text{tg}\alpha_2}, \quad (149)$$

$$\text{or } tg\alpha \leq \text{tg}(\alpha_1 + \alpha_2).$$

Thus, the soil will not be unloaded by a smooth cylindrical roller provided

$$\alpha_{krit} \leq \alpha_1 + \alpha_2. \tag{150}$$

An integral part of the comb-forming machine is its conical parts, in which the working surfaces are inclined to the horizon at an angle approximately equal to the angle of internal friction of the soil (see Fig. 94).
The nature of soil compaction by wedge-shaped rollers will be somewhat different than under the action of a smooth roller [79,80]. Wedge roller with its ribs triangular profile deforms the soil not only in the vertical and longitudinal, but also in the transverse direction. However, in the machine being developed, the tapered parts of the rollers roll over the compacted bottom of the furrow formed by the paw blade. In connection with this, and the fact that the angle of the wedge of the rollers is approximately equal to the sum of the two values of the angles of internal friction, then with a certain assumption it can be ignored in further analysis of the process of interaction of the comb-forming roller with the soil.
To find the values of the angle a (see Fig. 96), it is necessary to determine the thickness of the soil thrown by the paw-hills on the surface of the field $H_н$, depending on the depth of loosening of the soil h (Fig. 97). At the same time, with a certain assumption, we will not consider the volume of the soil thrown by the stubble during the movement of the machine, but only the cross-sectional area of the embankment in the transverse-vertical plane of $CENC_1$, and in the longitudinal plane the length of the mound shall be equal to unity (mm, sm, m, etc.).
The cross-sectional areas of the soil $ABCD$and $A_1B_1C_1D_1S$ and S_1 cut by the paw-hills are practically equal and are

$$S = AD \cdot h,(151)$$
$$S_1 = A_1D_1h_1,$$

where: h - is the established depth of loosening of the paw-hills;
$AD = A_1D_1 - 0{,}5$ of the width of capture of the paws-hillers and is equal to $AD = A_1D_1$ =11,25 sm.

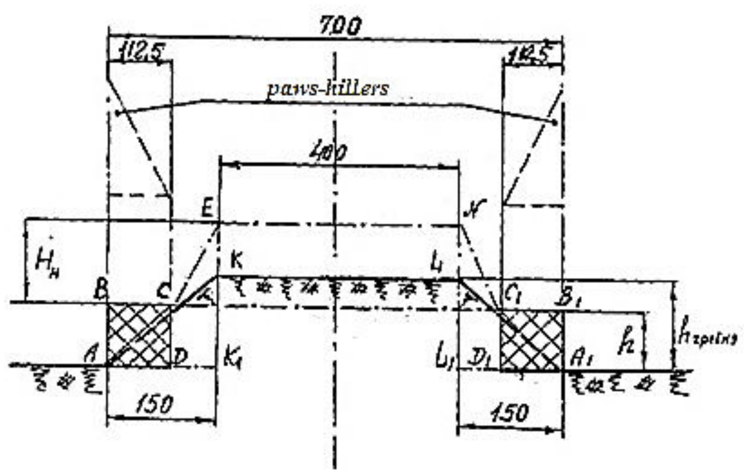


Fig. 97 - To the formation of the cross section of the profile of the ridge

Then the total cross-sectional area of the soil lifted by two paws-hillers at a depth of loosening of 8, 12 and 16 sm will be equal to

$$S_8 = (11{,}25 \times 8) \cdot 2 = 180 \text{ sm}^2,$$
$$S_{12} = (11{,}25 \times 12) \cdot 2 = 270 \text{ sm}^2, \tag{152}$$
$$S_{16} = (11{,}25 \times 16) \cdot 2 = 360 \text{ sm}^2.$$

Assuming that the cross-sectional area of the embankment, taking into account the angle of internal friction of the soil, takes the form of the trapezium $CENC_1$ (S_{imp}) and will be equal to the cross-sectional area of the soil lifted by the hillers, i.e. $S_{imp} \approx S_i$.The cross-sectional area of the trapezoid is

$$S_{imp} = \frac{EN + CC_1}{2} \cdot H_{\text{н}}, \tag{153}$$

where:EN= 40 sm, CC_1= 70 sm-2AD=70 sm-2·11,25=47,5 sm.

Then

$$S_{imp} = \frac{40{,}0 + 47{,}5}{2} \cdot H_{\text{н}} = 43{,}75 \cdot H_{\text{н}}, \text{ a}$$

$$(H_{\text{н}})_i = \frac{S_{imp} - S_i}{43{,}75} = \frac{S_i}{43{,}75}, \tag{154}$$

Substituting in (154) the values of S_1 from (152), we obtain that at a depth of loosening of the paws-hillers h = 8 sm, the height of the embankment $H_{\text{н}1}$ will be 4,11 sm, for h = 12 sm $H_{\text{н}2}$ = 6,17 sm and for h = 16 sm $H_{\text{н}3}$ = 8,23 sm.
Thus, the loose compacted soil should be compacted to 4,11; 6,17 and 8,23 sm at a depth of loosening of the hind legs, respectively, 8,12 and 16 sm.
Knowing the height of the embankment $H_{\text{н}}$, you can determine the length of the circumference of the roller, in contact with the soil, the chord and the angle of the circular segment of the EOC (see Fig. 96).
From the rectangular triangle EOM we find

$$(\text{EM})_i = \sqrt{\text{OE}^2 + (\text{OM})_i^2}, \tag{155}$$

where: OE - is the radius of a smooth cylindrical roller, equal to 11,5 sm;$(\text{OM})_i = OC - H_{\text{н}i}$, but $OC = OE$ = 11,5 sm.
Since the depth of loosening of the hawse-foot is 8,12 and 16 sm, the height of the soilmound is equal, $H_{\text{н}i}$ respectively 4,11; 6,17 and 8,23 sm, then $(\text{OM})_i$ = 7,39 sm; $(OM)_2$ = 5,33 sm and $(\text{OM})_3$ = 3,27 sm.
Then

$$\begin{aligned} (\text{EM})_1 &= \sqrt{11,5^2 - 7{,}39^2} = 8{,}7 \text{ sm};\\ (\text{EM})_2 &= \sqrt{11,5^2 - 5{,}33^2} = 10{,}19 \text{ sm};\\ \text{EM})_3 &= \sqrt{11,5^2 - 3{,}27^2} = 11{,}00 \text{ sm}. \end{aligned} \tag{156}$$

From the right triangle EE_1C we find the chord of the circular segment $(CE)_{i}$

$$(\text{CE})_i = \sqrt{(\text{EE}_1)^2 + (\text{E}_1\text{C})_i^2}, \tag{157}$$

where:$(\text{E}_1\text{C})_i = (\text{EM})_i$, but $(\text{EE}_1)_i = H_{\text{н}i}$.
Therefore, when:

$$\begin{aligned} &H_{\text{н}1} = 4{,}11 \text{ sm} \qquad (\text{CE})_1 = \sqrt{4,11^2 + 8,7^2} = 9{,}62 \text{ sm};\\ &\text{H}_{\text{н}2} = 6{,}17 \text{ sm} \qquad (\text{CE})_2 = \sqrt{6,17^2 + 10{,}19^2} = 11{,}9 \text{ sm}; \end{aligned} \tag{158}$$

$$\mathrm{H}_{\text{н}3} = 8{,}23 \text{ sm} \qquad (\mathrm{CE})_3 = \sqrt{8,23^2 + 11,0^2} = 13{,}74 \text{ sm}$$

According to the sine theorem, we have from the triangle *COE*

$$(\cos a)_i = \frac{\mathrm{EO}^2 + \mathrm{OC}^2 - (\mathrm{CE})_i^2}{2 \cdot \mathrm{OE} \cdot \mathrm{OC}}. \tag{159}$$

Then

$$\cos\alpha_1 = \frac{11,5^2 + 11,5^2 - 9,62^2}{2 \cdot 11{,}5 \cdot 11{,}5} = 0{,}6;$$

$$\cos\alpha_2 = \frac{11,5^2 + 11,5^2 - 11,9^2}{2 \cdot 11{,}5 \cdot 11{,}5} = 0{,}4646; \tag{160}$$

$$\cos\alpha_3 = \frac{11,5^2 + 11,5^2 - 13,74^2}{2 \cdot 11{,}5 \cdot 11{,}5} = 0{,}2862.$$

Therefore $a_1 \approx 49°$; $a_2 \approx 62^0$; $a_3 \approx 73^0$.
Depending on the type of soil and its minimum and maximum moisture content, the coefficient of internal friction is 0,7 and 1,15, respectively, and the soil friction coefficient for steel is 0,40 and 0,75 [81].
Assuming that the soil should be treated at the optimum humidity, the value of which is approximately between the minimum and maximum humidity, then the average value of the coefficient of internal friction can be taken equal to 0,975, and the external friction – 0,575, i.e. $a_2 \approx 43^0$; $a_1 \approx 30°$.
Then, according to (150), the critical value of the angleα_{krit} is equal to

$$\alpha_{krit} \leq 73^0. \tag{161}$$

Comparing the angles a_i, obtained for different depths of loosening of the soil by pawl-paws, with the critical angle α_{krit}, we see that at a depth of loosening of the soil by hills 8 and 12 sm, a smooth cylindrical roller Ø230 sm will not unload it ahead of itself. At a depth of loosening 16 sm, this roller can unload the soil. Therefore, for the proposed comb-forming machine, the maximum permissible depth of loosening of the paw-hills is 16 sm.
In order to find the pressure of a smooth cylindrical roller on the soil, it is necessary to know the total interaction area $S_{вз}$ (contact) of the roller with soil, which is determined from expression

$$(S_{\text{вз.}})_i = (L_{CBE})_i \cdot B, \tag{162}$$

where: $(L_{CBE})_i$ –is the arc length of the circular segment of the EOC (see Fig. 96);
B– is the length of the cylindrical part of the rollers of the comb forming machine.
The value$(L_{CBE})_i$ –is determined using the known formula for calculating the area of the circular segment of the EOC

$$(\mathrm{S_c})_i = \frac{1}{2}OC^2(a_i - sina_i) = \frac{1}{2}OC^2\left(\frac{L_i^0\pi}{180}\right) - sinL_i\,, \tag{163}$$

where: $a_i и L_i^0$ –the angles of the circular segment (160), respectively, in degrees and radians;
OC - is the radius of a cylindrical roller, which is 11,5 sm.
Thus

$$(\mathrm{S_c})_1 = \tfrac{1}{2} \cdot 11{,}5^2\ (0{,}8552 - 0{,}7547) = 6{,}65\ \mathrm{sm}^2;$$
$$(\mathrm{S_c})_2 = \tfrac{1}{2} \cdot 11{,}5^2\ (1{,}0647 - 0{,}8829) = 12{,}02\ \mathrm{sm}^2; \tag{164}$$

$$(S_c)_3 = \frac{1}{2} \cdot 11{,}5^2\,(1{,}2741 - 0{,}9563) = 21{,}01 \text{ sm}^2.$$

Knowing the area of the circular segment (164), we find the length of its arc, using the well-known formula

$$(S_c)_i = \frac{OC[(L_{CBE})_i - (CE)_i] + (AB)_i \cdot (CE)_i}{2}\ , \qquad (165)$$

where: $(AB)_i$ – height of the circular segment, which is equal to $OB - (OA)_i$
OB– radius of the roller, 11,5 sm;
$CE)_i$– the chord of the circular segment, the values of which are given in (158).
We find the value $(OA)_i$ from the right triangle OAC (see Figure 96):

$$(OA)_i = \sqrt{OC^2 - \left(\frac{(CE)}{2}\right)^2}. \qquad (166)$$

Thus

$$(OA)_1 = \sqrt{11{,}5^2 - \left(\frac{(9{,}62)}{2}\right)^2} = 10{,}46 \text{ sm},$$
$$(OA)_2 = \sqrt{11{,}5^2 - \left(\frac{(11{,}9)}{2}\right)^2} = 9{,}85 \text{ sm}, \qquad (167)$$
$$(OA)_3 = \sqrt{11{,}5^2 - \left(\frac{(13{,}74)}{2}\right)^2} = 9{,}26 \text{ sm}.$$

Thus

$$(AB)_1 = 11{,}5\text{-}10{,}46 = 1{,}04 \text{ sm},$$
$$(AB)_2 = 11{,}5\text{-}9{,}85 = 1{,}65 \text{ sm}, \qquad (168)$$
$$(AB)_3 = 11{,}5\text{-}9{,}24 = 2{,}26 \text{ sm}.$$

Taking into account the values $(CE)_i$ (158), $(S_c)_i$(164), $(OA)_i$(167) and $(AB)_i$(168), we find from (165) the arc length of the circular segment

$$6{,}65 = \frac{11{,}5[(L_{CBE})_1 - 9{,}62] + 1{,}04 \cdot 9{,}62}{2}\ ,$$
$$(L_{CBE})_1 = 9{,}901\text{sm}$$

$$12{,}02 = \frac{11{,}5[(L_{CBE})_2 - 11{,}9] + 1{,}65 \cdot 11{,}9}{2}\ ,$$
$$(L_{CBE})_2 = 12{,}28 \text{ sm}; \qquad (169)$$

$$21{,}01 = \frac{11{,}5[(L_{CBE})_3 - 13{,}74] + 2{,}26 \cdot 13{,}74}{2},$$
$$(L_{CBE})_3 = 14{,}69 \text{ sm}.$$

According to the scheme (Fig. 95), the total length of smooth cylindrical rollers of the first and second rows will be equal to

$$B = (3 \cdot 40) \cdot 2 = 240 \text{ sm}.$$

Then, taking into account the values of the arc length of the circular segment $(L_{CBE})_i$(169), the total interaction area of a smooth cylindrical roller with soil is given by formula (162)

$$(S_{\text{вз.}})_1 = 9{,}901 \cdot 240 = 2376{,}4 \text{ sm}^2,$$
$$(S_{\text{вз.}})_2 = 12{,}28 \cdot 240 = 2947{,}2 \text{ sm}^2, \quad (170)$$
$$(S_{\text{вз.}})_3 = 14{,}69 \cdot 240 = 3525{,}6 \text{ sm}^2.$$

The pressure (surface load) of a smooth cylindrical roller on the soil is found by expression

$$\sigma = \frac{P_k}{(S_{\text{вз.}})_i} \quad (171)$$

where: P_k- the weight of the comb-forming machine with seeds and fats filled with containers, falling on two rows of rollers. According to the adopted scheme of the comb-forming machine (see Fig. 93), both rows of rollers will account for 2/3 of the total weight of the machine with seasoned seeds and fat, i.e. $P_k = 0{,}67\text{P}$. Since $P = 1962$ kGs, then $P_к$= 1308 kGs.
Assuming that the weight P_k will be uniformly distributed between two rows of rollers, we obtain

$$\sigma_1 = \frac{1308}{2376{,}4} = 0{,}55 \text{ kGs/sm}^2,$$
$$\sigma_2 = \frac{1308}{2947{,}2} = 0{,}444 \text{ kGs/sm}^2, \quad (172)$$
$$\sigma_3 = \frac{1308}{3525{,}6} = 0{,}371 \text{ kGs/sm}^2.$$

In the process of the comb-forming machine, the ridges will form the first row of rollers, and the second row of rollers will correct the tops of the crests, which have been destroyed by the sowing holes and their seals. Therefore, the surface load of 1308 kGs will be perceived practically by the first row of the roller, the length of the cylindrical part of which will be 120 sm. Therefore, the interaction areas of a smooth cylindrical roller with soil $(S_{\text{вз.}})_i$will be equal

$$(S'_{\text{вз}})_1 = 9{,}901 \cdot 120 = 1188{,}12\text{sm}^2,$$
$$(S'_{\text{вз}})_2 = 12{,}28 \cdot 120 = 1473{,}6\text{sm}^2, \quad (173)$$
$$(S'_{\text{вз}})_3 = 14{,}69 \cdot 120 = 1762{,}8\text{sm}^2.$$

Then

$$(\sigma')_1 = \frac{1308}{1188{,}2} = 1{,}1\text{kGs/sm}^2,$$
$$(\sigma')_2 = \frac{1308}{1473{,}6} = 0{,}8876\text{kGs/sm}^2, \quad (174)$$
$$(\sigma')_3 = \frac{1308}{1762{,}8} = 0{,}742\text{kGs/sm}^2.$$

It is known [79,80] that when the soil is compacted, the maximum contact pressure (kGs / sm^2) should not exceed the limit of its strength, the value of which, when the smooth cylindrical roller is interacting with the soil, lies within 3-6 kGs/sm^2. The values of (σ'), which are given in (174), satisfy this condition.
It is also known that to obtain a soil density of 0,9-1,1 kGs/sm^3, set by agro-demand, it is necessary to apply a contact pressure of 0,6 ÷ 0,9 kGs/sm^2 to soil, for example, southern carbonate black earth [52, 82]. Practically the values of (σ') lie within these limits.
The obtained research results are used as a basis for the constructive-technological scheme of the seeder-comber SGS-4,2, intended for cultivating soy and other tilled crops by comb technology in the irrigated zone of Southern Kazakhstan.

The results of preliminary acceptance tests showed that theSGS-4,2 was satisfactorily aggregated with the T-150 tractor at speeds of 7,5 km/h. The seeder formed a trapezoidal crest resistant to destruction, 16 sm in height and a horizontal platform width of 37,5 sm and with a distance between the centers of the crests of 70 sm. Two rows of seeds with a spacing of 15 sm were placed on the ridge, and they were buried at a depth of 4,8 sm. Fertilizers with a given norm were applied to the crest separately from the seeds.
Compared with single-purpose machines used for soybean cultivation under these conditions, SGS-4,2 provides an annual economic benefit of 180-220 man-hours and a decrease in operating costs by 25-30%.

3.5. Working organs of sowing machines for introducing seeds into the soil into a given depth

The fulfillment of agrotechnical requirements for the uniform placement of seeds in the furrow and at the same depth from the surface of the field depends on how the furrow forms and how the seeds come in, that is, the shape and size of the grain flow when they fall to the bottom of the furrow. Thus, the overall picture of the opener is characterized by the combination of two independent, but interrelated processes - the arrival and distribution of seeds in the space bounded by the working surfaces of the opener, and furrow formation.
After several mechanical treatments of stubble fumes, the soil on them is sufficiently loosened and there is practically no stubble left on the surface of the field. As a rule, a significant supply of moisture accumulates on the steam fields, which makes it possible to carry out sowing in a narrow row. Therefore, it is recommended to sow theSZP-3,6 seed drill with the soil protection system of agriculture on the fallow fields. The working body of this seed drill is a two-disc coulter, which presents less stringent requirements to the quality of soil cultivation and the state of the field.
Dimensions and shapes of the groove, opened by the opener, depend on the relative position of the disks, characterized by the angle between them and the height of the location of the joint points of the edges of the disks, operating modes and physical and mechanical properties of the soil. These factors determine the nature of the movement of soil particles under the influence of the disc, and, consequently, the colter's ability to form a furrow of the required size and lay the seeds to a given depth.
Double disc coulters spread the soil to the sides with a small displacement of its particles and compact the walls and bottom of the groove. The width of the groove b between the lower points of the cutting edges, both disks, is determined as follows (Fig. 98).
The opener discs are placed one at the other at an angle ψ = 10 - 11^0 and touch the edges at the point m located at an angle α to the vertical. If the vomer is cut with the plane $m0m_1$ passing through the center of the discs, then in the section we get the true value ψ of the disk mounting angle. Having designed the point M by the radius m0, we get the point B, and projecting the latter onto the middle line $m0_1$, between the disks, we get the point c:

$$cB = mB \cdot sin\,(\psi/2) = b/2.$$

Since $mB = m0 - B0 = R - Rcos\alpha = R(1-\cos\alpha)$ then
$$cB = R(1-\cos\alpha)\cdot sin\,(\psi/2) = b/2, \text{or } b = 2R(1-cos\alpha)\cdot sin\,(\psi/2).$$

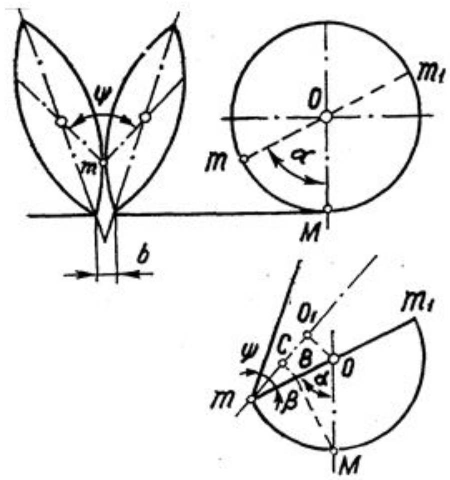


Fig. 98. Scheme for determining the width of the groove formed two-disc colter

With known values of the angle and radius of the disc R, it is easy to set the optimal values for the groove width b at the maximum depth of the disc travel. At the same time, the larger the angle α, the greater b, and at α = 45-50°, the width of the groove becomes so great that the discs begin to work separately, carrying two separate burrs between which a ridge is formed. In this case, wide-seed sowing becomes impossible. But this circumstance was successfully used when creating a two-disc coulter for narrow-row seeding (Bogachov'scolter).

Two-disc coulters push the soil aside, at the same time lightly lift it up and shift it forward. As a result, a pre-coulters soil hill is formed in front of the opener, the width of which determines the placement of the openers in one row in the transverse direction, i.e. $a_c > B_x$, wherea_c - is the distance between the vomers and B_x is the width of the pre-coulters soil hill. At $a_c < B_x$, the pre-coulters soil hills join to form a continuous felling of the soil, soil is congested, contributing to plowing. The process of furrow formation is disrupted.

The depth of seed filling, flowing into the groove cavity, will depend not only on the depth of the colter's stroke and the width of the groove, but also on the nature of the scree of soil mass that occurs under the opener [45].

During the colter's progress in the soil at any depth in the subsurface cavity, an inclined surface is formed from the crumbling soil, and the slope of this surface is directed towards the tip of the opener. The incoming seeds will be located on this inclined surface and be closed at different depths. With the increase in the speed of movement, the process of soil screeping the soil in the sub-coulters space proceeds more intensively, which, apparently, increases the unevenness of the seeding of the seeds in depth. This pattern is observed with a constant depth of the coulters (they were hardened on the frame of the laboratory installation) with an increase in the speed of movement from 9 to 15 km/h. Different depth of seed embedding is explained not only by the effect of translational velocity, but also by the influence of soil particles moving upward along with the surface of the rotating discs when leaving the groove. This phenomenon was established by the method of radiography and high-speed shooting [83].

The analysis of the scheme and designs of the two-disc coulters made it possible to establish that the following factors affect the uniformity of the seed placement in depth: the location of the guide relative to the axis of the discs; seed supply direction; presence of reflector. A method of dispersion analysis revealed [83] that at working speeds of the seeder 2,2-3,3 m/s, the effect of the seed supply direction in the double disc colter on the quality of seeding in depth is 3,2-10,2%; the direction of the guide relative to the axis of the disks is 8,3-26,6%, the reflector is 47,3-69,8%.

The uniformity of the laying of seeds improves with the direction of their flow in the direction opposite to the movement of the opener, and into the area where the disks enter the soil. When the seeds are fed to a seed hopper located in the front of the opener, the discs, at the time of entry into the soil, carry the seeds to the lower layers and help to lay them on the bottom of the furrow. Part of the seeds from the total flow directed to the side, which is opposite to the movement of the

opener, enters the bottom of the furrow with a lower incidence rate. For this reason, the number of seeds in the upper soil layers from the bottom of the furrow decreases.
A vomer with a funnel located in front of the axis of rotation of the discs and feeding the seeds to the side, in reverse, significantly improves the quality of sowing, laying the seeds in three one-centimeter horizons. However, with an increase in the speed of the seeding unit from 9 to 12 km/h, the unevenness of the laying of seeds increases so much that even changing the direction of their feeding (against or along the way) does not correct this. There is a need to introduce reflectors in the vomer to hold seeds at the bottom of the sulcus. The lamellar flexible reflectors proposed by the NGO VISKHOM and the Kirovohrad PKI for tillage and seeding machines [84] were blocked from the rear between the discs and from the top between the groove walls at a distance of 15 mm from the bottom, with a 270 mm extension of the back of the reflector back beyond the discs from their axis. In laboratory conditions, with a seeding depth of 8 cm and seeding rates of 9 and 12 km/h, the standard deviation values for the standard opener were 0,83 and 0,88 sm, respectively, and when equipped with an elastic reflector, the mean square deviation was reduced to 0,57-0,62. The number of seeds in a one-centimeter layer, located at a given depth of 8 sm, increased from 38-56% to 59-65%. The total number of layers with seeds decreased from five to three.
The uniformity of the seeding depth is also strongly influenced by the depth of the colter stroke, which depends on the coulter attachment system to the frame, and on the size and direction of the acting forces. The system of fastening two-disc coulters of grain seeders is individually-driven and single-hinged.
Typically, the action of colter forces under this suspension is presented in a simplified form, assuming that the forces acting on the opener can be reduced to three forces: the weight G of the opener with leashes (including the additional force from the spring pressure), the thrust force P and the resultant of all the forces of resistance R. For equilibrium, it is necessary that $\bar{G} + \bar{P} + \bar{R} = 0$. However, due to the variability of R, the colter's equilibrium is continuously broken and this condition can be considered as an instant one, therefore it is possible to estimate the stability of the colter's progress and the influence of the opener's parameters and the speed of the seeder's motion on the basis of an analysis of the dynamics of the colter's movement in the soil.
In simulating the effect of soil on the opener, the coefficients of proportionality of the change in the positional force K_1 and the damping C_1 of the rheological model of the soil are chosen as generalized parameters of the equivalent idealized mechanical system of the soil-colter-weigher and colter penetration mechanism (П-С-МНЗС) [85], and also reflecting all the design features of the МНЗС of various types, the generalized mass of the system m and the stiffness coefficient of the МНЗС (the stiffness coefficient of the equivalent elastic element) K_2, reduced to the center of inertia (center mass colter) (Fig. 99).
The coefficient of proportionality of the change in the positional force can be expressed by $K_1 = R_{\text{в}}/a$, where$R_{\text{в}}$ - the vertical component of soil reaction; a is the depth of the colter.

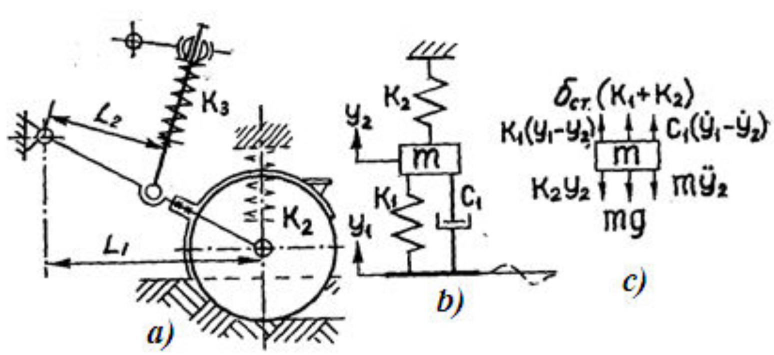


Fig. 99. The scheme for replacing the parameters of МНЗС and disturbances on the part of the soil by the generalized parameters of the П-С-МНЗС system: *a*) - the design of the МНЗС; *b*) - equivalent idealized mechanical system with generalized parameters; *c*) - is a diagram of forces acting in an idealized mechanical system

However, the change in the vertical component of the reaction of the soil when the vomer moves along the unevenness of its surface depends not only on the depth of the stroke, but also on the speed of its relative displacement. The action of the soil on the vomer in the idealized system is supplemented by a damping force C_1, that is, by the magnitude of the change in the force per unit of change in the relative velocity of the vertical vane displacement.
The stiffness factor K_2 of the MH3C can be expressed in terms of the stiffness coefficient K_3 of the elastic suspension element

$$K_2 = K_3(L_2/L_1)^2,$$

where: L_1-the action arm of the vertical component R of the soil reaction relative to the hinge of the leash; L_2 - the arm of the action of the force of the elastic element with respect to the hinge of the leash. Using the numerous data on the unevenness of the soil surface (spectral densities), it was concluded that, in order to obtain the required uniformity of the seeding depth, the coulters of grain drills must copy irregularities with an amplitude of 20-50 mm and a pitch of 1,4-3,0 m, because the irregularities with an amplitude of less than 20 mm are crushed by the opener, and unevenness with a step of more than 3 m is copied by the seeder, rather than by individual openers.
Taking the perturbations of the unevenness of the soil surface as periodic, the analytical expression for determining the relative displacement of the opener is expressed as (85):

$$y = \frac{(m\omega^2 - K_2)y_1 \sin(\omega t - \psi)}{\sqrt{(K_1 + K_2 - m\omega^2)^2 + C_1\omega}}, \tag{175}$$

where: y_1 - amplitude of unevenness of the soil surface; ω - is the angular frequency of the disturbing effects of the unevenness of the soil surface, equal to the angular velocity of rotation of the vector representing harmonic oscillations.
The angular frequency can be found from expression

$$\omega = 2\pi \cdot v/l, \tag{176}$$

where: v - speed of the seed unit; l - is the step of unevenness of the soil surface.
The angle ψ is proposed by the formula

$$\psi = arctg\frac{C_1\omega}{K_1 + K_2 - mv^2} \tag{177}$$

.

Dependence (177) is proposed for the experimental determination of the damping coefficient C_1 in terms of the magnitude of the phase displacement of ψ vibrations of the coulters under investigation with respect to artificial soil irregularities. Based on the operating conditions, the speed regime and the parameters of the openers, a nomogram is constructed that allows to determine the optimum parameters of the MH3C taking into account the unevenness of the fields (Fig. 100).
In the upper part of it horizontal lines are plotted, corresponding to the agrotechnical tolerance according to the depth of seeding ± 10 mm. At$У_1$ = 50 mm, the tolerance is 1/5 of the scale У, with $У_1$ = 40 mm - 1/4 of the У scale, and so on.

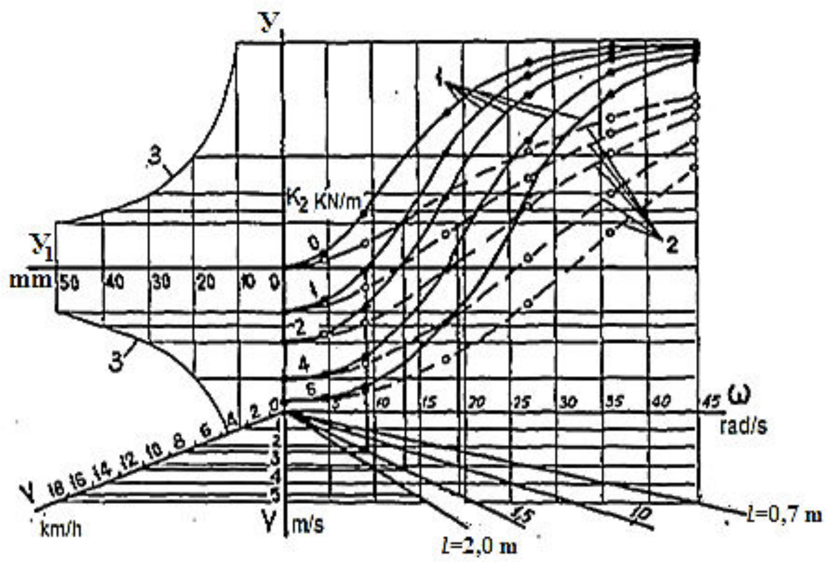


Fig. 100. Nomogram for determining the optimum parameters of the MH3C and the speed of the seed unit: C_1 = 0,3 kNs/m; K_1 = 4,0 kNs/m;$У_1$: 1 - at m = 12 kg, 2 - at m = 6 kg; 3 - Δy = f ($У_1$)

The nomogram shows an example of determining with its help the capabilities of two openers with a mass of 12 and 6 kg installed on the sameMH3C with a stiffness factor K_2 = 980 N/m at $У_1$ = 50mm and l= 1,5m. If, as the nominal seed rate of the aggregate to take v = 8 km/h, then at l = 1,5m $\omega = 2\pi v/l$ = 9,2 rad/s, which can be established without resorting to calculations, and using the lower part of the nomogram. The stiffness coefficient of the serial seeder is SZ-3,6 K_3 = 4,9 kN/m, and $K_2 = K_3(L_2/L_1)^2 = 1$ kN/m The vertical corresponding to ω = 9,2 rad/s crosses the curve (1) at K_2 = 1 kN/m, corresponding to the mass of the vomer, m = 12 kg, almost at the level of the zero mark of the y axis, and curve (2) corresponding to the mass of the opener m = 6 kg, slightly lower. Moving upwards to the right along the curves (1) and (2) to the intersection with the horizontal, denoting agro-admission at the deviation of the seeding depth from the unevenness set at the amplitude $У_1$ = 50 mm, and then down to the intersection with the beam l = 1,5 m, and to the left before crossing with the speed scale, it can be verified that for the colter m = 12 kg, the speed limit is 11,5 km/h, and for the colter m = 6 kg - 17 km/h. As a result of the studies carried out [85], the following parameters of the MNSA for a two-disc colter are recommended: m = 12 kg, the limits of the change in the bending force applied to the colter 200 ... 400N, the stiffness factor K_2 = 1,86 kN/m, the spring action lever of the L_2 levers for the front row of coulters 280, and the back of the coulters - 485 mm; have adjustment of the deeper force from 0 to 0,3 kN with an interval of 0,05 kN and a stiffness coefficient ranging from 0,6 to 2,2 kN/m with an interval of 0,1 ... 0,2 kN/m.
If there is a significant amount of stubble on the surface of the field, seeders with double-disc coulters are ineffective - they are clogged with stubble residues and deformed soil, they do not deepen on compacted soil. Therefore, with many crop residues and high soil moisture for sowing cereal crops, it is recommended to use a LDS-6 seed drill, in which the working bodies are spherical discs assembled into batteries attached to the drill frame by means of a pivotal-radial suspension and spring-loaded. The seeds along the guide-vas deferens are fed into the furrow formed by the disc, from the side of the convex sphere. The seeds coming off the next disc are closed.
However, for all positive qualities, it has two significant drawbacks: a large unevenness of the seeding of the seeds in depth, i.e., practically the seeds are distributed throughout the thickness of the treated soil layer, and intensive spraying of the treated layer on light soils in mechanical composition that can cause wind erosion. Unfortunately, almost no one is working to remove these shortcomings in our country.
In ordinary years, that is, when the soil has the optimum moisture content, the sowing of cereals with the soil protection system of agriculture is carried out by seeders with tubular coulters

equipped with napkins, without overlapping with the pre-sowing soil treatment, or by a seeder with tubular openers, in which the napkins are replaced by a flat-edged lancet paw small width of capture. In this case, the seeding and pre-sowing operations are combined.

Tubular vomer with cultivator point and lancet paws is row cultivation with a spacing of 23 sm, and when placed in the sub-satellite space of special switch-gears - a wide strip to 12 - 13 sm or a continuous scattered crop.

The geometrical parameters of the tubular colter with the nose-piece (pipe diameter, width of the nose-piece, etc.) were not substantiated, but were accepted mainly based on strength and constructive considerations, from the condition of free passage of seeds and fertilizers along its internal diameter and the formation in the soil of a groove of the necessary width and depth.

For the selection of the optimum values of the geometrical parameters of the lancet flat-paw to the tubular colter (pawter), the same principles and provisions are used that are used to justify the optimal values of the geometric parameters of the wide-cut planar paws described in Section 2.2.1.

Features of the opener paw: treatment of moist soil to a shallow depth, in which the processes of soil sticking the working surfaces of the paw and the enveloping of its blades by plant roots intensify; the need to create a subsatellite space of the right size for the implementation of scattered sowing; Strict requirements for the uniformity of seeding the depths cause a more careful choice of its geometric parameters.

Many researchers recommend the following values of their geometric parameters: the angle of the solution is 60 - 65°, the angle of placement of the share to the bottom of the furrow is 20 - 28°, the width of the seizure is 290 - 450 mm and the height of the formation is 25 - 40 mm.

The penetration of seeder-cultivators when working on stubble backgrounds, as well as anti-erosion cultivators, depends on the parameters of placement of the paws on the frame. Studies [52] found that the passability of the seeder-cultivator with opener paws, whose parameters are within the above limits, is ensured with a distance between the legs in the row of not less than 450 - 600 mm, and between the rows of 500 - 700 mm. Therefore, to obtain the minimum possible row spacing, the coulters on the frame of the seed drill are placed in 2-3 rows. This makes it possible to obtain a spacing of 23 sm at the width of the foot of 290 mm, which ensures the passability and efficiency of the cultivator on stubble backgrounds. But with the inter-rowing and the large seeding rate there is a thickening of the seeds in a row, and with a separate harvesting method, sagging is noted up to the soil surface, which entails an increase in losses, especially with a small number of plants per 1 sq. Km. In addition, the placement of the paws in three rows predetermines different conditions for their work: in the first row they work under conditions of non-free cutting, in the second row - semi-free, and in the third row - under conditions of free cutting. Accordingly, the amount of soil support changes, causing a large shift of the soil with the third row of openers in the sides. Therefore, as shown by experiments, the difference in seed depth between the first and third rows can reach 3 - 5 sm.

The difference in the depth of the seeding of seeders between cultivators between the first and third rows of paws can also be obtained since the feet of the first row, working in conditions of not free (blocked) cutting, have greater resistance compared to the third row operating under conditions free cutting. Since the racks of the opener coulters are hinged and spring-loaded to the frame, the first-row openers can deviate in the longitudinal-vertical plane by a larger angle from the vertical than the third, which leads to a deep-seated embedding of the seeds (Fig. 101).

To determine the maximum allowable tilt angle of the opener paw relative to the surface of the field in the longitudinally vertical plane, a formula is proposed [87]

$$\theta_{dop} = arcsin\frac{2\Delta h \cdot \mathrm{tg}\,\gamma}{K \cdot b\sqrt{\mathrm{tg}\varphi_0 . \mathrm{tg}\,\gamma}}, \tag{178}$$

where: Δh - tolerance of seed placement by depth allowed by agricultural demand; γ - half of the angle of the colter opener; φ_0- is the angle of the natural slope of the soil; K - is a coefficient that depends on the operating conditions and characterizes the conditions for seed distribution in the soil ($K \cong 0{,}9$);
b - working width of the opener paddle without taking into account overlap between adjacent paws, that is, $b = B$ - (30...40 mm); B - constructive width of the paw-opener.

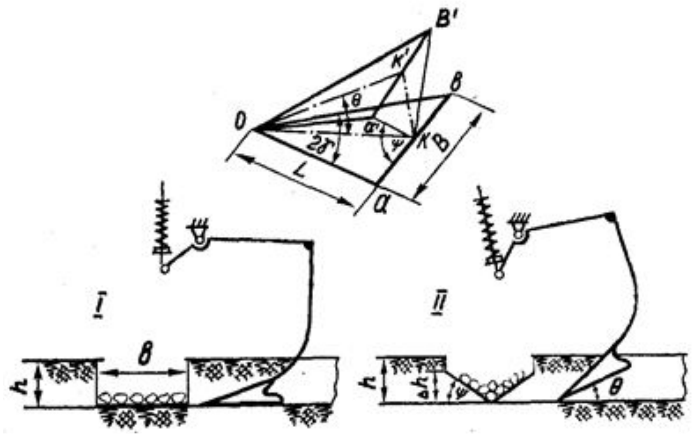


Fig. 101. Seed arrangement in soil in case of deviation of paw-opener in the longitudinal-vertical plane by an angle θ>0

With the optimal parameters of the opener paw, the set seeding depth (taking into account the agrotechnical tolerance) is maintained at θ_{dop} ≅3°. Uniform distribution of seeds along the width of the sown strip is preserved at θ_{dop} = 0-6^0. And further increase in the number of seeds increases the concentration of seeds to the axis of symmetry of the opener paw, when it is unevenly sown, it turns into a ribbon belt, and when the belt is sown, the width of the belt decreases so much that the crop turns into a row seed.
It is possible to avoid seed filling of the seeds between the first and third rows of the paws due to the change in their spring stiffness - greater stiffness should have the first-row openers, the smaller - the second and the even smaller - the third. Unfortunately, there was not enough research on this topic. In [25], studies on the determination of the main parameters of the suspension - the working organs of seeder cultivators (Fig. 102) - were summarized, Table 21.

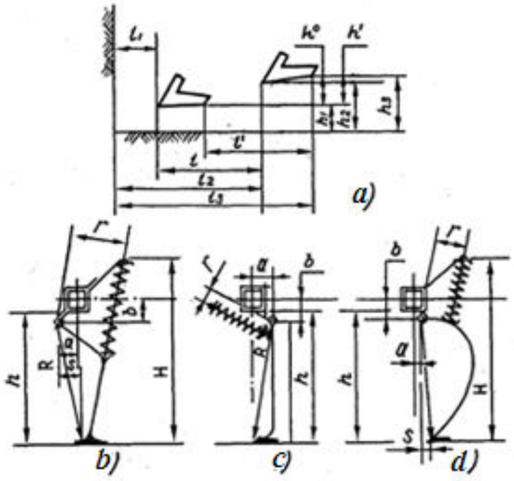


Fig. 102. Schemes and parameters of the suspension of working bodies seeder-cultivators: *a*) - moving the paw-opener;
b) - SZS-6/12; *c*) -SZS-2,1; *d*) -SU-279 (Canada)

Table 21-Parameters of hangers of working bodies

Suspension parameters, m	Variants of values of suspension parameters, m				
	SZS-6/12			SZS-2,1	SU-279
	1	2	5	3	4
Spring:					
length	0,470	0,300	0,340	0,300	0,240
diameter	0,067	0,069	0,065	0,069	0,061
diameter of the wire	0,012	0,014	0,014	0,014	0,012
H		0,852		0,660	0,860
h		0,720		0,526	0,650
S		0,050		0,100	0,120
R		0,510		0,60	0,590
r		0,240		0,100	0,110
a		0,095		0,120	0,025
b				0,075	0,110

From the analysis of the results of studies of suspension variants (Fig. 103) it follows that the suspension of SZS-6/12 with the experimental spring (Curve 1) has the required elastic properties. It begins to actively deform at a load of 0,5-0,7 kN. In this case, the value of the angle θ, which is in expression

$$\theta = arc\text{tg}\frac{h_3 - h_2}{l_3 - l_2},$$

does not exceed the permissible value $\theta \approx 2°$.
Suspensions with springs from SZS-2,1 (curves 2 and 3) (Fig. 103) showed unsatisfactory results in view of the high rigidity of both the spring and suspension in general. Springs began to actively deform at a load of 1 kN, which adversely affects their ability to perform safety and damping functions, which can lead to deformation or breakage of the paw when it hits the obstacle. The vertical displacement of the toe of the paw, depending on the horizontal movement of the paw, depends on the horizontal movement of the paw at the suspension SZS-2,1 (Curve 3) in the working load area (0,5-0,7 kN), 8 mm deep, and the angle θ exceeds the permissible value. This is due to the kinematics of the suspension (the hinge is located behind the toe of the paws), which in the field can lead to breakage of the working organs and their dehiscence.
Suspension of cultivator paws SU-279 (Curve 4) has a nonlinear elastic deformation. This is due to the presence in it of two elastic elements - springs and pillars. The best performance for the suspension SZS-6/12 (Options 1 and 5). It can respond to fluctuations in the load, and the values $\theta$and h do not exceed the allowable limits.
In order to ensure a steady movement along the depth of the working organs of seeder cultivators or seedbeds, the suspensions of which are made according to the above schemes (see Fig. 102), the ratio of the decay moment from the compression force of the spring and the receding moment from the soil resistance of the paw plays an important role.

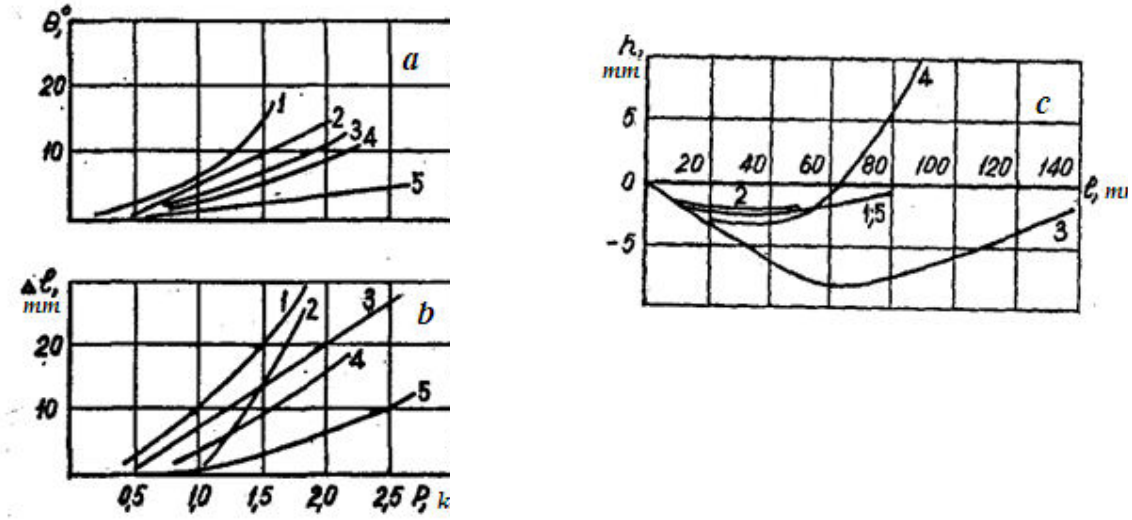


Fig. 103. Angular deviations of the paw ***a*** and horizontal movementher toe ***b*** depending on the load: 1- SZS-6/12 with an experimental spring; 2 - SZS-6/12 with a spring from SZS -2,1; 3- SZS -2,1; 4- SU-279 (Canada); 5 - SZS-6/12; ***c*** changing the height of the toe of the feet with different parameters of the suspensions depending on their horizontal displacement

The use of insufficiently rigid springs or other resilient elements leads to a frequent exit of the working bodies from a stable operating mode and to a violation of a given depth of seed sowing or tillage. Excessive tightening of them is unacceptable, since the working element will not be deeper when the obstacle is hit and thus will be susceptible to breakage. The working organ of the seeder-cultivator SZS-6/12 consists of a stand with a cultivator paw, which is pivotally connected to a bracket rigidly fixed to the beam of the frame, a compression spring with a guide rod, which connects the stand to the bracket. The guide rod is fixed to the stand of the working element at a point lying along a straight line passing through the geometric center of the contour of the cultivator foot D' and the point of connection of the rod and the bracket (Fig. 104).
As shown above, the operating element with this suspension worked more stably than the rest of the suspension schemes.
However, the direction of the axis of the rod through the geometric center of the paw of the working member does not ensure a stable position of the paw of the working organ when moving in the soil and a guaranteed return to its original position with a sharp increase in the resistance of the soil to the movement of the paw. The reason for the lack of this is that the component of the soil resistance force is constantly acting on the compression spring of the working element R'_{xy} (see Fig. 104), pushing the spring upwards and preventing the working element from holding it in a stable equilibrium, and also the presence of a large (not optimal) leg $l'_2 = l'_1 + \Delta l'_1$ for which the depressing moment $M_{o.з} = P'_{np} \cdot l'_2$ created on his shoulder becomes less than the deeper moment $M_{o.в}$. This action of forces and moments on the working organ does not ensure its steady movement in the soil at a given depth of loosening. In order to eliminate this drawback, a working body is proposed, the scheme of which is shown in Fig. 105, and the scheme of action in the process of work of forces and moments in Fig. 106.

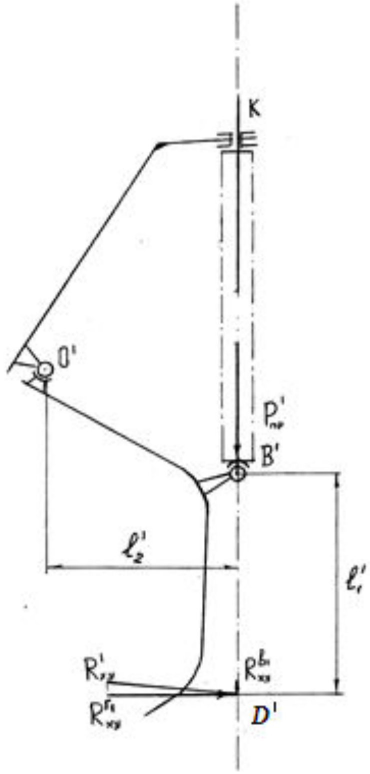


Fig. 104. The scheme of the forces on the worker organ of seeder-cultivator SZS-6/12

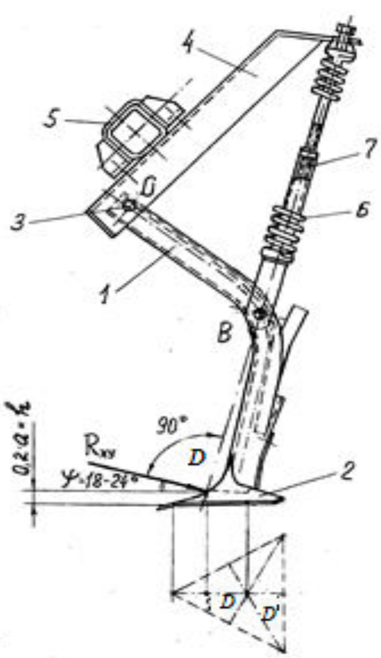


Fig. 105. Scheme of the proposed working tool for seeder-cultivators or pre-seeding tillage tools

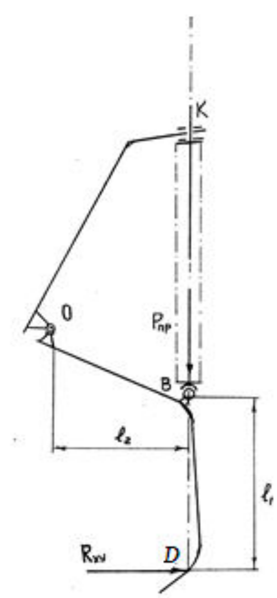


Fig. 106. The scheme of the forces in the process of work on the proposed working body

The working body (Fig. 105) includes a rack 1 with a cultivator foot 2 connected by a hinge 3 (point 0) to a bracket 4 rigidly fixed to the frame bar No. 5. The post 1 is additionally connected pivotally at point B to the bracket 4 via a compression spring 6 with a length-adjustable guide rod 7 such that their axis extends perpendicularly to the resultant force of the soil resistance R_{xy} to the working member at the point D of its application to the foot 2. In this case, the force R_{xy} acts on the paw 2 of the working element at an angle ψ = 18 - 24°. The guide rod 7 in the operating member is connected to the bracket 4 with the possibility of moving along its axis with respect to the bracket No. 4.
Stabilization of the position of the proposed working organ during its movement in the soil occurs as follows. When the working element is mounted on the frame of the seeder-cultivator or on the implement for the pre-sowing treatment of the soil, the spring 6 is pre-compressed by changing its length by the rod 7. Compression is carried out to the force P_{pr} ensuring compliance with the condition (Fig. 106)

$$P_{pr} \cdot l_2 \cong R_{xy} \cdot l_1; \tag{179}$$

$$\text{or} \quad M_{o.з} = M_{o.в}, \tag{180}$$

where: P_{pr}- force of compression of a spring 6;
l_2- the lever between the points O and B;
R_{xy} - resistance of soil;
l_1 - the lever between the hinge B and the point of application of the resistance force D;
$M_{o.з} = P_{pr} \cdot l_2$ - deepening of paw 2 moment;
$M_{o.в.}$=$R_{xy} \cdot l_1$- extending paw 2 moment

Under the conditions (179) and (180), the paw 2 is in a position of stable equilibrium and stable movement in the soil at the established depth of loosening. In this case, the ratio of the arms will be l_1/l_2 =1. If the resistance of the soil to the working element increases, then the deepening moment $M_{o.в}$also increases, which, when at R_{xy}>P_{pr} tending, tends to turn it relative to the hinge 3 (point O). At the same time, the resistance of the soil will increase sharply, as the angle of entry of the paw 2 into the soil increases. At the same time, the force of compression P_{pr} of the spring 6 is increased, and a moment comes when the bending moment $M_{o.з}$ becomes greater than the depressurizing moment $M_{o.в}$ and the working member returns to its original position. Thus, at l_2/l_1 =1 the system ensures the movement of the working member at a fixed depth in the stable equilibrium mode.
In the working organ SZS -6/12 (see Fig. 104), the axis of the spring and rod passes through the geometric center of the paw of the 2 working elements (see Fig. 105, point D'). The resultant force $R'_{xy} = R_{xy}$, acting $R_{xy}^{г_1}$ at point D' and decomposing into components acts on this working element (see Fig. 104) and $R_{xy}^{в_1}$ the force $R_{xy}^{г_1}$ creates a deeper moment $M_{ов} = R_{xy}^{г_1} \cdot l'_2$, where $l'_2 = l_2 + \Delta l$, and the component $R_{xy}^{в_1}$ acts along the axis of the spring, compressing it.
The equilibrium condition of the system (the stable path of the paw) will be

$$P'_{pr} \cdot l'_2 = R_{xy}^{г_1} \cdot l'_1 . \tag{181}$$

Because the $l'_1 = l_1$, but $l'_2 = l_2 + \Delta l$ and $P'_{pr} = P_{pr}$ we have

$$P'_{pr}(l_2 + l)= R_{xy}^{г_1} \cdot l_1, \tag{182}$$

$$\text{or} \quad M_{о.з} = M_{о.в} , \tag{183}$$

where: P_{pr}- force of compression of a spring;

$l_2' = l_2 + \Delta l$–the shoulder between the points O' and B';
$R_{xy}^{\Gamma_1}$ - component of the resistance force R_{xy}, creating a deeper moment;
$R_{xy}^{B_1}$- component of the resistance force R_{xy}, creating a force on the spring;
$l_1' = l_1 + \Delta l$ – the shoulder between the points B' and D';
$M_{0.3} = P_{pr}\ (l_2 + \Delta l)$- deepening moment of the paw 2:
$M_{o.B} = R_{xy}^{\Gamma_1} \cdot l_1$- receding moment of the paw 2.

If the conditions (182), (183) are satisfied, the claw 2 of the CZS-6/12 moves in the soil in the unstable equilibrium mode, since $l_2/l_1 = l_2' - \Delta l$, i.e.$l_2/l_1 < 1$. This is explained by the fact that in this case the increase in the shoulder $l_2' = l_2 + \Delta l$, and the reduction due to the force $R_{xy}^{B_1}$ acting on the paw of the force from the preliminary compression of the spring P_{pr} will accelerate the outward growth of the receding moment with increasing R_{xy} and the output of the working member beyond the boundaries of equilibrium motion and will not allow it to return in the shortest possible time to initial position.

Thus, the proposed operating element, due to the choice of the optimum direction of action of the compression force of the spring P_{pr}, ensures the movement of the working element in the soil with a sharp increase in its resistance in the stable equilibrium mode. Therefore, it is advisable to establish a proposed working body for them when developing new models of seeder-cultivators or tools for pre-sowing tillage.

To reduce the negative properties of seeders-cultivators, due to the wide spacing (22,8 sm), distributing devices are placed at the outlet of the seed flow from the tube-vas-deferens in the subsurface space, which, depending on their type and parameters, allow for the same placement of the paws on the same The frame of the seed drill should be planted in a wide strip or in a spreading manner. The use of scattered or banded sowing allows to distribute the seeds more evenly than in the case of a row seed, according to the area of nutrition, and hence the yield increases, the weediness decreases, and so on.

It should be noted, however, that in the literature about the advantages of scattered (or striped) sowing before the row, experts of an engineering profile more often approve. Analysis of the results of research by agronomists on this issue does not give an unambiguous answer. Unfortunately, such a trend, when only mechanical engineers are engaged in determining the influence of a technological process or process on crop yields, for example, the effect of soil compaction, root-mean-square deviation in depth during basic tillage, etc., on yield, is increasingly spreading in us in the country. From our point of view, these works must necessarily be carried out together with soil scientists and crop growers.

In connection with the foregoing, adaptations to the paws in the form of switch-gears in the sub-stem. space should be mainly considered as a means of increasing the universality of the use of seeder-cultivators depending on weather and climate conditions, such as soils, seeding rates, culture, predecessor.

Almost all the devices for distributing seeds to the seeder-cultivators for spreading the opener paw use the energy of free fall of seeds. Seeds falling on the distributor installed in the subsatellite space are reflected and distributed at the bottom of the furrow along the width of the shovel.

The shape and parameters of the distributor are chosen mainly based on two considerations: how to use the energy of free fall of seeds with a high efficiency and how to inscribe the flight trajectory of the seeds reflected from the seed distributor into a space that is limited in height. The height of the latter is determined by the stringent requirements for the working capacity of the opener paw as the working organ for pre-sowing soil cultivation, i.e. geometric parameters - the width of the paw knife and the angle of its inclination relative to the furrow bottom. Numerous studies have established that the height of the sub-satellite space for spreading should not exceed 0.4 - 0.6 depth of the opener.

Currently, the passive distributors of the paws of seeders in cultivators are divided into two types: with rectilinear (cones, prisms) and curvilinear generators (in the form of brachistochrone). The advantage of the latter is shown, and their parameters are determined [86].

In some works [88] it is indicated that the trajectory of the flight of seeds does not intersect with the vault of the sub-satellite space, if the velocity of the seeds after descent from the distributor is directed horizontally. In addition, the fact that the seeds from the vas-deferens do not fall just at the top of the distributor, but also at some distance from it, is not taken into account, which is one of the reasons for the significant deviation of the actual trajectory from the calculated one. At the same time, the seeds that changed the calculated trajectory create obstacles for those seeds that slide along the brachistochrone.

A theoretical analysis of the seed distribution process in the real conditions of the sub-satellite space was also carried out [86,89]. It is established that, taking into account the parameters of the arch of the longest flight range, the seeds are reached at

$$v_0 = \sqrt{\frac{gm^2}{k^2} \cdot \frac{1}{2(m-h)cos^2 a_0}}, \tag{184}$$

$$tg a_0 = \frac{k(2h-m)}{m},$$

where: $$k = \frac{H \cdot cos\theta}{B}; \qquad m = H - \frac{x_0\, Hcos\theta}{B}; \tag{185}$$

H - the height of the arch in the middle part of the paw-opener; 2B - the width of capture of the paw-opener; x_0 is the radius of the base of the seed distributor; θ - the angle between the plane of flight of the extreme seeds and the direction perpendicular to the direction of movement of the aggregate; g - acceleration of gravity.

When using the energy of free fall of seeds, the limit of initial speed control is small (practically the speed is not regulated, and it is about 1,6 m/s). Therefore, it is necessary to vary the parameters of the opener, although they can also be changed in insignificant limits, and the distributor, achieving the required values of the angle $\alpha_o$and the width of the seed distribution l (Fig. 107).

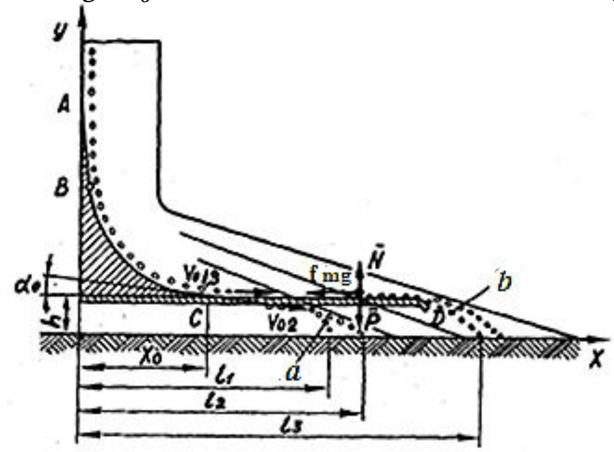


Fig. 107. Seed distribution in the socking space of the paw-opener: *a* and *b* - the trajectory of the seed flight when using the valves, respectively, without plates and with them.

Half the dispersion width l_2 is recommended to be determined taking into account the free flight of the seeds after descending from the distributor by expression

$$l_2 = \frac{v_0^2 \cdot sin2\alpha_0}{2g} + \frac{v_0^2}{g}\sqrt{v_0^2 sin^2\alpha_o + 2gh + x_0}, \tag{186}$$

where: h - is the height of the distribution base from the bottom of the furrow.

It is believed that the flight range of seeds under conditions of the subsatellite space can be significantly increased, and, consequently, the uniformity of their distribution is improved if the free flight of seeds is replaced after descending from the distributor by sliding along a smooth surface (see Fig. 107, section CD). At the same time, part of the kinetic energy accumulated by seeds during free fall, which was previously lost when the seeds hit the soil, will be used to increase the range of their flight. In this case, it is recommended to determine half the width of the dispersion by the formula

$$l_3 = x_o + \frac{v_0^2}{2fg}, \tag{187}$$

where: f - is the coefficient of friction of seeds against the surface of the distributor.
However, in this case, the "smooth surface", which should be located at a height h from the bottom of the furrow, practically closes the bottom of the bottoms from below, forming a narrow gap for the seeds to escape, which can be clogged not only by large-grained impurities entering the seeds, but also by the soil when the paws -openers, etc.
In connection with the design difficulties in creating a simple and reliable distributor, which would evenly distribute the seeds across the entire width of the opener, currently used distributors, providing a sowing strip with a width of 12 - 13 sm.
The uniformity of seed placement by depth with the opener paws depends also on the type of suspension mechanism on the seeder-cultivator frame.
An analysis of the experimental data shows [89] that the paw colter with different suspension mechanisms reacts differently to the variation of numerous perturbing factors. The greatest values of the root-mean-square deviations of the variations in the depth of the stroke (Table 22) are observed in the paw with the radial suspension mechanism ± (4,0 - 4,3)sm and the smallest - with the hinge ± (2,1 - 2,6)sm.

Table 22. Performance indicators of the paws with different suspension mechanisms

Type of suspension	The speed Speed of movement, km/h	Indicators of depth of seeding		
		m, sm	σ, sm	v, %
Radial	4.6	9.9	4.0	40.4
	5.2	9.6	4.3	44.7
	6.6	9.4	4.3	45.7
	8.4	8.3	4.2	50.6
Parallelogram	4.6	10.3	3.1	30.1
	5.2	9.9	3.4	34.3
	6.6	9.9	3.4	34.3
	8.4	9.0	3.2	35.5
Hinged	4.6	11.4	2.6	22.8
	5.2	11.4	2.2	19.6
	6.6	11.5	2.1	18.6
	8.4	11.7	2.4	20.5

With an increase in the speed of motion to 8,4 km/h, the values of the root-mean-square deviations change insignificantly. This is explained by the fact that with increasing speed, the frequency of the influence of the disturbing factors acting on the paw-opener is also increasing. The paw-opener, having damping properties due to the rigidity of the safety springs, works as a low-frequency filter, smoothing out high frequency disturbances. Virtually the paws with open-link mechanisms do not copy the irregularities of the microrelief but cut them off.
From the character of the flow of the amplitude-frequency characteristics (Fig. 108), it is evident that the greatest swing of oscillations from the action of soil resistance forces is observed in the paw with the radial suspension mechanism (A ≅2,4 sm), and the smallest at the opener paw with the hinge mechanism (A ≅1,0 sm). The differences in $A(\omega)$ determined the corresponding mean square deviation and seed depth (see Table 22).

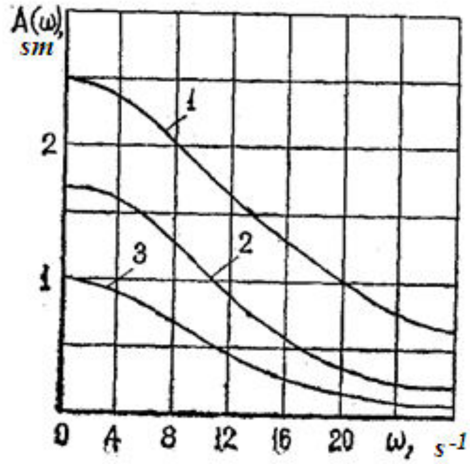


Fig. 108. Amplitude-frequency characteristics of the paw-opener with various suspension mechanisms:
1 - radial; 2 - parallelogram; 3 - hinged

It should be noted that the articulated linkage of the paw colters in the frame of the SZS-2,1 and SZS-2,1L seeders, as mentioned above, has not been designed correctly: the toe of the opener is much forward along the seed drill with respect to the seed drill projection of the hinge axis on the horizontal plane. When encountering an obstacle or with a short-term increase in resistance, the leg should bend back in the longitudinal-vertical plane, with its toe immediately beginning to grow deeper. However, carrying the toe of the paw forward with respect to the hinge does not allow it to do so, and in order to tilt backward over a certain angle, the toe of the paw must at the initial moment become even more dense. In this case, not only the traction resistance of the opener is sharply increased, but also the angle of inclination in the longitudinal-vertical plane relative to the field surface increases, which, as indicated above, leads to a deep-seated embedding of the seeds.
The diameter of the opener paw pipe (≈60 mm) promotes a sharp increase in tractive resistance, especially at increased speeds. In this regard, work was carried out to clarify the parameters of the paws and the scheme of its suspension of seeders-cultivators SZS -2,1 for work at higher speeds.
The results of the research made it possible to reasonably decrease the angle of the paw knife setting relative to the bottom of the furrow, to reduce the thickness of the opener paw, to remove the hinge of the paw hitch mechanism in front of its sock. Opener paws with specified parameters are used on seeders-cultivators SZS-6, SZS -12, SZS -8 and SZS -14.
However, these working bodies, which have a suspension scheme (see Fig.102-b), have several shortcomings, as indicated above. In this regard andas proposed a new working body for seeders or cultivators of pre-sowing soil, which is appropriate to use on newly developed machines and tools.

3.6. Working organs of sowing machines for post-sowingfield leveling and soil compacting

After the seed drill passes behind their working organs, which form grooves in the soil and seal seeds in them, the ridges and furrows of unacceptable dimensions remain on the surface of the field. This sharply reduces the uniformity of seed distribution in depth, leading to a decrease in field germination by 40 - 50%. For leveling ridges and furrows or bringing their size to the permissible agrotechnical (initial) requirements of the limits, seeded anti-erosion machines are equipped with leveling devices.
Cultivated plants show the highest productivity only at optimal soil density. Both low and high density impairs the water and air regimes, the conditions for germination of growth and development of crops. It is assumed that the optimal seed density of different soils varies between 0,9-1,1 g/sm^3. However, this is not entirely correct, since, for example, the initial density of sierozem of sandy loam (south of Kazakhstan) lies in the range 1,1-1,3 g/sm^3. Therefore, it is expedient and necessary to carry out work to justify the optimal density for the main soil types when sowing cereals. Such work for the southern carbonate chernozem of Northern Kazakhstan during the sowing of spring wheat was carried out at the All-Union Scientific Research Institute of Plant Protection [82]. Its main results are as follows:

- compaction of southern carbonate chernozem improves the contact of seeds with soil, promotes the acceleration of seed swelling;
- the best conditions for seed germination are added at a density of the super-seed soil layer of 0,95-1,0 g/sm^3 for narrow-band (5sm) sowing, and 0,9 g/sm^3 for broadband (8 sm) and clusters;
- with increasing soil moisture, the negative effect of over-consolidation increases. Practically, at a moisture content of 30% of the PPV, there is no need even for light packing. The most complete and amicable shoots of wheat are obtained at a soil density of 0,9-1,0 g/sm^3 - the average daily soil temperature in the 0-10 sm layer in a ten-day period after sowing was 1,3 degrees higher in the compacted row than in loose soil.

Based on these studies, it is recommended that:
- density of the soil of southern carbonate chernozem in the sowing layer should be 0,95-1,00 g/sm^3 at row packing, at a continuous one – 0,90 g/sm^3;
- in anti-erosion seeding machines, it is necessary to provide for the possibility of removing rollers, as when sowing in moist soil, the packing process disappears, and the rinks only complicate the sowing.

Unfortunately, for other types of soils and species of sown crops, similar studies have not been carried out in the soil protection system of agriculture.

The main requirements for seeding machine rollers for soil protection agriculture are not only to compact the seed layer of soil to the optimum density, but also to keep the maximum amount of crop residues on the field surface, spray the top (0-5 sm) soil layer, mixed with stubble residues, and do not clog.

Due to the variety and complexity of the processes occurring in the soil under the deformers and a large number of empirical coefficients in the formulas proposed for calculation, the theoretical justification for the type of rollers and their parameters ensuring the optimal density of the seed layer of the soil is not available today.

In order to choose the most promising, comparative tests on light (Southern Kazakhstan) and heavy (Northern Kazakhstan, Krasnodar territory) on the mechanical composition of soils of the following skating rinks were carried out: ring-shaped with a wedge-shaped rim; ringed with an oval-shaped rim; annular with a round rim shape and a distance between the rings of 70, 95 and 114 mm; closed ringed with a round rim shape with a pitch of 95 mm; ring-chain; ring-spar; ring with a rubberized rim (roller of atmospheric pressure).

Their evaluation was carried out according to the degree of soil compaction in the layer 0-5 and 5-10 sm; spraying the soil; preservation of stubble, sticking of the rollers with soil and clogging, traction resistance. The tests were carried out at speeds of 6-7; 8-9; 5 and 11-13 km/h.

Analysis of the results of the studies [90] showed that each of the rollers has advantages only in separate indicators, completely none of them meets the requirements of the agro-requirements, they are more responsive to the ringed, wedge-shaped rim, annular with a round rim shape and 114 mm, ring-sparrow.

A laboratory study of the compaction process by these three types of rollers was carried out [91] using a radiometric method for measuring soil density. The experiments were carried out with loads on the ring of the roller 6 and 9 N/sm^2 and the speeds of 6,3; 10 and 13,6 km/h on light and heavy in texture soils. The initial moisture content and density of the soil were adjusted to values corresponding to the conditions of operation of seeders when sowing on these types of soils. Density measurements were carried out along a grid (0,04x0,04 m) to a depth of 0,24 m, away from the axial line of the ice rink - up to 0,20 m.

The results of determining the degree of soil compaction (for sierozem sandy loam), depending on the type of roller, the speed of movement, the load on its ring and the type of soil are shown in Fig. 109;110 and 111. Similar curves were obtained for heavy soils with a mechanical composition. The best working out of a layer of soil in a zone of arrangement of seeds both on easy, and on the heavy ground on a mechanical structure, is provided with a ring-shaped rink with a round form of a rim. The ring-spiral roller gives deeper than a ring-shaped rink with a round rim shape, the location of the zone of maximum compaction of the soil, but the lowest was observed behind the ringed rink with a wedge-shaped rim.

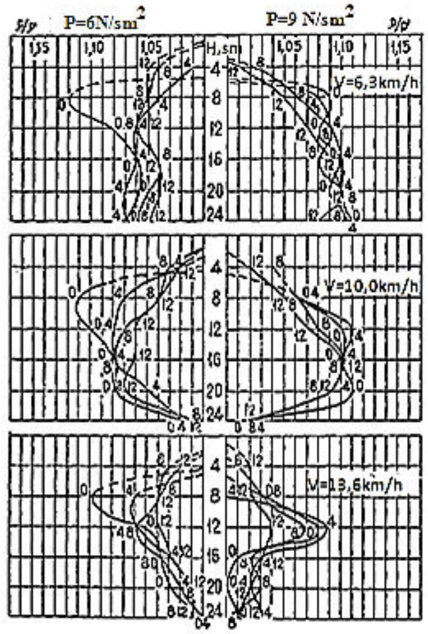


Fig. 109. The degree of compaction of the soil with a ring-shaped rink with a round rim shape (soil - light sandy-loamy sierozem)

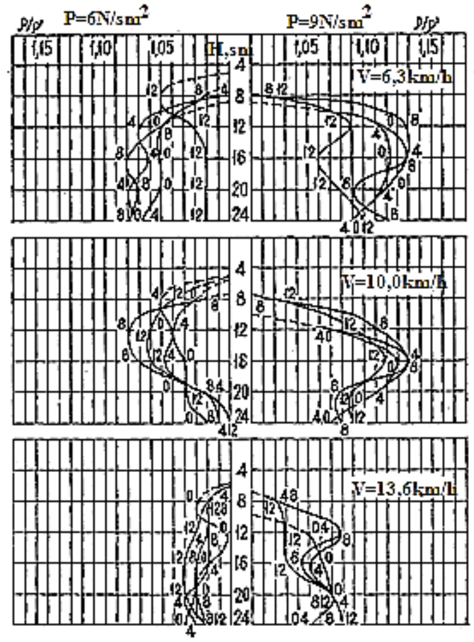


Fig. 110. The degree of compaction of the soil with a ringed roller with a wedge-shaped rim of the ring (the soil is light sandy-loamy sierozem)

With increasing load on the roller ring from 6 to 9 N/sm^2, the degree of soil compaction behind all the rollers increased by 25-30%, with the maximum compaction zone being located deeper than 7-8 sm.
With an increase in speed from 6,3 to 13,6 km/h, the degree of compaction is reduced by 20-25%, and the depth of working the soil is reduced by 15-20%.
When investigating the technological process of row dressing of crops with annular rollers, its optimal parameters and operating conditions are justified [52]. It has been established that the optimal parameters for the rows for rowing are: diameter of the roller 450-550 mm; the width of the rim is not less than 40 mm; The shape of the working surface of the rim is spherical (drop-shaped).

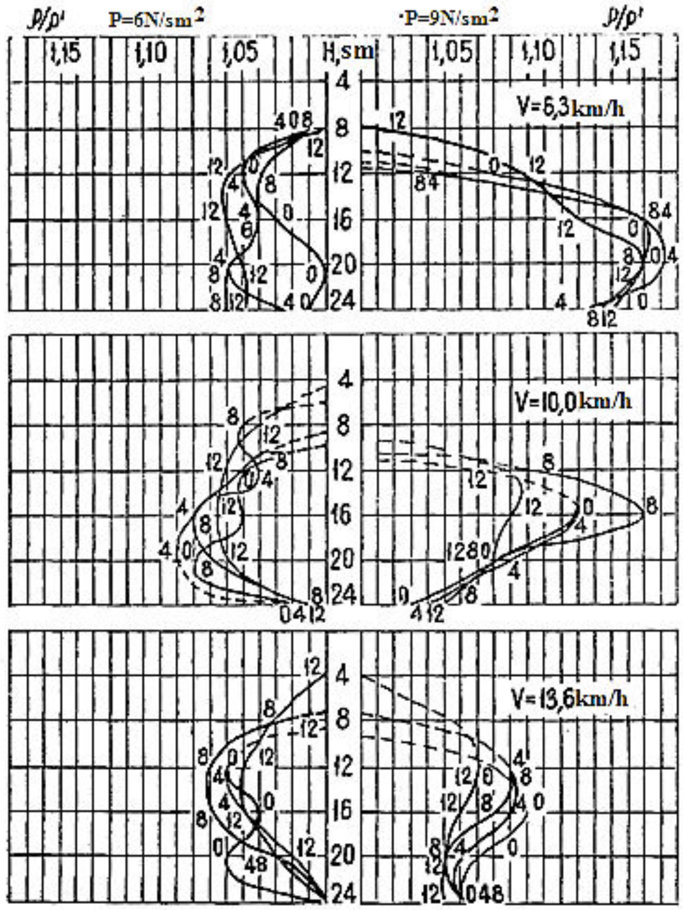


Fig. 111. The degree of compaction of the soil is ring-sporinated roller (soil-light sandy-loamy sierozem)

To obtain the density given by agrotechnological (initial) requirements, it is necessary to act on the soil (southern carbonate black earth) with a surface load of 6 - 9 N/sm^2. Application of higher pressures leads to excessive compaction and adversely affects the field germination of wheat and its further development.
The contact time of the crochet with the selected parameters should be at least 0,15-0,16 s, as otherwise the packing effect will decrease.
The traction resistance of the roller ring with a surface load of 6 N/sm^2 is 120 - 160 N.
The results of these studies were used to create the seeding devices for seeders-cultivators SZS-2,1, SZS-6, SZS-12 and SZS-8 for row soil coiling simultaneously with sowing.
In case of spreading or broadband sowing, the row-wise method of soil compacting is unacceptable. Analysis of the zones and depth of compaction of the soil under the influence of various types of rollers [91] allowed us to recommend ring-shaped skating rinks with a round rim for compacting the soil in spread or broadband sowing. With a roller diameter of 550 mm, the diameter of the ring rim should lie within 28-32 mm, and the distance between the rings is not less than 114 mm. These parameters practically provide not only a continuous compaction of the seed layer of the soil in the zone of seed deposition along the width of the sown strip, but also the drillability of the seeder along stubble backgrounds.
Rollers with such parameters were used to create a seeder-cultivatorSZS -2,1Л. However, it should be noted that the diameter of the rink is not justified, but adopted, based on the parameters of the kinematic scheme of seeders-cultivators SZS -2,1. This predetermines the need for research to determine the optimum diameter of the roller for stubble seeders.

Ring-shaped rollers with a wedge-shaped and round-shaped rim for row and broadband packing of crops not only compact the soil to optimum density, but also smooth the surface of the field, forming a shallow-grained surface. Such a surface improves the uniformity of the seeding depth and helps reduce the wind speed in the surface layer, thereby protecting the soil from wind erosion.
It should be noted that the seeding device of the seeder-cultivators has one significant drawback: they not only fulfill the function of compacting the soil, but also are the supporting and driving devices of seeders in operation and in transport. Therefore, when sowing on waterlogged soils, when it is not advisable to roll up the crops as mentioned above, it is practically impossible to remove the rollers and replace them with a screed device, since in this case it is required to change the basic kinematic and constructive scheme of the seeder. Research and development work on finding a new scheme for seeder-cultivators, which would allow for the replacement of the supporting and packing on a simple leveling device without significant expenditure of labor and time, are still under way.
Sealing of the soil after the passage of the lumberer-seeder LDS-6, working with high soil moisture, is impractical. Therefore, after the passage of the serfic discs of the LDS-6 seeder, the field surface is only leveled by the flat discs those have been assembled in the batteries at an angle to the direction of travel. It is possible to change the angle of attack of disks depending on working conditions.

4. Machines for intrasoil application of mineral fertilizers for soil protection agriculture

4.1 Basic agrotechnological (initial) requirements for machines

At present, the introduction of mineral fertilizers is one of the main ways to increase the yield of agricultural crops and preserve soil fertility. Therefore, work is continually being carried out to develop technologies and technical means to improve the efficiency of their application in the cultivation of a crop in specific soil and climatic conditions.
The introduction and development of grain-steamed crop rotations with short rotation, the field of pure steam with wings from high-stemmed crops, the strip placement of crops and vapors, the abandonment of dumping plowing and the transition to non-waste tillage by new machines that keep stubble on the surface of the field provide both effective protection of soils and crops from wind erosion, and significantly improve the moisture supply of plants. In the steam and other fields of crop rotation under this system, the plants make maximum use of mineral substances to form the crop. In the northern regions of Kazakhstan, where in the vast majority of ordinary and southern chernozem, as well as dark chestnut soils, there is a lot (especially in pure vapors) of available nitrogen and assimilable forms of potassium. This amount is sufficient to obtain a high yield of spring wheat. At the same time, here everywhere there is a lack of mobile forms of phosphorus, which determines the phosphorus starvation of spring wheat [44,92].
The soil protection system of agriculture improves the moisture content of plants, enhances the processes of nitrification and accumulation of nitrate nitrogen in the soil, and increases the mobility of soil phosphates. All this regulates the nitrogen-phosphorus regime, contributes to the fuller utilization of the accumulated moisture on the formation of the crop.
With the soil protection system of agriculture excluding the rotation of the reservoir, the continuous surface spread of mineral fertilizers followed by the imposition of soil tillage machines did not yield positive results. Such introduction of the main dose of mineral fertilizers and subsequent multiple cultivation of soil by cultivators-plane cutters contribute to their placement in the 0 - 6 sm layer. But as this layer of soil dries quickly, the efficiency of the fat significantly decreases [93].
Local application of fertilizers provides for their incorporation into the soil at a given depth in the form of a continuous screen, tape, stitch, row or nests. These methods imply a much smaller mix of fertilizers with soil, which contributes to a more efficient use of their plants and an increase in the yield of all crops, including cereals, by 0,2-0,5 t/ha.
An analysis of the results of studying the various methods of introducing mineral fertilizers in the arid conditions of virgin areas (according to the data of the All-Russian Research Institute for Experimental Research and Experimental Stations of Northern Kazakhstan) during the cultivation of grain crops shows [92] that:

- surface application of fertilizers with subsequent embedding by a complex of flat cutting machines is inferior to the effectiveness of deep, especially in dry years;
- in connection with the uneven use of phosphorus from various soil layers, the depth of application of fertilizers has a significant effect on phosphorus nutrition and the yield of spring wheat. Under the soil protection system of agriculture in the conditions of Northern Kazakhstan, the optimum depth of sealing the main dose of super-phosphate into the steam when applied under the first flat-topped treatment was 10-12 sm depth, and in the fall under deep loosening - 20 sm. For example, on an average for four years, the additional collection of grain of spring wheat from super-phosphate deposited at a depth of 10-12 sm amounted to 0,36 t/ha, for 20sm – 0,3 t/ha, and for 25sm – 0,25 t/ha; the uniform distribution of fertilizers is more efficient than the belt. On average, over the years of research, the yield increase of spring wheat with uniform application was 0,34 t/ha, and for tape – 0,17 t/ha.

The effect of super-phosphate, added to the steam in the form of the main fertilizer dose in the soil conservation system, is not limited to one year. The aftereffect was noted throughout the rotation of the four-field cereal crop rotation. So, for example, when applying the main dose of super-phosphate 0,060-0,080 t/ha to the pure vapor of the active substance to a depth of 15-25 sm, the yield of spring wheat increases by 0,25 - 0,35 t/ha in the first year of operation, and taking into account the two-year after-effect, the yield increase reaches 0,8-1,0 t/ha. Coupled fertilizers have a long aftereffect, which continues throughout the period.

In soil-protecting agriculture, we also apply a row-wise method of introducing fertilizers in small doses simultaneously with the sowing of grain crops by combined seeders. According to the data of the All-Union Scientific Research Institute, the annual application of 0,020 t/ha of granulated super-phosphate by the active ingredient contributed to the formation of the same gross harvest for rotation of rotation as for the main application of 0,060 t/ha of super-phosphate in pure lea field. At the same time, paying 1 kg of the active substance of phosphorus fertilizers with grain in a four-field grain-crop rotation is obtained practically the same.

With a row application of fertilizers with small doses, you can get a return in the first year on a much larger area. However, due to the small depth of fertilization in the soil in dry years, the effect is practically absent. In addition, the row method is associated with additional time and, most importantly, is tied to a period of intense spring sowing. Fertilizers in pure lea field are introduced once for rotation in a freer summer-autumn period.

Thus, under the soil protection system of agriculture in the conditions of Northern Kazakhstan, it is recommended to make the main doses of phosphorus fertilizers in the steam field at a given depth with a continuous screen, and in moist years - additionally in rows simultaneously with the sowing of cereals for the second and subsequent crops after the lea [92].

The main requirements for the technological process of introducing mineral fertilizers in rows simultaneously with the sowing are set out in the section above. Anti-erosion seeding machines such as SZS-6 and SZS -12, SKBM-12, as well as press grain seeders of type SZP-3,6 are designed with these requirements in mind and satisfy them.

When cultivating winter wheat under conditions of insufficient moisture [53,92], the efficiency of using mineral fertilizers and the technical means used for their superficial application are also established, which meet the requirements for them.

Machines for intra-soil application of the basic dose of mineral fertilizers in the soil protection system of agriculture must satisfy the following requirements:
- to apply the basic dose of granular mineral fertilizers locally intra-soil to a depth of 12 - 25 sm simultaneously with soil-free tillage;
- to ensure the introduction of mineral fertilizers with a continuous horizontal screen or ribbon with the size of an unfertilized strip between the bands not more than 15 sm;
- to ensure a dose of 0,1-1,6 t/ha in the physical weight. If higher doses are required, mineral fertilizers are recommended for subsequent soil treatments;
- deviation from the specified rate of fertilizer application can not exceed 8 - 10%;
- uniformity in the distribution of the width of the applied fertilizer band does not exceed 20 - 25%.

In addition, these machines must also meet the specific requirements for soil-tillering machines of soil conservation, which were discussed in previous chapters.

The technological process of the main intra-soil application of mineral fertilizers is carried out by machines with two structural-technological systems, each of which performs the following functions in a certain sequence:
- capacity for the continuous filling of the machine with fertilizers with a device for their dosing and feeding them to the working bodies along the fuel lines;
- working organ for the formation of the bottom of the furrow at a given depth and the uniform distribution of fertilizers on it and their sealing with soil.
Taking into account the requirements for the technological process of the intrusion of mineral fertilizers and the operating conditions of machines designed to perform this process, let us consider the main provisions for selecting the type and parameters of the above-mentioned structural and technological systems.

4.2. Containers and dispensers of machines for intra-soil application of the main dose of mineral fertilizers

In combined or fertilizer seeders for seeding (row) application of small doses of granular fertilizers, a dowel-coil seeder with non-movable coils, i.e., the constant length of its working part, is used as a dispenser. A special-shaped coil (Fig.112) with a diameter of 60 - 63 mm with two rows of pins arranged along the generator coil in staggered order. It is placed between the cheeks of the body attached to the back wall of the grain container. The length of the coil is constant – 34,5 - 35,5 mm.

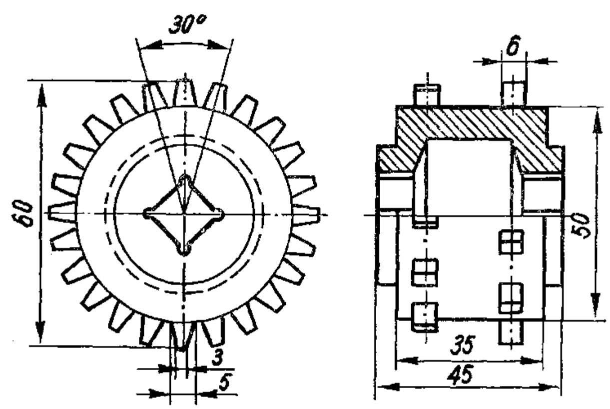


Fig. 112. Non-movable coil of the fat-seeding tractor

In the development of combined or fertilizer drills for the application of the main dose of mineral fertilizers, either tray-scraper devices or coils are used as dispensers. When choosing the parameters and operating modes of the coils, the manifold physical and mechanical properties of fertilizers are necessarily taken into account.
The working process of the bowl-scraper seeder can be divided into three successive phases: removal of fertilizers from the container to the working surface of the plate; transportation of fertilizers with a dish from the seeding hole to the point of discharge; dropping fertilizers from the dish into the sown windows [13].
The process of removal of fertilizer from the tank, assumed that there is no piling in it, is based on the law of the flow of loose material through the hole. With regard to the disk type fat seeder, this process proceeds as follows. When the plate rotates, the linear velocities of the fertilizer particles along the radial hole length are not the same and change (Fig. 113) from $v_1 = r_1\omega$ till $v_2 = R(\omega)$.

The average velocity of the fertilizer layer is defined as

$$v_{\text{cp}} = \omega(r_1 + R)/2. \tag{188}$$

Sowing hole area

$$F_0 = h(R - r_1), \tag{189}$$

where: h - is the height of the fertilizer layer (depth of the dish).
The second removal of fertilizers q by a dish from the container depends on the cross-sectional area of the sowing hole and the average linear velocity of the fertilizer layer v_{cp}

$$q = \rho_y F_0 v_{\text{cp}}, \tag{190}$$

where: ρ_y – density of fertilizers, kg/sm^3.
Then, taking (188) and (189) into account, formula (190) takes the form

$$q = \frac{\rho_y \omega h (R^2 - r_1^2)}{2}. \tag{191}$$

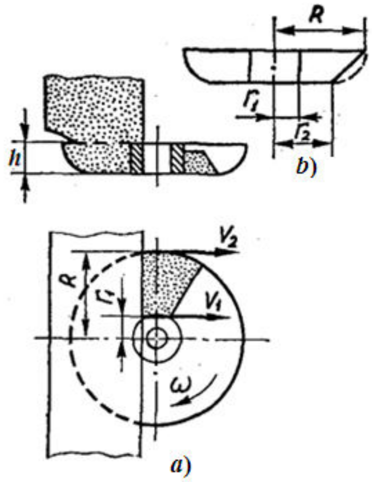


Fig. 113. The scheme of removal of fertilizers by a plate:
a) - scheme of removal;
b) - the side of the plate

The machine for the internal application of fertilizers with the width of grip $B_{\text{м}}$ at the translational speed$v_{\text{м}}$ (m/s) and at the set fertilization rate $Q_{\text{н}}$ (kg / ha) should make the following amount (kg / s) of fertilizers

$$q_c = 10^{-4} \cdot Q_{\text{н}} B_{\text{м}} v_{\text{м}}$$

and each plate

$$q_c' = 10^{-4} \cdot Q_{\text{н}} B_{\text{м}} v_{\text{м}} / z \tag{192}$$

where: z - number of plate dispensers on the machine.
Equating q_c and q_c', we obtain

$$\lambda = \frac{v_{\text{cp}}}{v_{\text{м}}} = \frac{2 \cdot 10^{-4} Q_{\text{н}} \mathrm{B}_{\text{м}}}{z \cdot \rho_y h R [1 - (\frac{r_1^2}{R^2})]}, \tag{193}$$

where: λ – some kinematic index.
This indicator connects the operational parameters υ_{cp}, $\upsilon_{\text{м}}$ and constructive R, $B_{\text{м}}$, r_1, h, and depends on the values of the velocities v_{cp}, $v_{\text{м}}$ and the depth of the plate h.However, the increase in the rate of v_{cp} is limited to the velocity of the $v_{exp.}$ of fertilizer through the seed hole. The cross-section of the sector of the plate is filled with fertilizer at $v_{\text{cp}} < v_{exp.}$. Therefore, at a given speed of the machine, the indicator λ can be changed at the expense of speed, but only up to a certain limit and by establishing the optimum depth of the dish.
The depth of the plate is found by its working volume, by which should be understood the volume of fertilizers W_0(m^3), discarded by a plate in one turn. Consequently, the plate can ensure the supply of fertilizers in the amount q_c' (kg/s) only if

$$W_0 = 60q_c'/(\rho_y \cdot n_т), \tag{194}$$

where: $n_т$ - frequency of rotation of the plate, rp/m.
Taking into account (192), we obtain

$$W_0 = 6 \cdot 10^{-3} Q_н B_м \upsilon_м/(\rho_y \cdot n_т \cdot z). \tag{195}$$

The volume of a plate, if its side is represented as an inclined plane (see Fig. 113), consists of the volumes of two concentric rings and is equal to

$$W_0 = h\pi[W_0 = h\pi[(r_2^2 - r_1^2) + (R^2 - r_2^2)]/2. \tag{196}$$

Equating the right-hand sides of (195 and 196) and solving with respect to h, we obtain

$$h = \frac{6 \cdot 10^{-3} \cdot Q_н B_м \upsilon_м}{\rho_y \cdot n_т \cdot z \cdot \frac{\pi}{2}[(r_2^2 - r_1^2) + (R^2 - r_2^2)]}. \tag{197}$$

Since the agrotechnological (initial) requirements envisage the application of fertilizers from 0,1 to 0,6 t/ha in physical weight, in the design of the machine the changes in the index λ at the expense of the speed υ_{cp} should also be provided for a wide range of changes in the rotation frequencies of the dish.
During the operation of the plate-type dispenser, the plate must rotate at a frequency such that fertilizer particles can not spontaneously descend from it.
Let us assume that the side of the plate is a rectilinear plane inclined at an angle a to the horizon (Fig. 114), and its upper edge is removed from the axis of rotation by a distance R. The gravity particle mg, the centrifugal sider mv^2/R, and the normal reaction caused by them act N and frictional force F.
The force T, which tends to push the particle up along the inclined plane of the bead, will be

$$T = (mv^2/R) \cdot cos\alpha - mg \cdot sin\alpha. \tag{198}$$

It is prevented by the frictional force F directed along the inclined plane downwards, i.e.,

$$F = tg\varphi[(mv^2/R)sin\alpha + mgcos\alpha], \tag{199}$$

where φ - is the angle of friction.

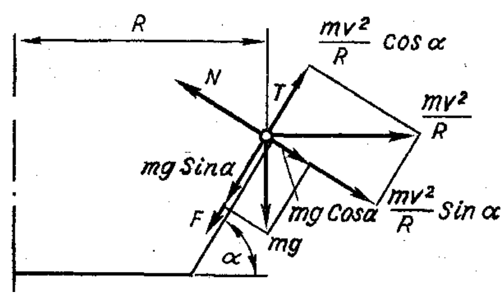


Fig. 114. Force acting on a particle of fertilizers locatedon the inclined plane of the bead side

Consequently, the particle can only rise under the condition T≥F. This inequality can be written in the form

$$K_{pr} = v^2/gR \geq \mathrm{tg}(\alpha + \varphi), \tag{200}$$

Where:K_{pr} is the limiting value of the kinematic regime.

With $K < K_{pr}$spontaneous dropping of fertilizer particles from the dish into the sown windows will not occur. Therefore, the limiting velocity v_{pr} forward at the beginning of the drop is (94)

$$v_{pr} \le \sqrt{gRtg(\alpha + \varphi)} \tag{201}$$

Because the $v_{pr} = \pi nR/30$, то

$$n \le \frac{30}{\pi}\sqrt{\frac{\text{gtg}(\alpha + \varphi)}{R}}. \tag{202}$$

Consequently, that the fertilizer particles are not spontaneously dropped from the tray into the sown windows, its rotation speed should be less or equal $\frac{30}{\pi}\sqrt{\frac{\text{gtg}(\alpha+\varphi)}{R}}$.

Fertilizers from the plates are dumped into the sowing windows with the help of a special device - ejectors, the parameters and setting of which ensure a complete dropping of the fertilizers supplied to them; uniformity of seeding; additional intensive grinding of fertilizer lumps and mixing of particles.

For a more even supply of fertilizers, a divider is installed in front of the right ejector to each ejector above the plate, dividing the total flow of fertilizers into two equal parts (Fig. 115).

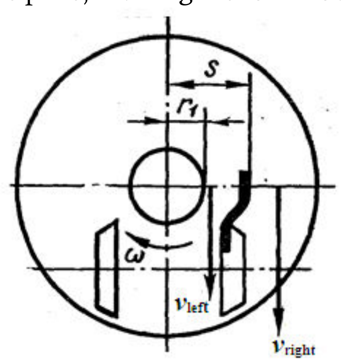


Fig. 115. Diagram of fertilizer flow dividers

Assuming that the thickness of the fertilizer layer h on the plate is the same and that the volumes of carried out fats by each ejector are equal

$$hv_{\text{left}}(S - r_1) = hv_{\text{right}}R - S), \tag{203}$$

where:$v_{\text{left}}иv_{\text{right}}$ - are the average velocities of the fertilizer layers applied to the left and right spreaders. The velocities of the movement of fertilizer layers are expressed as

$$v_{\text{left}} = \omega(S + r_1)/2\,,$$
$$v_{\text{right}} = \omega(R + S/2.) \tag{204}$$

Substituting (204) into (203) and solving with respect to S, we obtain

$$S = \sqrt{(R^2 - r_1^2)/2}\,. \tag{205}$$

Thus, the installation of a divider at a distance S from the axis of rotation of the plate ensures that each ejector releases the same amount of fertilizer.

It is possible to achieve a uniform and stable sowing of fats by a disk-scraper dispenser only if continuous forced feeding of fertilizers to the sowing windows is ensured. This feed can be achieved with active tedder, rotating a little faster than the sowing disk.

Active mowers are proposed (95) to be made from spiral-bent fingers fixed on a vertical shaft. Both the tedder and the disk are rotated from the same worm gear (gear ratio 0,078 and 0,063 respectively) so that they rotate counter-clockwise during operation. Together with the disk, the lower layer of the corks also rotates. Since the frequency of rotation of the tedder is greater than the rotational speed of the disc, the spiral fingers move the particles of this layer from the center of the container to the periphery, where the guide scrapers push them through the seed windows into the funnels of the track lines. The fingers of the mover are in different horizontal planes, so that the particles of the lower layer move. Thus, when the dispenser operates, the particles of the lower layer of fat are forcibly moved to the seeding windows and this ensures the stability of the seeding process. In this case, the rods of the wagon are curved along a logarithmic spiral.

The unobstructed movement of particles along such a spiral occurs when the angle between the radius vector and the tangent to the helix exceeds the friction angle of the feces about the material of the mover. Installation of such a tedder allows 10-15% to increase the productivity of the dispenser and increase the uniformity of seeding in a larger range of fertilizer application rates.

The principle of operation of the disk-type tractors is to move the fertilizer to the sowing windows or slits due to the rotation of the plates. The displacement is caused by the action of frictional forces arising as a result of the pressure of the mass of fertilizers on the disk of the traction apparatus. The pressure of the mass of fertilizers on the plate from its center to the belt of the container for one complete cycle is not constant.

On conventional dish-mounted fertilizer containers, fertilizer containers were installed with a small capacity of 0,025-0,035 m^3, as the pressure of the fertilizer mass on the plate increases in tanks with larger capacity, which can adversely affect not only the reliability of the machine, but also its quality parameters - seeding rate, unevenness, and so on.

When a dish-scraper dispenser mounted on a container of considerable capacity was operated, it was established that the regularity of the amount of sowing remained to the height of the fertilizer above the plate 300-350 mm (Fig. 116) [93].

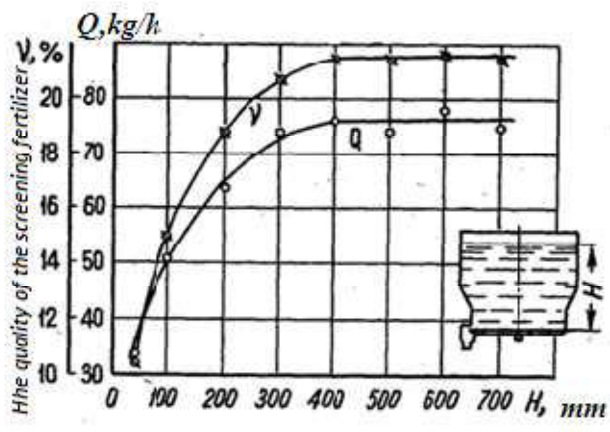


Fig. 116. Number of fertilizers sown and quality of sieving depending on the filling capacity

Further unloading leads to a decrease in the number of sown materials, i.e., the equilibrium vault is broken, and the changing hydrostatic pressure starts acting on the plate. This phenomenon is explained by the property of loose materials to form a balance of equilibrium. The code consists of many small arches, which, closing one another, form one common, assuming the gravity of the overlying material. The disappearance of the arch occurs when the proportions between the corresponding quantities that determine its presence are violated. Therefore, to prevent the disturbance of seeding stability, it is necessary to maintain the indicated level of fertilizers above the plate in the container. Such quantity of fertilizers (0,023 - 0,027 tons per each device) does not allow to use all capacity of the container and causes premature loading of the machine.

In the process of operation of the traction apparatus mounted on containers of considerable capacity, at the end of discharge in the tank, larger fractions predominate with respect to the initial granulometric composition (Fig. 117).

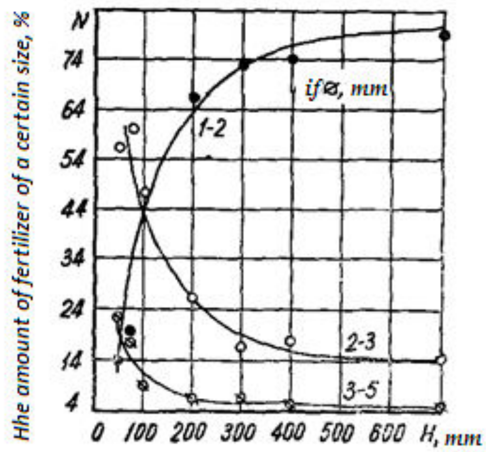


Fig. 117. Redistribution of fertilizer particles under the influence of the fertilizer during the unloading of the containerпри∅частицразмером∅ 1-2 mm; ∅ 2-3 mm; ∅ 3-5 mm

This phenomenon is explained by the presence of the active layer, i.e., by the fact that part of the fertilizers is mixed and rotated together with the plate under the influence of the plate and the tedder. Stirring of the particles helps to compact them and "float" larger ones into the upper layers. To maintain the stability of the seeding rate, hydrostatic pressure should be taken as a basis, which must be kept constant throughout the entire operation cycle, for which a special device (stabilizer) is installed in the area of the vault's equilibrium, which assumes the gravity of the vault, and the plate will act on the plate only the weight of the fertilizers located under the stabilizer.
Fig. 118 shows the change in the number of sown fertilizers depending on the filling of the container when the stabilizer is installed above the plate at a height of 100, 160, 200 mm.

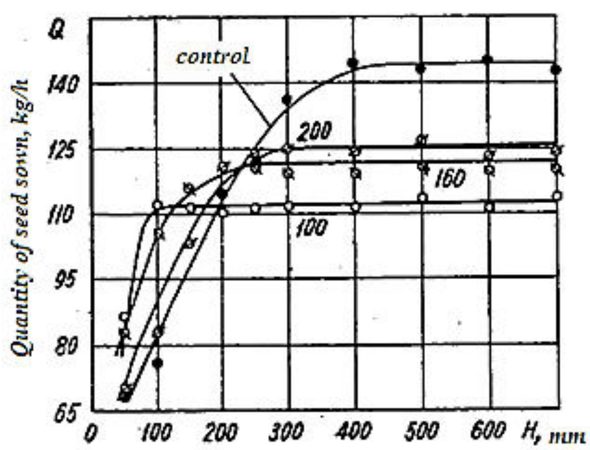


Fig. 118. Number of sown fertilizers depending on the capacity of the tank when installing the stabilizer above the plate

As a control, a full discharge of the tank without a stabilizer is taken. It can be seen from the graphs that the stability of sowing is practically not disturbed to the height of the stabilizer above the plate, i.e., practically at full discharge of the container. The installation of the stabilizer prevents the redistribution of mineral fertilizer particles by fractions to the end of the container discharge.
Thus, in order to prevent the disturbance of the stability of seeding and the redistribution of fertilizer particles when the bowl-scraper seeder operates with capacities of considerable capacity, it is necessary to install a cone stabilizer in the zone of the arch of equilibrium above the plate. The gap between the side walls and the periphery of the stabilizer should be 20 mm.
Serial plate-scraper fat seeding devices have significant drawbacks: they are material-intensive, have not quite satisfactory indicators for uniformity and stability of seeding. The construction of the device is cumbersome (two pairs of cylindrical and two pairs of bevel gears), the bracket, belt and other details are not technologically advanced, it is difficult to withstand a gap of 0,5-1,0 mm

between the sowing disk and the belt along the entire circumference. Each machine has two seed dosage adjustments, which complicates maintenance and care. Therefore, work continues the selection of the type and parameters of the new fat-sowing apparatus [96]. Table 23 shows the technical characteristics of the serial, modernized and new tractors.

Table 23- Technical characteristics of fat-sowing apparatus

Indicators	ATD-2	ATD-2A	TsVA-2C	TVP-2	ATTs-1	ATP-1
Device	Disk-scraper		Chain	Spiral Screw	Chane	Palmate
The level indicator of the fat	Float		No	Float	No	No
Treadmill reaper	Palmate		Palmate	No	No	Pendulum
Mechanism of transfer	Toothed		Chain	No	Chain	Worm
Weight, kg	26,5	34,0	27,5	15,4	15,0	30,6
Number of sowing windows	2		1	1	1	1

The highest and stable quality of seeding by the TVP-2 apparatus, in which the seed rate is changed by the gearing mechanism, i.e., by varying the rotation speed of the seeding mechanism with a constant value of the seeding windows. However, the process of sowing the fats from the tank at TVP-2 is pulsating: the diffuser of the flow consists of one or two fingers with a diameter of 1 mm and makes an oscillatory motion that is insufficient to equalize the flow.

When using the energy of free fall of mineral fertilizers to feed them from the dispenser to the embedding working bodies, the flow lines - the installation angle to the horizontal plane, the geometric dimensions, the material, etc. - have a significant effect on the seeding. This is explained by the fact that for most mineral fertilizers with a change humidity parameters of their basic physical and mechanical properties vary within a wide range [94]. The slope of the pipeline to the horizontal surface must be at least 60°.

Currently, there is a steady trend in the world for using electronic devices in mineral fertilizer machines to control the fertilizer application rate using the GPS system and JSOBUS communication facilities, to introduce precision agriculture technology with differential differentiation of optimal fertilizer doses and electronic documentation of the work performed [97].

The aforementioned drawbacks of the machines for the application of mineral fertilizers, as well as the use of their free-fall energy to supply a uniform distribution over the area, do not allow meeting the requirements of the above-mentioned world trend in the development of machines for the introduction of fat. Therefore, is carried out studies on the choice of the layout scheme of the machine for soil application of mineral fertilizers with the use of CSS and the rationale for the types and parameters of its dosing, distribution and transport systems.

The technology of differentiated application of mineral fertilizers presupposes a higher accuracy of their dosage and a multiple change in the preset dose within the field being treated. Successful implementation is possible only if uniformity and stability of dosing of fats are ensured. The termination of the expiration of mineral fertilizers due to the formation of static vaults in the reservoir above the outlet window, or the unstable flow of fertilizers due to the periodic formation of static and dynamic arches, as well as fertilizer outflows with uneven exhaustion predetermine the need for them. preparation before feeding to dosing devices [97].

Analysis of possible ways of solving the problem of preparation of fertilizers for differentiated application has revealed the expediency of using spring-helical cylindrical and conical tedderers that have a conveying capacity that allows continuously feeding fertilizers to the screening windows, while fertilizers are transported to the outlet window along a screw line with a certain angle of inclination, which leads to a constant supply and excludes its pulsation. The location of the screw tedder above the seeding windows allows using their "sifting" effect.

For the purpose of substantiating the basic parameters of the tedder, the interaction of the elementary volume of mineral fertilizers with the screw surfaces of the cylindrical and conical tedder was examined [97] and dependences were obtained for determining the path of displacement of an elementary volume of fertilizers along the coil of a tedder:
- cylindrical (S)

$$S = \frac{D}{n}t + \frac{D}{n^2}a\left(e^{-\frac{n}{a}t} - 1\right) + \frac{(ka - fn)\mathrm{sin}kt + (n + kfa)\mathrm{sin}kt}{k^3a^2 + kn^2}\ , \qquad (206)$$

where:

$$D = \frac{B\omega^2r^4}{C}\left[\frac{cos\frac{\pi}{8}}{tg\left(\frac{\pi}{4} - \varphi_2\right)} + D_1 - D_2\frac{cos\frac{\pi}{8}}{sin\left(\frac{\pi}{4} - \varphi_2\right)}\right] - f_2k_4;\ C = sin\alpha - f_1cos\alpha$$

$$D_1 = sin\frac{\pi}{8} + f_1cos\frac{\pi}{8};\ D_2 = \cos\left(\frac{\pi}{4} - \varphi_2\right) - f_1\sin\left(\frac{\pi}{4} - \varphi_2\right);$$

$$B = \pi^2\eta^2\mathrm{V}(1 + f_2)cos^2\alpha;\ k_4 = \frac{\omega^2r}{\mathrm{g}};\ k = (1 - \eta)\omega;$$

- conical (x)

$$x = \frac{q\cos\xi}{2f\omega\mathrm{sin}\alpha}\left[\frac{\cos\xi}{2f\omega}\left(l^{\frac{2f\omega t}{\cos\xi}} - 1\right) + t\right]\frac{\cos\xi(f^2\omega r + f_1q\cos\xi - 2f^2\omega q)}{\omega^2(cos^2\xi + 4f^2)}\cdot$$

$$\cdot\left[\frac{\cos\xi}{2f}\left(l^{\frac{2f\omega t}{\cos\xi}} - 1\right) + sin\omega t\right] + \frac{cos\omega t - 1}{\omega^2(cos^2\xi + 4f^2}\cdot\cdot f[2f\omega^2r + q\cos\xi(1 + 2f_1)]. \qquad (207)$$

Equations (206) and (207) allow one to determine the ways of moving an elementary volume of mineral fertilizer through the coil of a screwdriver spiral depending on its relative speed and the most important design and technological parameters, such as the angle of the cone forming its axis (ξ), the angle of the helix (α), the radius of the helical surface (r), the coefficients of the inner (f_1) and outer (f_2) friction of the fat and the angular velocity (ω). Varying each of these parameters, you can choose their optimal combination to ensure effective tedding.
To work in conditions of automatic modification of the dose of fertilizer application, it is necessary to have simple, reliable and reliable traction apparatuses. The pin coil is most suitable for these requirements, where the coil pins are made in the form of tetrahedral truncated pyramids located at the intersection of the crossing left and right multi-thread helical lines, Fig. 119.

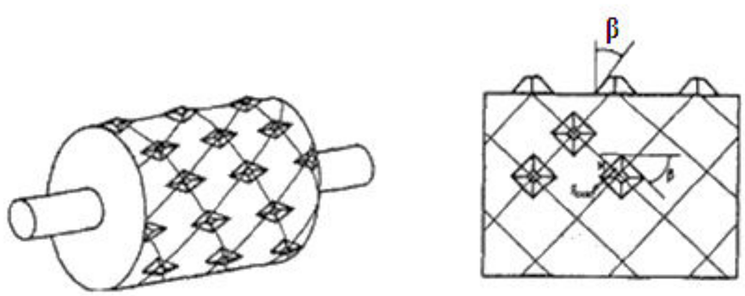


Fig. 119. The proposed reel-pin dispenser for mineral fertilizers

The execution of the pins in this form excludes the "passive zones" inherent in serial coil-pin devices, and their location at the intersection of the left and right multi-thread helical lines with an

angle β equal to the friction angle of the particle with the surface of the coil pin or the friction angle between the particles does not give fertilizers to stick.
The number of fat-fed by the proposed traction apparatus depending on the frequency of rotation of the coil and its width is determined by the formula

$$Q_2 = \pi L \gamma n_k \left\{ \left[(R_k + h)^2 - R_k^2 \right] - \frac{1}{3} K_1 h (a^2 + b^2 + ab) + \left[(R_k + h + C_{present})^2 - (R_k + \right. \right.$$
$$\left. \left. + h)^2 \right] \right\}, \qquad (208)$$

where: L - is the length of the coil; n_k —the frequency of rotation of the coil, rpm; γ - density of fertilizer, g/sm^3; R_k - is the radius of the coil base; h- is the height of the pin;K_1 - is the number of pins; α and b are the length of one side of the upper and lower bases; $C_{present}$ ...
The analysis (208) shows that the theoretical supply of the number of fats with the proposed coil spring-making machine is linearly dependent on its width and rotation speed. Search experiments have shown that the less flowability of fats, the smaller the thickness of the active layer of granules and the sifting of mineral fertilizers is possible only through the forced removal of granules by the pins of the apparatus coil.
To ensure the planned yield on a particular field, taking into account the variability of the NPK food elements, the machine dosing system must provide a differentiated application of fertilizers in the entire range of required doses from D_{min} to D_{max} and must satisfy the condition

$$B_a \cdot v_m \cdot D_{min}/n \leq q \leq B_a \cdot v_m \cdot D_{max}/n, \qquad (209)$$

where: B_a, - width of the machine, m: v_m - speed of the machine, m/s; n - number of fat sowing units installed on the machine; q - second feed of fat, kg/s.
The valuesof $D_{min}иD_{max}$, taking into account the planned yield, are predetermined by the variability of the batteries in a particular field. Under certain conditions, with a relatively high soil fertility potential, $D_{min}\rightarrow$ 0. The valuesof D_{max} are due to the presence of zero sections with a minimum content of nutrients and the number of nutrients that must be introduced to obtain the planned yield. The analysis performed, the variability of the basic elements of nutrition in the experimental field of the Kazakh Agrotechnical University named under S.Seyfulina showed that the deviation of D_{max} from the average dose of D_{aver} can reach 80 - 90%. These data were used to justify the requirements for the lower and upper limits of the second delivery of the dosing system (Fig. 120).
For example, to ensure differentiated doses in the range 0- 400 kg / ha with a working width of the unit of 2,0 m and the speed of the fertilizer machine 6-12. km/h, the specific productivity of the dosing system should vary between 0-300 g/s.

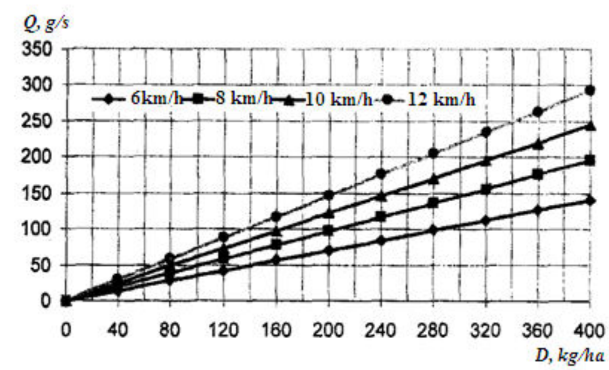


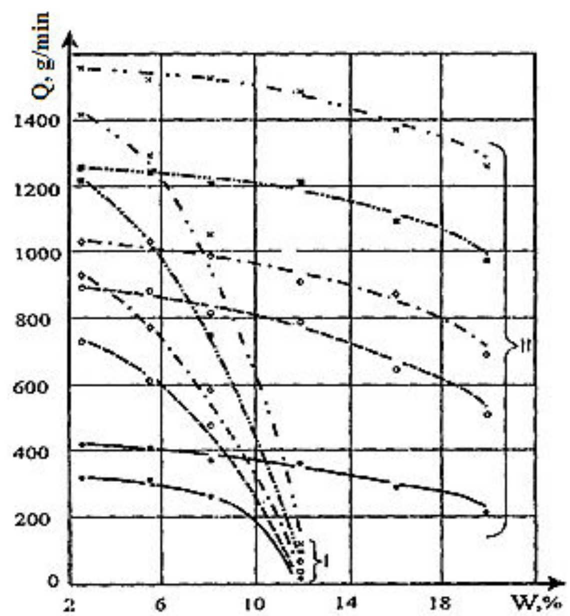
Fig. 120. Dependences of the second supply of the number of carotene system carcasses from the dose of application D and the speed $\upsilon_м$ of the machine

To substantiate the optimal parameters of the pin coil, a central [97] composite rotatable second-order scheduling was used. When sowing granular super-phosphate, the influence of the angle of inclination of the lateral surfaces of the pin on the vertical and horizontal surfaces (β), the pitch between the pins (S, mm) and the height of the pin (h, mm) on the unevenness of seeding between the apparatuses ($У_1$) and the seeding instability ($У_1$)) and the following regression equations are obtained:

$$У_1=3{,}549+0{,}241x_1+0{,}102x_2-1{,}640x_3+0{,}96x_1x_2-0{,}687x_1x_3-$$
$$-0{,}275x_2x_3+0{,}379x_1^2+0{,}651x_2^2+2{,}271x_3^2; \quad (210)$$
$$У_2=4{,}12+0{,}322x_1+0{,}158x_2-0{,}336x_3+0{,}462x_1x_2+0{,}275x_1x_3-$$
$$-0{,}671x_2x_3+0{,}789x_1^2+0{,}594x_2^2+0{,}487x_3^2; \quad (211)$$

the analysis of which allows us to justify the optimal values β^0=40-45^0, S=12-13 mm and h = 7,5-7,7 mm.

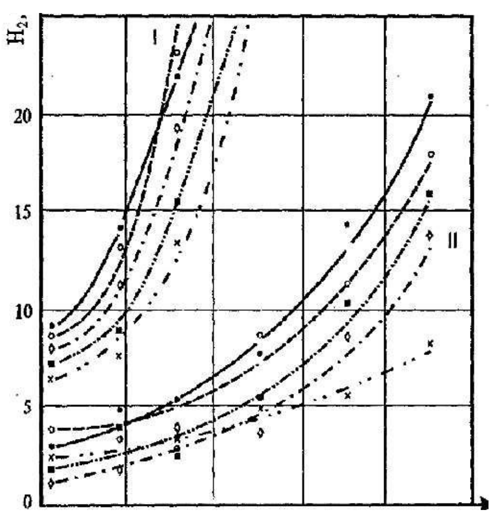
The experimental verification of the efficiency of the proposed tedder and coil-pin dispenser showed the following [97]. The serial pin coil with humidity (W%) of fertilizer 8,2% is practically no longer sowing, and the use of tedder allows to increase the moisture limit of fats for serial coils up to 12%. With a further increase in the humidity of fertilizers, the technological process of dosing stops, because the coils have turned into "cylindrical rollers". The proposed coil when sowing fertilizers of different humidity works more steadily, as the moisture content of the material increases, the number of sown crops (Q g/min) gradually decreases (Fig. 121). The number of sown crops with the use of tedders at the same humidity is more 1,5-1,7 times than when working without a tedder. The use of agitators also increases the indices of instability of the sown crops (Fig. 122).



I - Serial pin coil
II - Proposed pin coil

—•— $n_k=1{,}05^{s-1}$
--o--o-- $n_k=2{,}09^{s-1}$
–· –◊ –· $n_k=3{,}14^{s-1}$
–··–▪–··– $n_k=4{,}19^{s-1}$
–··–x–··– $n_k=5{,}23^{s-1}$

Fig. 121. Dependences of the amount of powdered superphosphate being sprayed on moisture and the frequency of rotation of the coil



I - Serial pin coil
II - Proposed pin coil

—•— $n_k=1,05^{s-1}$
- -o- -- $n_k= 2,09^{s-1}$
—· –◊ –· $n_k = 3,14^{s-1}$
—··—▪—··— $n_k= 4,19^{s-1}$
—··—x—··— $n_k= 5,23^{s-1}$

Fig. 122. Dependences of the instability of seeding powdered super-phosphate on humidity and the frequency of rotation of the coil

4.3. Working bodies for the intrusion of mineralfertilizers at a given Depth and their distribution at the bottom of the furrow

On sowing machines of soil conservation, which provide for the possibility of introducing mineral fertilizers in small doses simultaneously with sowing, the intra-soil application of fodder is carried out by the same working organs as the sowing of cereals.
With the soil protection system of agriculture, the main dose of mineral fertilizers should be applied, as indicated above, simultaneously with the planing of the soil - this determines the use of flat cutting feet as working tools. The values of their geometric parameters, found on the basis of working conditions and specific requirements for soil-cultivating machines for soil conservation, have already been considered by us. Therefore, the main task is to apply a flow of mineral fertilizers to a limited sublacial space after the dispenser and, with allowable agricultural demands, distribute them unevenly along the width of the seam at the bottom of the furrow.
Technical solutions providing the distribution of fertilizers in a limited in height sub-space, a lot is known. The simplest, reliable and universal working body is a passive cone-shaped or wedge-shaped distributor (Fig. 123a).
Fertilizers flowing by gravity through the pipeline 1 are reflected from the working surfaces of the distributor 2 and are dispersed in the sub-space 3. The main disadvantages of such a switchgear are the clogging of a massive distributor under the fuel line with mineral fertilizers and the formation of a "hard" on the working surfaces of the distributor. At the height of the sub-pile space 30 - 40 mm, the width of the sieving band does not exceed 150 - 200 mm. A wider sieving band is formed by working elements with a vibrating distributor (in Fig. 123b a vibratory line is not shown). But the vibration drive very complicates the design and reduces its reliability.

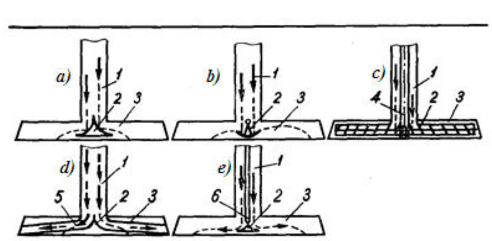


Fig. 123. Devices for the distribution of mineral fertilizers in the soil with the width of the claw: a) - with a passive distributor; b) - with a vibrating distributor;c) - with a mechanical distributor; d) - with pneumatic conveying and sieving; e) - with pneumomassive

Working organs with a mechanical (screw) distributor (Fig. 123c) reduce the uneven distribution of fertilizers to 10-12% [98,99]. However, in the case of a change in the application rate, the physico-mechanical properties of mineral fertilizers that vary widely, or the modes of operation of the screw, the unevenness of the distribution increases sharply. To the drawbacks of mechanical distributors is the drive mechanism, which complicates the design of the machine and creates a lot of difficulties in maintenance and operation.

Technical solutions that ensure the supply of mineral fertilizers and their sifting with air flow should be considered more successful and promising. Let's consider two of them: the first one with pneumatic conveying and sieving (Fig.123d), the second one with mechanical feeding of the fat into the cavity of the rack and pneumomassing them in the sub-paw space (Fig. 123e).

In the first variant, the fat conduit I is divided into two horizontal channels 5, the shape of which can be box-shaped (most often) or as a trough with an open rear wall. Fertilizers move along the channels together with air and pass through the open side or into the gap in the sub-space. The shape and cross-sectional area of the channels greatly influence the uniformity of the distribution of fertilizers along the width of the claw, which narrows the scope of application of such a technical device, the parameters of which are usually calculated for a certain mode of operation, the norm and the physico-mechanical properties of fertilizers applied to the soil. In addition, mineral fertilizers can adhere to the surface of the ducts, especially at the beginning of the horizontal channels. Another drawback of such devices is that a change in the air resistance at the outlet under one wing of the paw increases its flow under the other wing, which increases the unevenness of the distribution.

The second variant lacks the drawbacks inherent in the former, since the air stream is brought to the top of the distributor in the sub-space. Fertilizers that flow through separate channels to the working surfaces of the distributor 2 are entrained and ejected by air jets under the wings of the plane-cutting paws. Placement of the pneumatic nozzle at the top of the distributor allows to increase the flight range of fertilizer particles in the sub-space, since the energy of the air jet is used most fully.

However, if the pneumatic nozzle is positioned directly at the top of the distributor, the necessary uniformity of dispersion will be achieved only if the flow of mineral fertilizers in the cross-section of the track line is uniform. With the simultaneous supply of air and fertilizers to the track, the flux density in its cross section becomes more uniform. In this case, the qualitative performance of the dispenser on the uniformity of dispersion is improved.

The range of flight of the fertilizer particle after leaving the edge of the distributor, which has some form of a reflecting surface, is determined, as is known, by expression as

$$l_1 = x_0 + \upsilon_0\sqrt{2h/\mathrm{g}}\,, \tag{212}$$

where: x_0 - is the distance of the edge of the reflecting surface of the distributor from the axis of the track line; υ_0- is the initial velocity of the fertilizer particle movement after leaving the surface of the reflector; h - is the height of the sub-space; g - acceleration of gravity.

It can be seen from formula (212) that the range of flight of fertilizer particles can be increased by removing the edge of the reflecting surface of the distributor from the axis of the track, increasing the height of the sub-foil space, or by increasing the initial velocity.

The large removal of the edge of the reflector surface of the distributor from the axis of the conduit is fraught with dangerous consequences, since it can turn out that most of the energy of free fall of a particle or air flow will be expended to overcome the frictional forces arising from the movement of the fat along the surface of the switchgear. It is not possible to increase the height of the sub-space, since in this casethere may be a process of soil unloading, and, consequently, the technological process of tillage is disrupted.

When using free fall energy to feed the fat to the distributors of wide-grasping paws, the particle velocity in a strictly vertical flue pipe is insignificant and does not exceed 3,0-3,5 $\mathrm{m/s}$ [93]. Such a speed is not able to provide a high initial velocity of free flight of particles after impacting them on the reflecting surface of the distributor.

Therefore, to increase the speed of flight of fertilizer particles in the track, an air flow is used (Fig. 124).

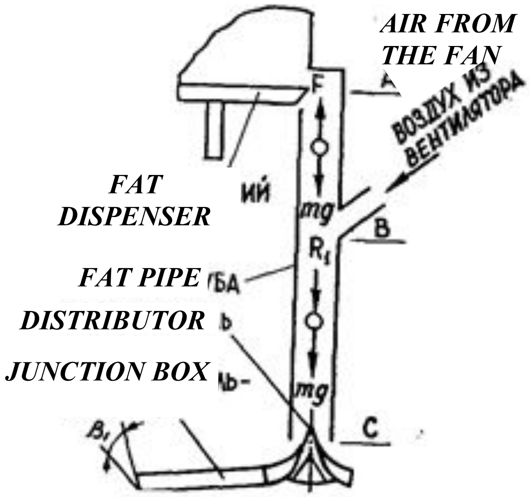


Fig. 124. The scheme of the action of forces on a particle of fertilizers in the track of the workingmember of the flattening fertilizer.

From the fertilizer feeder, the fertilizer particles enter the track and, in the section,AB, they acquire a certain speed by the force of gravity when they meet the air flow. In the dispersed part $BC$of the aircraft, the velocity of the fertilizer particles increases with the action of the air flow (Fig. 125).

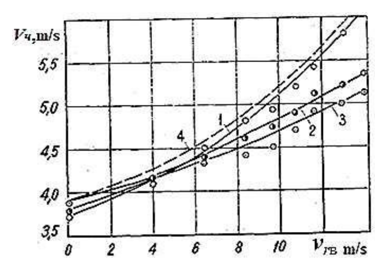


Fig. 125. The velocity of super-phosphate particles at the end acceleration section of the flywayBC, depending on the speed air flow:
1 - granules measuring 1 - 2 mm; 2 - 2 - 3 mm; 3 - 2 - 5 mm:
4 - calculated for granules 1 - 2 mm

At the time of meeting with the air flow, large particles (2 - 5 mm) have a high speed. Then the particles of size 1 - 2mm catch up and overtake the larger ones.
To determine the velocity of particles at the end of the accelerating section, $\upsilon_{чс}$ with (at point BC), the following formula is proposed [93].

$$\upsilon_{rc} = \upsilon_{\text{в}} - \frac{\upsilon_{\text{в}} - \left\{\upsilon_{r\text{в}} + \left(\frac{2Pg}{K_x\gamma S}\right)^{\frac{1}{2}} \cdot \mathrm{tg}\left[\left(\frac{K_x\gamma gS}{2P}\right)^{\frac{1}{2}} \cdot t\right]\right\}}{1 + \left(\frac{K_x\gamma S}{2Pg}\right)^{\frac{1}{2}} \cdot (\upsilon_{\text{в}} - \upsilon_{r\text{в}}) \cdot \mathrm{tg}\left[\left(\frac{K_x\gamma gS}{2P}\right)^{\frac{1}{2}} \cdot t\right]}\ , \tag{213}$$

where: $\upsilon_{в}$ - speed of air flow; $\upsilon_{r\text{в}}$- the velocity of the fertilizer particle before it meets the airflow; P is the particle gravity; g - acceleration of gravity; γ - density of air; S is the area of the particle's projection onto a plane perpendicular to the direction of flow; t is the time the particle is in the air stream; K_x- coefficient of drag

$$K_x = \frac{Pg}{\gamma S\upsilon_{cr}^2}\ , \tag{214}$$

where: v_{cr} - critical speed (speed of wobbling).
The ratio g/v_{cr} characterizes the sailness of the fertilizer particle. The values of the coefficients of sail and critical velocities of particles of the main types of fertilizers, depending on their dimensions, are given in Table 24 [100].
The calculated values of the velocities of the fertilizer particles (super-phosphate) at the end of the accelerating section of the track line, found from the formula (213), agree satisfactorily with the experimental data (see Fig. 125, curve 4).

Table 24. Aerodynamic characteristics of some types of mineral fertilizers depending on the size of their particles

Indicators	Particle sizes, mm						
	super-phosphate					ammonium nitrate	potassium salt
	1	2	3	4	5	1,47	0,93
Critical speed, m/s	3,7	6,3	8,8	9,6	11,3	6,7	4,9
Coefficient of sail	0,73	0,24	0,12	0,10	0,07	0,22	0,41

Increase the initial velocity of the free flight of fertilizer particles after their descent from the distributor can be due to a rational choice of the shape of its reflecting surface. The particle has the greatest velocity when leaving the generator of the parabola (Fig. 126) [101].
In order to create a distribution device that ensures a stable uniform distribution of fertilizers, a universal method for constructing the reflecting surface of the distribution box based on the complex motion of the curvilinear generator has been developed. The essence of the method is as follows (Fig. 127).

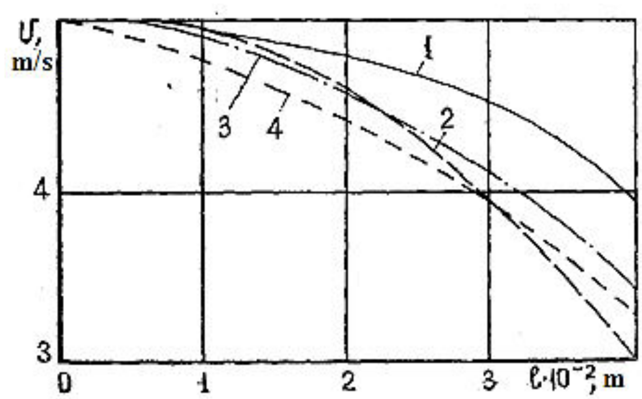


Fig. 126. The velocity of a material particle moving along generators in the form of a parabola (1), a hyperbola (2), an exponential curve (3), and an arc of a circle (4)

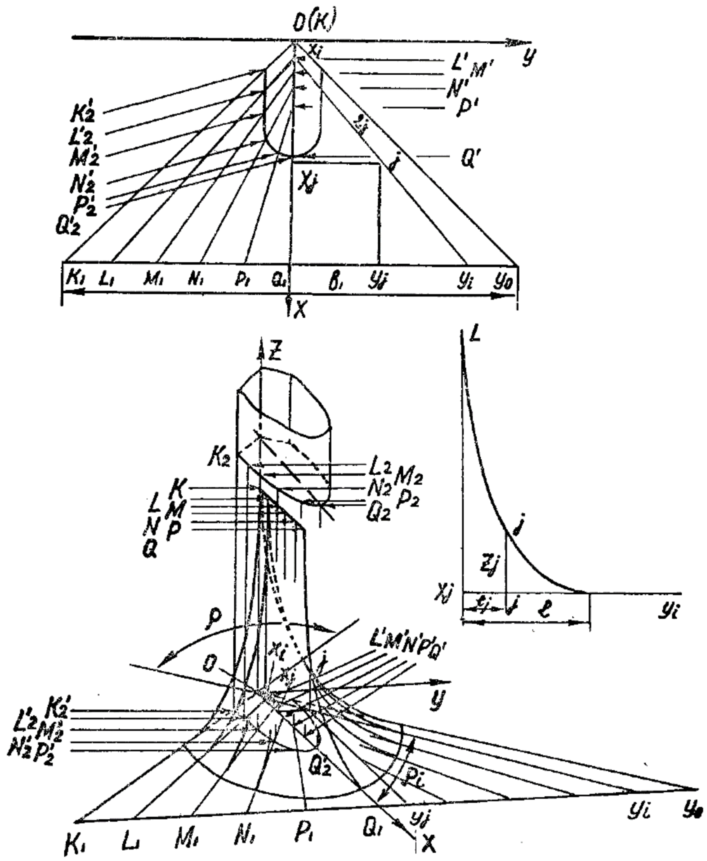


Fig. 127. The scheme for constructing a reflecting surface
of fat distribution box

On the vertical axis, a segment of OK is plotted, equal to the height of the generatrix of the surface. From the point K a straight line parallel to the OX axis is drawn, and the segment KQ, the edge of the fat distribution box, is deposited on it. The segment OQ_1 is the projection of the edge onto the axis OX. The leader of the corks (fat-drive) is placed above the distributor in such a way that the longitudinal axis of its projection on the horizontal plane coincides with the axis OX.
The segment OQ' and the segment K_1Q_1 equal to half the sieving width are divided into the same number of parts, and the division points are connected by the lines OK_1; $L'L_1$; $M'M$, etc. Let us assume that these lines are projections onto the horizontal plane of the trajectory of the motion of fertilizer particles. It is assumed that in the fat particle path the move vertically. Then, with the angle of inclination of the tangent to the curvilinear generator to the vertical γ<10°, the trajectory of the motion of each particle after reflection will lie in a vertical plane passing through the normal to the reflecting surface at the point of its collision with the particle.
The actual trajectories will be close to the expected ones if each projection of the generator to the horizontal plane coincides with the projection of the proposed trajectory. In this case, the particles passing through the section of the conduit bounded by the quadrilateral$OL'L_2'K_2'$ will be sown through the sectionK_1L_1, the particles passing through the cross section LM_1 will be sown through the section$L'M'M_2'L_2'$, etc. Therefore, if the areas of the elementary quadrilaterals and the uniformity of the flow are equal in the fertilizer line the fertilizers will be evenly distributed along the width of the sifting.
The presented method of constructing a reflecting surface is universal and can be used in the design of various fat spreading devices. Depending on the specific conditions of the problem, generators of various shapes can be used to construct reflective surfaces.
For convenience of practical application of the proposed method, it is necessary to compose the equation of the reflecting surface of the distributor. In the design scheme, some design parameters

of the switchgear are taken into account: the sieving angle ρ, the sieving width b_1 the length of the edge of the tug distributor l_1. Since any point of the projected onto the horizontal plane divides the projection of the edge into the same plane and half the width of the sieving by proportional parts, we have

$$y_i = (b_1/2l_1) \cdot (l_1 - x_1), \tag{215}$$

where:$x_1 \leq l_1 -$ is the coordinate of the point located on the edge of the fat-distributor.
From the triangle $x_1 y_i Q_1$we have

$$tg\rho_i = y_i/(OQ_1 - x_i. \tag{216}$$

Taking into account (215), the formula (216) takes the form

$$\mathrm{tg}\rho_i = \frac{(b_1/2l_1)(l_1 - x_i)}{OQ_1 - x_i}, \qquad 0 < \rho < 180^0. \tag{217}$$

Knowing the angle ρ_i, the relationship between the coordinates y_j and x_j of the points of the reflecting surface can be written in the form

$$y_j = (x_j - x_i) \cdot \mathrm{tg}\rho_{i\,.} \tag{218}$$

Substituting (217) into (218) and taking into account that OQ'=$b_1/2\mathrm{tg}\frac{\rho}{2}$
we get:

$$y_j = \frac{b_1(x_j - x_i) \cdot (l_1 - x_i)}{2l_1\left(\frac{b_1}{2\mathrm{tg}\frac{\rho}{2}} - x_i\right)}. \tag{219}$$

Ifl is the length of the projection of the generator onto the horizontal plane, then
$x_i \leq x_j \leq l + l_1$; $l_j^2 = y_j^2 + (x_j - x_i)^2$; $l_j \leq 1$.
Then

$$l_j = (x_j - x_i)\left[1 + \frac{b_1^2(l_1 - x_i)^2}{4l_1^2\left(\frac{b_1}{2\mathrm{tg}\frac{\beta}{2}} - x_i\right)}\right]^{\frac{1}{2}}. \tag{220}$$

Using the formula (220), we can obtain an expression for determining the coordinates z of the points of the reflecting surfaces with generators of different shapes. For a reflecting surface formed by the motion of a parabola,
we have:

$$z_j = a\left\{l - (x_j - x_i)\left[1 + \frac{b_1^2(l_1 - x_i)^2}{4l_1^2\left(\frac{b_1}{2\mathrm{tg}\frac{\beta}{2}} - x_i\right)^2}\right]^{\frac{1}{2}}\right\}^2. \tag{221}$$

According to this technique, reflecting surfaces formed by the motion of the parabola $y = ax^2$ with different values of its coefficient, and various valuesof the design parameters of the distributor-the length of the base, the height, the angle of sifting, and the length of its rib [101] were constructed for the distribution device to wide-cut planar paws. The necessary stability and uneven distribution of fertilizers is ensured with the following values of the parameters of such a distributor: the parabola coefficient a = 0,9 -1,1; base length$l$ = 30 - 40 mm; height Z = 90 - 150 mm; sieving angle ρ = 96 - 160°; rib length l_1 = 10-25 mm.

Thus, it is possible to significantly increase the initial speed of flight of fertilizer particles after leaving the surface of the reflector both by using the air flow in the track line and by selecting and constructing a rational reflective surface of the distributor. But, even a significant increase in the initial velocity of the fertilizer particles after leaving the surface of the reflector will not allow them to overcome the distance from the middle to the edge of the paw because of the low height of the sub-lobe space only by dropping the pellets onto the bottom of the furrow. Therefore, it becomes necessary to install a box (plane) at an angle of $\rho_1 \cong 10°$ to the horizon in the sub-lacial space (perpendicular to the movement of the aggregate), the material of which would have a lower coefficient of friction than the soil.

The distribution device developed with the use of the proposed method and the installation in the sub-crustal space of the box allow the fertilizer to be sifted along the width of the gripper of the flat-paw with a non-uniformity of 13 - 15% over a wide range of application rates and pressure head (Fig. 128). With a head pressure of P_g= 60 N/m^2 and application rates of up to 320 kg/ha, a stable distribution of the main types of mineral fertilizers and their mixtures is achieved.

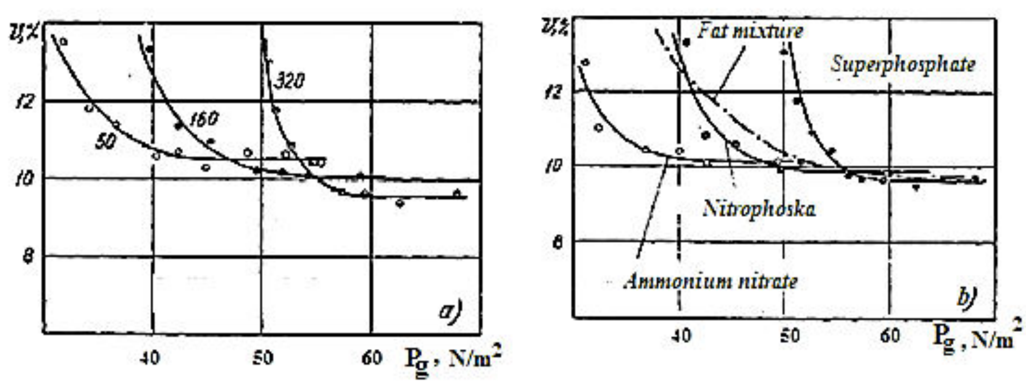


Fig. 128. The uniformity of sieving tocharacterdata with the proposed parameters depending on the dynamic pressure of the air flow and application rates

a) in the sifting of super-phosphate when application rates 50,160 and 320 kg/ha;
b) for the sifting of various fertilizers and mixed fertilizers with a rate of application of 320 kg/ha

4.4 Selection of the type and parameters of the working element for the subsoilapplication of herbicides to combat the bitter creeping (pink)

In modern agricultural production, chemical plant protection products play an important role in increasing production and improving the quality of crop production. However, they have an extremely negative impact on the environment. As a result, the natural composition of the atmosphere, water, soil, etc., is gradually but inevitably changing.

Machines for plant protection are primarily hook-on, hinged and self-propelled rod sprayers, as well as special equipment for them. The firms Berthoud, Norac (Canada), Tecnoma, RDS France, Hardi-Evrard S.A. (France), MB M. Beyn, Delvano (Belgium), John Deere, Amazonen-Werke SA (Germany), and others produce the full range of spraying equipment that allows timely, high-quality and low-impact on the surrounding environment to protect agricultural plants from pests and diseases [5,6].

Leading European manufacturers of spraying equipment in the development and improvement of modern structures are guided by the legislative base, based on the draft of the model rules on the further application of plant protection products presented by the EU Commission. One of the main

requirements considered is the protection of the environment and the health of consumers. In this regard, the main trends of their further development are directed [6] to:
- minimization of their negative impact on the environment in accordance with international quality standards; reducing the costs of herbicides, insecticides, fungicides and pesticides per unit of treated area without reducing the efficiency of their application by automating the control and management of technological processes and the use of information technology; Increase in productivity due to increased working width and their load capacity; creation of highly unified and automated separate assembly units and units that allow them to be used on machines of other firms;
- equipment for working with the satellite navigation system (GPS), systems for creating an air curtain to prevent the drift of pesticides by wind.

Well-known technologies and technical means of introducing means of chemical protection of plants by spraying them are also applicable for soil-protecting agriculture. However, in soil-protecting agriculture, the most acute problem is the introduction of so-called soil chemical protection of plants.

Currently, soil herbicides are introduced in two ways:
- herbicides are sprayed by sprayers on the field surface and their subsequent incorporation into the soil by soil cultivating machines for small loosening of the soil (by harrow lappers, etc.);
- intra-soil application of herbicides simultaneously with soil treatment or sowing.

The first method of introducing soil herbicides in the system of soil conservation agriculture is practically unacceptable because of the specific requirements specified in Section 2 for tillage machines. The second method is most promising, as it is in principle acceptable not only to conventional tillers and implements, but also used in soil-protecting agriculture.

So, for the introduction of herbicides simultaneously with the cultivation of soil, the production of attachments has been created and mastered, for example, OKZh-5,6 (Agro-Tech LLC, Rostov Region), which is installed on various tillage machines, and allows the supply of a specified volume (100 - 120 liters) herbicide solution per hectare.

Taking into account the specific nature of soil conservation agriculture, Research and development work was carried out to develop machines for intra-soil application of herbicides on the basis of a rod cultivator and a flat-cultivator [102,103]. The extensive economic tests of these machines in the conditions of Northern Kazakhstan have confirmed the effectiveness of their application. However, during their operation, frequent sticking of the outlet holes on the rod is noted because of the constant contact of the outgoing solution with the soil, and the unevenness of the distribution along the width of the claw capture during various operating modes of the planar cut increases.

Research studies on the choice of the type and substantiation of the parameters of working organs for the intrusion of herbicides continue, especially in connection with the tightening of environmental requirements for technologies and technical means for the use of chemical plant protection products [104,105,106,107].

As is known, the quarantine weed bitter creeping causes a tremendous damage to agricultural production, which is spread in the Russian Federation by 400 thousand hectares, Ukraine - 500 - 600 thousand hectares, Armenia more than 15 thousand hectares [104,106,108,109].

According to the data of the RSE "Phyto-sanitary", the bitter creeper is distributed in all 14 regions of Kazakhstan and its total area from 650 thousand hectares in 1965-1970 increased to 2,6 million hectares by 2005 [110,111]. As a result of such clogging of crops and agricultural lands, creeping annual losses of products amount to more than 3 billion Tenge.

In the struggle against bitter creepers, such agrotechnological methods as plantation plowing, numerous mechanical processing of steam, semi-steam, meliorative fields (from 8 to 15 treatments) are used. However, due to their high cost and labor intensity, these methods of struggle have not found wide circulation.

According to the data available in the literature, the best results on combating smartweed give a combination of applying herbicides with methods of deep-till soil cultivation, since it has been established that segments of bitter roots less than 5 cm practically do not survive [112,113].

When developing an integrated system for combating bitter cropland on agricultural crops in conditions of rain-fed farming in the southeast of the Republic of Kazakhstan, the Kazakh Institute of Plant Protection and Quarantine has substantiated the expediency of simultaneously loosening

the soil with the working elements of a flat-top cultivator to a depth of 10-12, 14-16 and 18-20 sm [113]. At the same time, the loss of bitter was 60,9%, which undoubtedly indicates a high efficiency of this technological method, and with the simultaneous intra-soil application of a uniform herbicide solution screen to a depth of 14 - 16 cm, the degree of destruction of bitter is sharply increased. In connection with the foregoing, Research and development work was carried out to create a three-level cultivator-planar with simultaneous intrusion of herbicides into the second layer to a depth of 12-14 sm. [114].
In principle, there are no special difficulties in the development of such a tool because:
- the developers have a great experience of Research and Development work on the creation of cultivators-flat cutters (see Section 2);
- for application of herbicides simultaneously with soil cultivation, there is a mounted equipment OKZh-5,6 (LLC Agro-Tech, Rostov region, Taganrog), which can be mounted on various types of soil-cultivating machines, including flat-top cultivators, and allows to ensure the supply of a given volume of herbicide solution per hectare (approximately 100 - 120 liters).
However, as mentioned above, the system developed for a soil cultivator-flatter for intra-soil application of herbicides [103] did not ensure a stable uniformity in the distribution of the herbicide solution over the entire width of the claw capture when the speed of the aggregate, the depth of loosening, the physical and mechanical state of the treated soil layer, etc, In this connection, it became necessary to consider the process of interaction of the planar cutting foot with the soil and the spray flares of the herbicidal solution from the paw of the soil layer coming off the knife.
A straight wedge (990 mm) with an angle of setting the knife to the bottom of the furrow$\alpha_0 = 25^0$ and an angle $\gamma = 35^0$ to the direction of movement and height of the formation OZ = h = 35 mm is laid in the basis of the wide-cut (990 mm) flat-paw of cultivators-flat cutters (Fig. 129).

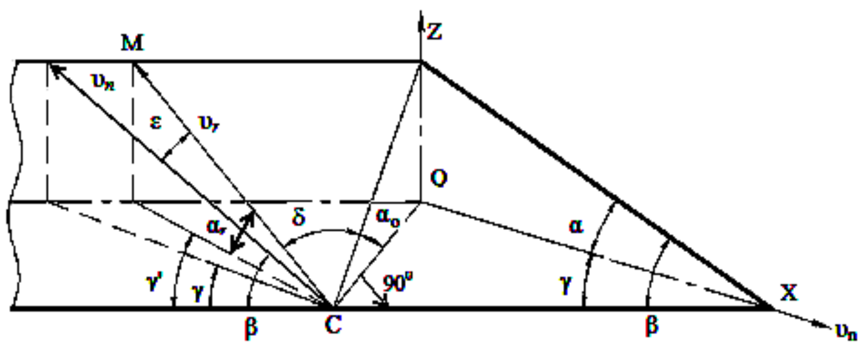


Fig. 129. To the analysis of the work of the flat-paw

In this connection, the layer of soil moving along the surface of the oblique wedge will descend with a speed v_r at a certain angle -ε to the direction of its translational velocity v_n. In this case, the height and range of the soil formation are determined by the parameters of the flight trajectory as a heavy material point M thrown at an angle α_r to the horizon at a certain speed v_r. Between the lower surface of the soil layer coming down from the paw and the bottom of the furrow, a space is formed that is insignificant in height, length and width. Therefore, there are certain difficulties in providing an allowable ± (5,0 ÷ 10,0)% non-uniformity in the distribution of the screen of the herbicidal solution over the area under the wide-cut planar paw. This circumstance predetermines the advisability of using slotted sprayers of a herbicide solution (for example, type ST), forming a thin cone spray nozzle with an angle at the apex of 80° - 120°. This will allow a smaller number of nozzles to be installed under the wide-grip paw, which simplifies the design and reduces the chances of plugging out the system's nozzles during operation.
The diameter of the aperture (caliber) of the slit nozzle depends on the given volume (100 - 120 L) of the volume of the herbicidal solution per hectare. With the width of the capture of a 3-level flat-cultivator-plane of 2,77 m, the speed of its movement of 8 km/h and the rate of application of a solution of 120 L/ha, the volume of supply of a solution under one paw will be 1,3 L/min. When installing, for example, seven slit nozzles on the width of the claw, the volume of solution delivery

with one atomizer will be equal to 0,19 L/min, which is ensured, for example, by a commercially available spray gun ST 110-01.
Sprays should be located along the contour of the back cut of the paw-leg paw with a thickness of 16 mm at a height of approximately 27 mm from the surface of the bottom of the furrow and below by 8 mm of the horizontal surface of the tampon (Figure 130).

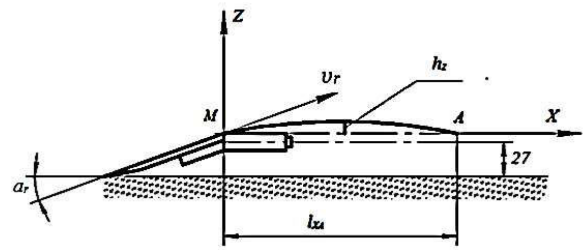


Fig. 130 - To determine the parameters of the trajectory of the flight of the soil coming down from the paw

A planar conical flare formed by a slotted nozzle, the plane of which must be parallel to the bottom of the furrow, will meet with the layer of soil emerging from the paw at point A (see Fig. 130). The soil layer saturated with herbicidal solution will gravitate down to the bottom of the furrow under gravity and should form a herbicidal screen, which will destroy directly or interfere with the development of the groove of the root system of the creeper located below the bottom. The contour and coordinates of the line of intersection of the spray of the spray and the lower plane of the layer of the soil coming off the paw and will determine the required number of nozzles under the paw and the angle of deflection of the axis of the flare relative to the direction of its translational velocity. Determining the rational values of these parameters and will ensure the necessary uniformity of distribution of the solution over the area under the paw.
The trajectory of flight of a layer of soil coming off the paw can be considered as the movement of a heavy material point M, cast at an angle to the horizon α_r at a certain speed v_r. The trajectory of the motion of a heavy material point M, thrown at an angle α_r to the horizon in the coordinate system XMZ is determined by the equations [115]

$$\left.\begin{aligned} l_{XA} &= v_r \cdot t \cdot cos\alpha_r \\ h_z &= v_r \cdot t \cdot sin\alpha_r - \frac{1}{2} g t^2 \end{aligned}\right\} \tag{222}$$

where: l_{XA} - the range of flight of the soil after descending from point M,
g - is the acceleration of the free fall of the body,
h_z - height of the flight trajectory of the soil at the point M.
Let us find the equation of this trajectory in a nonparametric form. To this end, from the first equation of system (222) we define t, substitute into the second equation and after transformation we obtain

$$h_z = l_{XA} \cdot tg\alpha_r - \frac{q l_{XA}^2}{2v_r^2 \cdot cos^2 \cdot \alpha_r} \tag{223}$$

To determine the range of the flight of the soil after descending from the point M, we put in (223), that$h_z = 0$. In this case $l_{XA} \cdot \left(tg\alpha_r - \frac{q l_{XA}}{2v_r^2 \cdot cos^2 \cdot \alpha_r}\right)$, i.e. we obtain two points of intersection of the trajectory with the axis MX, i.e. origin$l_{XM} = 0$, and

$$l_{XA} = \frac{v_r^2 \cdot sin2\alpha_r}{\text{g}} \tag{224}$$

To find the maximum lifting height of the trajectory h_z, we equate the derivative

$\frac{dh_z}{dl_{XA}}$ with zero in (223), and, transforming, we find

$$h_z = \frac{v_r^2}{2g} \cdot sin^2 \alpha_r \tag{225}$$

In the formulas (224) and (225), the velocity of the soil, which is cast at an angle to the horizon, for the flat-topped paw, is equal to its relative velocity along the surface of the paw knife, i.e. v_r.
According to the experimental and calculated data (see Section 2.2.1) for a flat-topped foot with parameters α_0= 25^0, γ = 35^0, h = 35 mm and its translational speed of 2,17 m/s (≈ 7,8 km/h), the relative speed the displacement of soil along the surface of the blade knife is υ_r about 1,7 m/s.
The angle α_r formed by the vector of relative velocity of soil movement along the surface of the knife with a horizontal plane near the flat-topped paw (see Fig. 129) is the actual cutting angle and is found by formula

$$tg\ \alpha_r = tg\ \alpha_0 \cdot sin\gamma' \tag{226}$$

where: γ'- the angle between the projection of velocity υ_r on the horizontal plane and the blade of the knife of the oblique wedge and is equal to

$$\gamma' = arc\ tg(ctg\ \delta \cdot \cos \alpha_0) \tag{227}$$

The angle δ characterizing the deviation of the velocity vector of the soil movement along the surface of the knife from the perpendicular to its blade (see Fig. 128), with the above parameters of the paw and the speed of its translational motion, will be approximately 45^0. Then, according to (227) and (226), the angle γ'≈ 41^0, and the actual cutting angle α_r will be ≈ 17^0.
The angle ε characterizing the deviation of the vector$\bar{\upsilon}_r$ from $\bar{\upsilon}_n$ on the surface of the knife is determined from relation

$$\varepsilon = 90^0 - (\beta + \delta), \tag{228}$$

where: β - the angle of the cut of the knife of the paw and is determined by the known la$tg\ \beta = \frac{tg\gamma}{\cos \alpha_0}$ and at γ = 35^0 and α_0= 25^0β will be 37^0. Then,
according to (228), the angle ε will be 8^0; the soil layer will move toward the longitudinal axis in the horizontal plane of the paw axis when leaving the paw $\gamma' > \gamma$.
To determine the time t of the soil from the point M to the point A (see Fig. 130), we substitute the value l_{XA} (224) into the first equation of system (222) and transform

$$t = \frac{2\upsilon_r}{g} \cdot sin\ a_r\ . \tag{229}$$

At a_r= 17^0 and υ_r= 1,7 m/s according to (229), (224) and (225) t = 0,1 second, the path l_{XA}= 16,5 sm and the height of h_z = 1,2 sm.
Knowing the length l_{XA}= 16,5 sm and angle ε = 8^0, it is possible to construct an assumed contour of the line of intersection of the spray of the sprayed herbicide solution with the lower plane of the soil formation. Depending on the speed of movement of the aggregate and the physical and mechanical properties of the soil, the descent of the soil from the point M can occur both from the surface of the rear contour of the girder, at a low speed of movement of the aggregate, and from the upper point of the paw knife at a higher value of this velocity. To construct an assumed contour line of intersection, we take arbitrary closely spaced points of the formation from the teardrop and postpone the path length l_{XA} = 16,5 sm in the direction $\bar{\upsilon}_r$, we obtain the line A_nA_n (Fig. 131).

Similarly, we construct the contour line of $A_{\text{н}}A_{\text{н}}$ when the formation comes down from the upper point of the knife. Since in the course of operation the actual position of the contour line will vary between its extreme positions A_nA_n and $A_{\text{н}}A_{\text{н}}$, then it is expedient to construct a contour line $A_{\text{c}}A_{\text{c}}$,, located between the contour lines A_nA_n and $A_{\text{н}}A_{\text{н}}$,, in order to calculate the required number of sprays and the direction of the axis of the spray torch.

To ensure a uniform distribution of the herbicide solution over the area under the paw, the projection of the length of the lines of intersection of the cone-sprayer's torch with the soil layer on the paw-capture width characterizing the line should be equal to each other. This condition can only be fulfilled if the center line of the cone spray is perpendicular to the contour line of the formation of the soil, for example, the line $A_{\text{c}}A_{\text{c}}$, and the vertex of the cone flame will be at the same distance from this line. When using a slit spray with a flare angle of 110^0 for a flat-topped paw with a grip width of 990 mm, seven nozzles are required. The scheme of their placement (three on the right and three on the left side of the paw and one on the center) is shown in Fig. 131.

Thus, if the contour line of intersection of $A_{\text{c}}A_{\text{c}}$ does not change its position (position), and each of the seven injectors feeds the same volume of solution per unit time, the necessary uniformity of the distribution of the liquid over the area under the paw will be ensured.

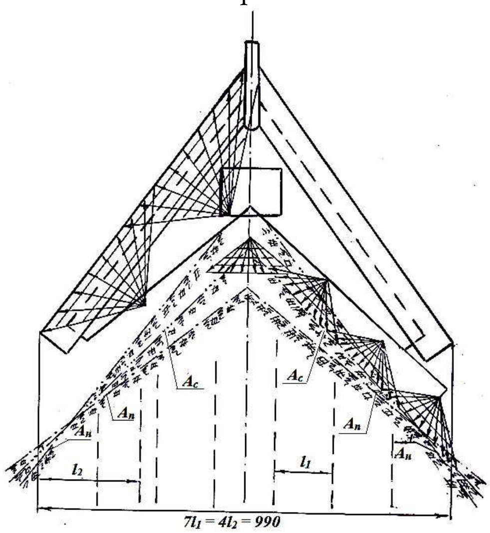


Fig. 131 - Determination of the location of slit nozzles for feeding a herbicide solution to a flat-paw

However, due to changes in the speed of movement of the aggregate, the physical and mechanical properties of the soil and the depth of loosening, the contour line of intersection of the layer of soil coming down from the paw with a torch can be in any position between their extreme coordinates A_nA_n and $A_{\text{н}}A_{\text{н}}$. In this case, since the coordinates of the arrangement of the sprayers do not change, the length of the lines of intersection of the torch and the soil layer at each diffuser will be different. This will not allow the necessary uniformity in the distribution of liquid over the area under the planar paw.

Thus, when a herbicidal solution is applied to a wide-cut flat-paw with the use of the principle of spraying a spray, the liquid under the pile of soil descending from the paw is practically unacceptable.

Solve the task posed is possible when using a soil free from the soil of a paw sub-space. At the same time, as a contour line for crossing the spraying torch, take the blade of the paw knives, properly positioning the sprayers along the line of the trailing contour of the tampon. In this case, if each of the injectors supplies the same volume of solution per unit time, the uniformity of the distribution of the liquid over the area will be ensured, since the coordinates of the soil free from the soil do not change when the speed of the aggregate changes, the physical and mechanical properties of the soil and the depth of loosening. In addition, a considerable distance of the blade of the paw knives from the rear contour of the tear-maker will increase the length of the line of intersection of the spraying torch with the knife. This will reduce the number of nozzles under one paw knife from seven to four. The position of the nebulizers in this case and the lines of the spray flares are shown in Fig. 131.

Experimental studies to determine the effectiveness of the proposed system for uniformity in the distribution of the herbicide solution over the width of the claws were carried out on a special bench (Fig. 132), the results of which are given in Table 25.

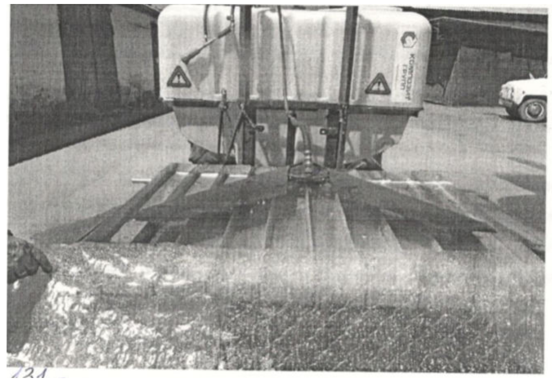


Fig. 132. An installation for determining the uniformity of the distribution of the working solution of herbicides along the width of the claw of the cultivator-plane cutter

Table 25 - Uniformity of the distribution of the working solution of herbicides along the width of the claw of the cultivator-plane cutter

No. of injector -i	The volume of liquid, (see the cube) fed into the i-th nozzle by replicas			The volume of liquid supplied by the i-th nozzle for all replicates	Average replicates of fluid volumes of the i-th injector
	Repetition-n				
	1	2	3		
1	400	410	400	1210	403,3
2	440	400	430	1270	423,3
3	400	390	400	1190	396,7
4	550	550	540	1640	546,7
5	400	410	410	1220	406,7
6	340	340	330	1010	336,7
7	420	410	400	1230	410,0
The sum of all injectors of the n-th repetition	2950	2910	2910	8770	2923,4

Analysis of the data given in Table 25 showed that the non-uniformity of the supply of the volume of liquid between the nozzles was 9.21%, and the instability of feeding the fluid along the width of the cultivator's paw was 6%. According to the agrotechnical (initial) requirements for this operation, the given indices of unevenness and instability of the distribution of the working fluid of herbicides by nozzles along the width of the claw of the cultivator-plane are in the range of acceptable values.

Taking into account the results of the research, a cultivator-planer was developed for 3-tier soil cultivation (10 - 12 sm, 14 - 16 sm, 18 - 20 sm) with simultaneous application of herbicides to the working organs of the second tier. The principal constructive scheme of the proposed tool is shown in Fig. 133.

The tool consists of a frame 1, tillage working bodies 2, support-adjusting wheels 3, a container 4 with the appropriate equipment for introducing herbicides. Aggregated with a 50kN class tractor.

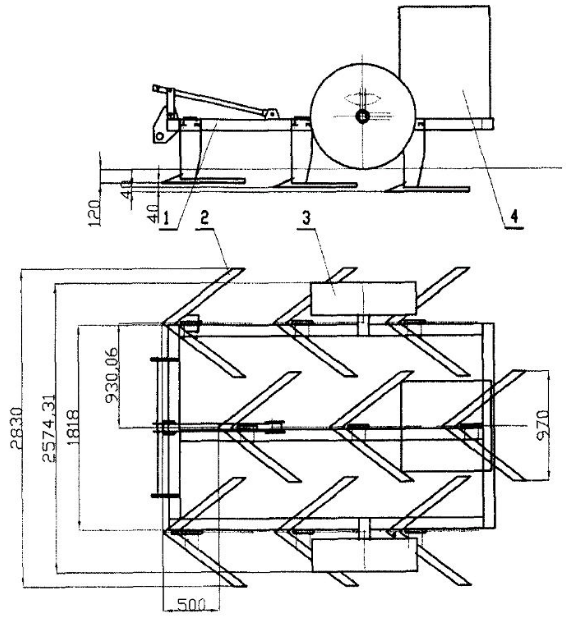


Fig. 133 - Schematic diagram of a 3-tier flat-cultivator with a device for the intrusion of soil herbicides

The results of acceptance tests of the prototype KPTG-2,8 showed that the implement steadily carries out the technological process, ensures uniform supply of the herbicide solution to the sub-space. Destruction of the bitter creeper after combined layer-wise mechanical and herbicidal treatment was 96%.

5. Techniques and means of mechanization for accumulation of moisture in soil

5.1 Accumulation of moisture in soil due to winter precipitation and its effectiveness

With full and rational use of atmospheric precipitation, the annual amount of which in the steppe regions of Kazakhstan is on average 200 - 400 mm, it is possible to significantly reduce the negative effect of drought and obtain relatively high and stable crop yields. However, to form a crop, the precipitation is often used to an insufficient degree, which depends both on the nature of their fallout by the seasons of the year and on the economic activities of the farmers.

Significant weakening of drought, and in some years and its complete prevention, can be achieved by maximum accumulation of moisture in the autumn-winter period. So, the moisture saturation of a meter layer of soil of pure steam to the level of the lowest (field) moisture capacity does not occur. By the time of sowing of spring wheat, the reserves of productive moisture in a meter layer of steam are 120 - 130 mm. At the same time, the field moisture capacity of a meter layer of southern chernozem reaches 180 mm. Consequently, the shortage of soil moisture is 50 - 60 mm. If we consider a one-and-a-half-meter layer of soil, then its deficit is 100 - 120 mm [116]. During the growing season, spring wheat consumes all the productive moisture from the meter layer of soil, leaving only 20-30 mm at the time of harvesting. In years with a high yield, its content is at the level of dead stock.

In autumn, due to rain, a significant amount of soil moisture accumulates. Therefore, before leaving for winter, the moisture content in a meter layer of soil under spring wheat sown with 1st, 2nd and 3rd culture after pure steam averages 30 - 80 mm. Consequently, the shortage of soil moisture in a meter layer on the fields after grain crops is estimated at 100-150 mm.

Thus, both on the steam fields and in the fields after cereal crops, there is a need to improve the water regime with winter precipitation, which, for example, in North Kazakhstan falls on average 50-100 mm. In addition, in the steppe conditions, there are great opportunities for the accumulation of snow in the fields by collecting it from unprocessed lands, since during snowfalls and heavy blizzards the migration of snow takes tens and hundreds of kilometers [118].

In the soil protection system of agriculture, effective methods of detaining and accumulating snow in the fields are: leaving stubble on the surface of the field; cutting of the snow shafts; seeding of the

wings from high-growth cultures on pure vapors; strip seal of snow; stubble scenes and mulching [117].
The stern performs two functions - it protects the soil from wind erosion and contributes to the accumulation of snow in the fields. The snow immediately begins to be deposited and spread evenly over the area. When the stubble was left at a height of 12-15 sm, the thickness of the snow cover averaged 14,7 sm on average over two years; at 20-25 sm – 22,3 sm; at 35 - 40 sm – 30,8 sm. The average height of stubble is usually 15 - 18 sm. After the autumn tillage of cultivators with flat-top cultivators and deep-ripper plowshares, many raised blocks with stubble forms on its surface. As a result, the total height of the stubble increases somewhat, and the average thickness of the snow cover reaches 20-25 sm.
With a snow density of 0,3 g/sm^3, the water reserve will be 60-75 mm, and considering that only 2/3 of snow water is absorbed from the general moisture reserve, we will get an increase in moisture by 40-50 mm. Summing this moisture with a reserve of soil moisture before leaving for the winter, we get a moisture reserve in the meter layer of only 80-130 mm, i.e., the moisture deficit remains on average 50-100 mm. Consequently, the stubble of the usual cut, improving the water regime of the soil, does not allow to eliminate the deficit of productive moisture due to accumulated snow.
It can be eliminated through the production of snow shafts. The work of snowfalls is possible only with a snow cover of 12 - 15 sm in height. Such a layer occurs in the initial period of winter only on stubble backgrounds. Consequently, the abandonment of cereal crops on the fields does not exclude but predetermines the possibility and necessity of using snowfalls. The use of snowfalls makes it possible to bring the thickness of the cover up to 40-50 sm with a twofold (more rarely a single) snow retention, which ensures optimal moisture charging of the fields.
The stern wings 30-35 sm high and 1,5-2,0 m wide through 4,0-5,5 m provide snow accumulation of the same power as a continuous high stubble [117]. However, their wide introduction is hampered by the following: with a low-growth grain, it is not possible to obtain stubble wings of the required height; when harvesting by direct combining, it is necessary to create a special header that allows harvesting crops at a high cut without loss of grain and ears.
Small doses of straw (1 - 2 t/ha) remaining on the field after harvesting, with a single mulching do not affect the water regime and the yield of spring wheat. Positive results were obtained from applying large doses of straw (4 - 8 t/ha), which can not be obtained from the harvest of one year. The accumulation of straw on the same site for several consecutive years improves the water permeability of the soil and reduces the evaporation of moisture, so that the reserves of productive moisture in the meter layer increase by 13,3-19,9 mm compared to the control. With an annual application of straw at a dose of 1-2 t/ha on average over a period of 10 years, the yield increases in comparison with the control by 0,10-0,13 t/ha.
Due to the absence of continuous snow cover on the fields and the impossibility of using snowfalls, wings from various high-stemmed crops are used. The best rocket culture in the steppe conditions is the mustard of the grade Neosypayaschy-2. It grows rapidly, has a vegetation period of 80-85 days. Consequently, it can be sown later - in early July, after two or three surface treatments have been carried out on the steam field for the preliminary cleaning of the field from weeds. By the autumn the wings of mustard reach a height of 1,0-1,10 m, branch well, have time to grow stronger and serve as a reliable means of snow retention.
The difference in moisture reserves between clean and wedge pairs after snow retention is 80-100 mm. However, for example, in Northern Kazakhstan, from the snowfall to the sowing of spring wheat, a lot of time passes (1-1,5 months), during which intensive evaporation of moisture takes place, and by the time of sowing wheat, the difference in moisture accumulation in favor of the wings is 30-50 m, which affects, in the final analysis, the productivity of spring wheat.
However, the accumulation of the necessary thickness of the snow cover on the field surface does not yet mean that the deficit of productive moisture will be eliminated without additional agrotechnical measures for the use of thawed snowmelt by the soil. Assimilation of snow water largely depends on the water permeability of the soil (thawed or frozen). It is noted that the autumn moisture accumulates in the upper half-meter layer and is better preserved in the fields with the autumn planing of the soil. However, deep soaking of the soil (up to 1,0-1,5 m) occurs due to assimilation of winter precipitation [117].

From deep layers of soil (1,0-1,5 m) moisture is extracted by a well developed germinal (primary) root system, therefore it is used more productively. Summer rains soak the soil to a great depth (20-0.25 sm) and are largely spent for evaporation. At the same time, the moisture of summer rains can be effectively used by the nodal (secondary) root system penetrating into the soil at 30 sm [117].
The influence of the methods of autumn soil tillage on the accumulation of snow, the reserves of productive moisture in it and the yield of spring wheat was determined by the All-Union Scientific Research Institute of Plant Protection in 1971-1974, on the southern carbonate chernozem of the Tselinograd region [119]. The results are shown in Table 26.
Analysis of the data in Table 26 shows that soil cultivation by flat-cutting machines improves the absorption of meltwater by 25,2-31,1%, which increases the yield of spring wheat by 0,17-0,34 t/ha as compared to options without autumn tillage and treated with an autumn harrow BIG-3A.

Table 26 - Influence of methods of autumn tillage on accumulation of winter precipitation, stocks of productive moisture and spring wheat yield

Experiment Options	Amount of delayed precipitation, mm	Reserves of productive moisture in a meter layer of soil, mm		Yield of spring wheat, t/ha
		before leaving for the winter	before sowing	
Cutting on 12 - 14sm	71,4	66,4	122,7	1,22
Cutting on 25 - 27sm	70,0	69,1	138,2	1,34
Processing by harrow BIG-3A	68,1	65,8	106,5	1,00
Without autumn processing	63,2	69,4	104,2	1,05

Thus, in the steppe arid agricultural regions of the country, slicing snow rolls on stubble backgrounds in combination with flat-topped autumn tillage and sowing on the stubble pairs can eliminate the deficit of productive moisture in the soil due to winter precipitation and sharply reduce the negative effects of drought.

5.2 Technology of snow retention, coulis sowing and basic agrotechnological (initial) requirements to the means of mechanization

The winter period in arid zones of agriculture is characterized by increased wind activity and the predominance of sub-zero temperatures. The wind speed here reaches 10-15 m/s and more. The total number of days with a snowstorm is 25- 30 days. The air temperature in the winter months varies mainly from -5 to-30 ° C. The snow is constantly transferred and denser with wind (snowstorm). It has a diverse structure and physico-mechanical properties, which depend on its "age" and state. By the beginning of work on snow retention (on open backgrounds this is the second half of December, and on stubble backgrounds - an earlier time), the thickness of the snow cover reaches 10-15 sm, and to the period of massive snow retention - 15-25 sm.
Firnization ("aging") of snow is observed in the first month of winter. Snow cover has a pronounced layered structure: near the surface of the soil, it usually consists of individual large grains; above is grainy snow, loosened by the action of fusion processes; the upper layer is most often a fine-grained frozen snow - a windmill, in some places, covered with freshly-laid snow (spots) [119,120].
The thickness of the wind table reaches 10-12 sm. In the snow, ice layers formed during thaws are sometimes observed. More intense firnification of the snow and its evaporation (with the formation of voids at the surface of the soil) is noted on stubble backgrounds, where usually the snow is laid by a looser layer.

The lower layers of the snow cover have a density of 0,25-0,35 g/sm^3, the wind speed is 0,30-0,50, and the freshly-deposited snow is 0,15-0,27 g/sm^3. The average density of the snow cover is 0,3-0,35 g/sm^3, during the winter it varies insignificantly.

Dense snow at low temperatures has considerable strength and hardness: the hardness of the upper layer reaches 2 MPa and more, and the individual layers sharply oscillate, starting from 0,03 MPa. Resistance of snow to the shift τ_0, for example, in the zone of Northern Kazakhstan, varies within the range of 0,005-0,02 MPa, the tear resistance $\sigma_{отр}$ is 0,004-0,016 MPa, and the compressive strength $\sigma_{сж}$ is 0,015-0,18 MPa. In all cases, at minus temperatures $\sigma_{сж} > \sigma_{отр}$. This indicates that the snow here is mostly in a fragile state and only when thaws it acquires plastic properties.

High density and hardness of snow, as well as constant winds, determined the following technology for the creation of snow rolls: cracking of the snow crust, formation of snow blocks and laying them in rolls. The unevenness of the walls and the top of the rolls contribute to the twist of the snow-air flow and the settling of snow by a long train, and the voids between the blocks contribute to the production of high shafts. The lumpy structure gives the roll stability against inflation. Experience has shown that this technology of making snow rolls doubles the accumulation of snow in comparison with pre-chopped snow and subsequent sealing it directly in the roll.

Snow compaction is effective only if many snow falls in a short period of time, in windless weather and at an air temperature close to zero [118]. This combination of circumstances in the open steppe areas is extremely rare. Snowstorm, which prevails here at low temperatures, does not condense. The snow, which is composed of loosely interconnected grains, is also not compacted at low temperatures, but is crumbled under the action of a sealant.

The bulk of the snow (up to 92%) is carried by a snowstorm in the surface layer up to 10 sm. On this basis, some researchers [121] come to the conclusion that for snow retention there is no need to make high rolls and that the best snow-retaining capacity is not in the form of rolls, but in the form of trenches formed in the snow cover. However, it is not possible to make a deep trench in shallow snow, while a large capacity for accumulating snow forms between the high snow rolls, and the rolls themselves serve as a reliable obstacle in the way of snowstorm. Cutting trenches causes multiple retention of snow, which is unacceptable because of the huge cost of working hours and the amount of work, since snow retention is carried out on many millions of hectares.

During snowstorms, the snowflakes fall out of the snow-wind flow when the wind speed decreases sharply. Snow blades, cut with a snowfall, being blow-back obstacles, effectively reduce the wind speed at a distance equal to fifteen-fold the working part of the roll (exceeding the roll over the snow cover). At the height of the working part of the roll 30-35 sm, the uniform accumulation of snow in the intervalue spaces occurs when the rolls are cut atadistanceno more than 4-5 m between their centers (Table 27).

In practice, different methods of cutting snow rolls are known. Some experts mistakenly believe that the figured methods (cross, spiral, zigzag, etc.) are more effective than cutting the rolls in one direction - across the dominant wind direction. Such a representation is based on the fact that in figured ways, the effect of changing the direction of the wind on the accumulation of snow in the fields is excluded. However, figured methods do not have an advantage over cutting the rolls across the dominant direction of the winds in winter and are less economical than the latter (Table 28).

Table 27-Snow accumulation depending on the width of the inter-rollersspaces (averages for 1978-1979)

Distance between centers of rolls, m	Snow cover thickness, sm	Stock of water in snow, mm
Without snow retention (control)	25	77,3
12	27	82,5
8	31	95,1
5	43	136,0
3	46	142,4

Table 28 - Efficiency of different ways of cutting snow rolls (СВУ unit - 2,6 + ДТ-54)

Variants of the arrangement of rolls	The thickness of the snow cover on April 4, sm	Thickness of additional accumulated snow, sm	Productivity per hour of clean work, ha	Fuel consumption, kg/ha
Across the prevailing wind direction	43	3	6,4	1,23
Cross	38	18	3,2	2,46
By the unfolding spiral	39	19	5,8	1,36
Zigzagging	43	23	5,7	1,42
Without snow retention (control)	20	-	-	-

The operating mode of the tractor unit in winter depends on the combined interaction of the tractor propellers and the snow-covered machine. With the increase in the thickness of the snow cover, the maximum traction power of tractors K-701, K-700, which are mainly used for snow retention, is sharply reduced. Thus, with a snow cover thickness of 8; 15 and 22 sm (snow density 0,33-0,34 g/sm^3), the maximum traction power of the tractor K-701 is 101,9, respectively; 93,8 and 64,8 kW. With a snow cover thickness of 40 sm and a snow density of 0,30 g/cm^3, the K-701 tractor can not move even without a traction load. Local snow deposits thick 30-40 sm are found on the fields quite often [122]. Therefore, to improve traction and coupling properties of the tractor when working with snows, it is desirable that the snow is cleared from the tractor's propellers and used to form a swath.
In connection with the technology adopted for the manufacture of snow rolls and the conditions of snowfalls, the following basic requirements are imposed on them.
Snow should be used on soils of all types and in fields with dump and flat cutting, on crops of winter crops and perennial grasses, on meadows and pastures.
The minimum height of the snow cover at which the machine can be used is 12- 15 sm (at least 15-18 sm on winter plow and winter crops). The average height of the roll with the minimum snow thickness (12 sm) is not less than 40 sm. The distance between the centers of the rolls should not exceed 5 m. When working on the fields of winter crops and perennial grasses, leave a protective layer of snow 8-10 sm thick.
To ensure reliable traction between the tractors and the soil, the implement installed in front of the tractor must ensure that the strip is cleaned of snow before them.
The design of the snow trap should provide for the possibility of its use for cleaning the on-farm roads from snow.
The technology of seeding of rocking crops also has its own specificity, which takes into account zonal features. Mustard is sown in the middle of summer (from June 25 for Northern Kazakhstan), so that the plants can grow, blossom, grow strong and not fall in the fall under the influence of frost. The consumption of mustard seeds is small - 500-600 g per 1 ha of lea field. Like other small-seeded crops, it should be sown to a shallow depth. If by the time of sowing in the soil there is a sufficient supply of moisture, then the depth of seeding should be 3-4 sm. However, if the top layer of the soil is drained, then they must be sealed to a depth of 5 - 6 sm with mandatory packing. In this case it is necessary to carry out furrow sowing.
The wings are sown in a row-wise manner. For most areas of the steppe zone, two-three-row wings with a width of between rows of 15-23 sm are recommended, thickening in a row - 30-40 plants per 1 m.
The width of the interclusal spaces should correspond to the multiplicity of the passage of the tillage tools and aims to ensure a uniform accumulation of snow. When preparing lea field for spring wheat, the width of the interclusal spaces should be used equal to 12 m. After receiving the shoots

of the wings and the germination of the weeds, the interclusal spaces are treated in the same way as in the usual pure pair: flat-top cultivators or rod cultivators.
Seeding of the wings is sometimes suggested to be combined with surface treatment of steam, but this method can be acceptable only in selected cases. The fact is that the period of lea field treatment can be long, and the wings must be sown within one decade. Difficult is also the issue of combining the processing time of steam and the timing of sowing the wings. As is known, the vapors are treated depending on the germination of weed vegetation, sometimes earlier or later than the optimal period of sowing the wings. Hence, the combination of these two operations is inconvenient both for organizational reasons and for technological reasons. Therefore, it is advisable to carry out seeding of the rocking plants with a special rocker.
The technology of seeding of rocking crops and working conditions of seeders determined the following agrotechnological (initial) requirements for them:
- the seeding device of the seed drill must provide seeding of conditional seeds of mustard 30-40, maize, sunflower 8-10, hemp, safflower 15-20 pieces per 1 m row - with a deviation from the set seeding rate not more than ± 10% with an allowable instability ± 10% ;
- grinding of seeds can not exceed 0,3%;
- the drill is equipped with a removable furrower and working elements for the destruction of weeds in the protective zone of the row;
- seeds are sown to the bottom of the furrow evenly one-two rows with a distance between them 10 - 20 sm;
-the depth of seeding is regulated within 3-10sm, with at least 80% of seeded seeds to be embedded in the horizon corresponding to a given average depth and two adjacent 1,0 sm horizons;
- layer of soil in which the seeds are located, is rolled up.
The theoretical and experimental positions and conclusions used in the development of seeding machines for soil protection agriculture (see Section 3) are applicable to selecting the type and substantiating the parameters of the main structural systems of the rocking seeder.

5.3 Snowploughs

Working conditions and physical and mechanical properties of snow caused the choice of technology for the manufacture of snow rolls, the essence of which, as indicated above, is as follows: crumbling of dense snow on large blocks that are not blown by the wind; lifting and moving them to the middle of the machine (to form one roll) and placing the upper blocks in a stable position close to the vertical one.
This technology is best ensured by a ring-dump type of a snow trap with two dumps working in the fall [120].
The optimal values for the parameters of the snowfall pile are regulated by the following conditions: minimal crumbling of the snow layer; uniform motion of snow blocks on the dump (without loading); good car reclamation; minimum traction resistance.
In connection with the variability of the properties of snow, depending on its structure, density and temperature, it is most likely to expect different types of snow deformation by a blade, based on a skewed wedge. When the wedge acts on freshly fallen snow, its deformation will be like the deformation of loose sand [118]. Such snow, with anenough depth of the wedge and a small angle of crumbling, loosens and moves up the wedge or is loaded ahead of it.
Deformation of the compacted snow cover, which is a connected mass with a certain strength, can occur both in the form of detachment and shear. At a snow temperature close to 0°C, when it is more plastic, and at large angles of crumbling, it is necessary to expect a predominance of shear deformation, and at low temperatures, when snow is more fragile, and small corners of crumbling, the predominance of deformation of the separation is prevalent.
The maximum traction resistance of the wedge with snow shear is [120]

$$P_{\mathrm{r}(cd)} = \tau_0 \cdot H \cdot b cos\varphi_1 \sin{(a+\varphi)} / cos^2 \frac{a+\varphi+\varphi_1}{2}\,. \tag{230}$$

It can be seen from the analysis (230) that the maximum traction resistance of the wedge $P_{г(cd))}$ increases in proportion to the snow resistance τ_0, the depth of the wedge H and the width of its capture b. The resistance $P_{г(cd))}$ increases with increasing cracking angle - α. The effect of the friction angle of snow on steel φ is insignificant, since it varies in a relatively narrow range, for example, in the conditions of Northern Kazakhstan - within the range of 0,03-0,12. Great value in this case has an angle of internal friction of snow $\varphi_1 = arc\ \sin\left(\frac{\sigma_{сж}-\sigma_{отр}}{\sigma_{сж}+\sigma_{отр}}\right)$. With increasing density of snow and a decrease in its temperature, the difference between $\sigma_{сж}$ and $\sigma_{отр}$ increases angleφ_1,, which causes an increase in $P_{г(cd))}$.
With this type of deformation, the length of the piece of snow that is cleaved by a wedge (on the top) is determined by the formula

$$l_{cd} = \tau_0 H \cdot b\cos\varphi \cdot \cos\varphi_1 \cdot \cos\frac{\varphi + \varphi_1 - a}{2} / \sigma_{сж}\cos^3\frac{a + \varphi + \varphi_1}{2}\ . \qquad 231)$$

A large value of $l_{cd}$is to be expected with an increase in the temperature of the snow, when it sharply decreases $\sigma_{сж}$ and decreases, φ_1 and also with an increase in the crumbling angleα.
The deformation of a dense snow layer under the influence of a wedge proceeds in the following sequence: the wedge, penetrating the snow, crushes it in a certain area until the ultimate balance of forces occurs within the formation, after which a crack appears at a small angle to the horizon, advancing forward and upward. Moving on, the wedge turns off the torn piece, mixes it on its surface and simultaneously cleans the bottom, making small chips. At a certain thickness of the stripping layer, the formation of small chips is stopped, and the cycle is repeated.
The traction resistance of the wedge at the time of occurrence of a snow cracks in the layer is equal to

$$P_{г(отр.)}=\sigma_{сж}\cdot k\cdot H\cdot b\sin(a + \varphi)\Big/\sin a\cdot\cos\varphi. \qquad (232)$$

From this expression, it is clear that $P_{г(отр.)}$increases in proportion to the strength of the snow - $\sigma_{сж}$, the depth of the wedge - H, the width of its capture b and the coefficient $k = h_1/H($ h_1- the height of the wedge-shaped snow volume), which for each specific case is determined experimentally.
The traction resistance of the wedge after the appearance of a crack in the snow is found by the formula

$$P'_{(г)отр.} = \sigma_{(г)отр.} \cdot bH^2 \cdot sin\psi \cdot tg(\alpha + \psi)/6l, \qquad (233)$$

where: ψ - is the angle formed by the tangent to the leading crack and the horizontal; l- is the distance from the blade of the wedge to the center of application of the reduced forces separating the layer of snow from the massif. $P'_{отр.}$ has a maximum value at the minimum l, i.e., when a crack occurs.
When the tear is deformed, the length of the piece of snow (at the top) is given by the formula $l_{отр.} = k \cdot H \cdot (\text{ctg}\alpha + \text{ctg}\psi)$.
In connection with a sharp decrease in ctgα, with an increase in the angle α, we should expect a decrease in $l_{отр.}$, i.e., an increase in the crushing ability of the wedge.
Since one of the main technological requirements for snowfalls is the minimal crumbling of the snow layer, the choice of the most profitable type of deformation is determined in this case by the condition of the least possible crumbling of snow. If for a given snow $l_{отр.} > l_{cd}$, then the optimal angle of crumbling will have a wedge operating in the tear-off mode. Conversely, for $l_{cd} > l_{отр.}$, the technological requirements are better fulfilled with shear deformation.
The deformation of the detachment will predominate as long as the tear resistance is less than the shear resistance. The boundary condition for the transition of one type of deformation to another is expressed by the dependence [120]

$$P'_{(г)отр.}/P_{г(cd)} = \sigma_{сж}\cdot k\cdot \cos^2\frac{a+\varphi+\varphi_1}{2}/\tau_0 \ \sin a\cdot\cos\varphi\cdot \cos\varphi_1=1, \qquad (234)$$

or taking into account $\sigma_{сж} = \frac{\sigma_{отр}}{n}$, where $n \cong 0,10 \dots 0,25$

$$\sigma_{отр}/\tau_0 = n\cdot \sin a\cdot\cos\varphi\cdot \cos\varphi_1/k \cos^2\frac{a+\varphi+\varphi_1}{2}. \qquad (235)$$

Analysis of business trends shows the following. A definite relation $\sigma_{отр}/\tau_0$, characterizing the state of snow, corresponds to its own boundary angle $a_{гр}$. A wedge with a crumbling angle $a>a_{гр}$ produces a shear, and with $a<a_{гр}$ - the separation of a piece of snow from the monolith.
For snow of different structure but having close values of $\sigma_{отр.}/\tau_0$, we should expect the same limiting angles $a_{гр}$, which is confirmed by experimental data (Table 29).

Table 29 - Characterization of snow deformation by wedges with different cutting angles

Type of snow cover, density	Snow temperature, degrees C	$\sigma_{отр.}/\tau_0$	Snow cover thickness, sm	Type of deformation of snow (O-detachment, C-shear) at the angle of crumbling				Estimated boundary angle, α, deg.
				15^0	30^0	45^0	60^0	
A dense windmill, ρ = 0,37-0,42 g/sm^3	-19	0,65	8	O	O	O	C	54
	-7	1,05	8	O	C	C	C	24
Fine-grained frozen snow ρ = 0,31-0,35 g/sm^3	-22	0,73	8	O	O	O	OC	50
	-16	0,83	4	O	OC	C	C	46
	-16	0,83	8	O	OC	OC	C	46
	-16	0,83	12	O	O	O	OC	45
	-10	1,0	8	O	OC	C	C	35
Granular, loosened the action of firnification	-19	0,70	8	O	O	O	OC	52
ρ = 0,28- 0,32 g/sm^3	-10	0,94	8	O	OC	C	C	44

Hence,qualitative changes in the nature of the deformation of snow have occurred under certain conditions: with a change in the snow state, the transition from deformation to breakage deformation occurred at different angles of crumbling. Thus, in a dense wind bed at $t_{сн}$ = –19°C, the shear deformation was observed only at α = 60°, and at t_{sn}= –7°C for all angles starting at α = 30°. This is due to the fact that at -7°C snow acquired plasticity. A decrease in the boundary angle was also observed in the deformation of fine-grained frozen snow and granular snow, loosened by the action of firnification, when its temperature rose to -10°C. For the most characteristic conditions of snowfall operation $\sigma_{отр}/r_0< 0,9$ and snow cover thickness $H>$ 8sm, the deformation of the detachment will occur under the action of a wedge with a crushing angle $\alpha<$ 45°, and for $\alpha>$ 40° - shear deformation. Larger blocks were obtained at α = 30^0, which is optimal for snowfall piles.
The crushing ability of the blade largely depends on the angle of its setting to the direction of motion - γ (field experiment, ρ = 0,34 g/sm^3, t = -12°C, H = 15 sm, α = 30°, b = 0,50 m). At different angles of setting the wedge to the direction of motion - 15°; 30°; 45°; 60°; 75°, the length of the snow blocks is equal to: 0,76; 0,53; 0,35; 0,24 and 0,19 m. With the decrease of this angle, the size of the lumps increases in the direction of the blade of the blade (length of the lump of snow). Consequently, the snow blade must be installed at a minimum angle to the direction of travel. Admissible crumbling of the snow layer is ensured at γ = 15 - 30°.
In general, the initial position of the choice of angle γ by the condition of uniform displacement of blocks along the blade is γ<90 - φ. Since the angle of friction of the snow along the steel, as mentioned above, fluctuates in a small limit and is approximately 10 °, this condition is satisfied at γ = 80°. But we need to take into account the more stringent restrictions imposed on the selection of the angle by the γ crumbling ability of the blade. Proceeding from this condition, the angle γ should not exceed 30°.

Knowing the angles α that are optimal for the snowfall pile, and the γ,angle of installation of the share to the bottom of the furrow ε (in the plane perpendicular to the blade of the share) is given by the formula ε= arctg (tg $\alpha/\sin\gamma$).
The height and radius of the curvature of the dump are determined by the condition for laying the cropped snow on the working surface of the dump, which in turn is determined by the working capture of the blade b, the angle of its setting to the direction of motion γ and the angle of its placement to the bottom of the furrow ε.
The radius of curvature of the blade can be calculated by the formula of Academician V.P.Goryachkina taking into account the arc of the wing bend. It was experimentally established that the wing bend should not exceed 5 - 8 sm, while $\Delta\varepsilon$ = 20° - 25° [120].
The height of the blade is determined by the formula

$$H_0 = R(\cos\varepsilon + \sin\Delta\varepsilon). \tag{236}$$

With the assumed γ= 30°, $\Delta\varepsilon$ = 50°, $\Delta\varepsilon = 22^0$ and b = 1,5 m, the radius of the blade, found by the formula $R = b/\left(\frac{\pi}{2} + \varepsilon - \Delta\varepsilon\right)\cos\gamma$, will be approximately 93 sm, and the heap height H_0 = 94 sm.
The working grip of the implement should ensure the formation of snow rolls twice as high and three times as thick as the original snow cover.
The transverse cross-sectional area of the roll (Fig. 134), formed by two heaps working by the crest, is

$$S_{\text{в}} = (2S_1/\eta) + S_2\,, \tag{237}$$

where: S_1- is the cross-sectional area of the snow-cut snow layer, determined by the blade's capture width and the depth of the snow cover; S_2 - cross-sectional area of undisturbed snow, determined by the width of the vehicle's back clearance and the depth of snow; η - the coefficient of compaction of snow in the roll.

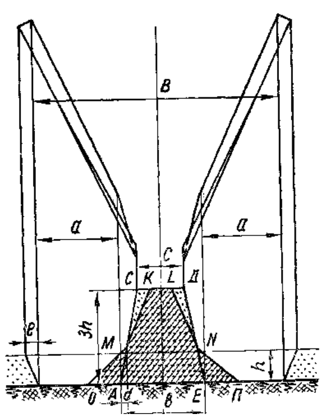


Fig. 134. Scheme of the bridge-snowplough and the snow roll formed by it

The compacting of snow in the felling does not almost occur, and η does not exceed 1,1-1,15. To ensure that the required roll height is taken into account, the maximum value (1,15) must be taken into account.
The formula (237) does not take into account the scattering of small blocks and loose snow, which increases the width of the roll base and decreases its height. As shown by measuring the volume of crumbling snow behind the heap, it is about 30-50% of the snow that they collect. Therefore, the working width of the blade should be increased by approximately 30-50% [120].
The back edge of the blade is run along an inclined straight line, determined by the most rational form of the cross-section of a snow roll - by a trapezoid, that is, taking into account the angle of the natural slope of the snow mass.
In this case, the cross-sectional area of the trapezium, due to the change in the density of the loose snow, should be approximately equal to or slightly larger than the cross-sectional area of the undercut snow, determined by the maximum possible thickness of the snow cover and the working

width of the two dumps. In this case, the cropped snow mass will not be retained in the window formed by the snow roll. In the case of snowfalls on the first snow retention, when the thickness of the snow cover is insignificant (15 - 18 sm), the cross-sectional area of the forming window should be reduced accordingly. For this purpose, the design of the snow trap provides for the possibility of installing additional adjustable openings on the dumps or the rear parts of the small-sized dumps are connected to the main ones by means of vertical hinges with the possibility of fixing them in the desired position.
The upper cut of the back part of the blade, made at an angle of ≈30° in projection to the horizontal plane and with a bend of $\Delta\varepsilon$ =- 25°, ensures the operation of the snowplough without spilling snow blocks through the heap. To increase the strength of the front cut the blade is executed in a plane perpendicular to the blade of the blade of the blade. The height of the front cutoff of the blade must be somewhat higher than the maximum thickness of the snow cover and is approximately 0,50 m. The main parameters of the working organ of the snowfall obtained as a result of the research carried out are realized in the snowplough-dump snowmobiles SVU-2,6 and SVSh-10.

5.3.1 Selection of a rational scheme for wide-graspingsnowplough-rake

In connection with the multi-purpose designation and the need to create a tool that would be aggregated with tractors of different classes, there were great difficulties in selecting the concept and design of the snow trap, which meets all modern agrotechnical, technical, economic and operational requirements. In addition, the development of the construction of the snowfall is complicated by severe winter working conditions; acting on the machine with dynamic loads due to the frozen unevenness of the field microrelief; fluctuations in the depth of the snow cover and its heterogeneity both in structure and in density. Low coupling properties of tractors on snow, especially wheeled, predetermine great difficulties in developing the design of wide-grab snag-swaths, because there is a need to clean the snow ahead of the tractor, i.e., before the propulsors.
Taking into account the difficult operating conditions of the snow retention units, conflicting opinions on the choice of the snowmobile scheme, the test results of the prototype of the hinged wide-spread snowplough created by the Tselinna MIS, it was decided to develop and conduct comparative tests of snowplough in various schemes and with various working bodies at the All-Russian Research Institute [123].
Scheme 1. In the unit for working with tractors of the K-700 type (Fig. 135), there is a blade front wedge 1, which is attached by pushing rods to the front of the tractor, and a trailing part consisting of two wedges 2, the same as the front, rigidly fastened to the frame-farm of the quadrangular form 3 and the sains 4. The working capture of one wedge is 3,9 m, and the entire unit is 14,5 m. In one pass the unit forms two rolls and two half-rolls. At a neighboring arrival the half-roll is shaped by an extreme blade of the wedge.
Each wedge rests on 3 skis of the plate type. They are equipped with a special mechanism and a hydraulic cylinder for lifting the wedge on skis. Trailing part of the unit has the ability to climb on skis at the corners of the unit, as well as to leave a protective layer of snow.
For a long move to the frame-farm are mounted two rolling pneumatic wheels with a hydraulic cylinder for lifting. When the unit is moved from the working position to the long-distance transportation position, the tractor is disconnected from the trailed part of the implement and attached to the end of the frame-truss by means of an automatic coupler; trailing part of the unit rises on transport wheels; the front wedge rises on skis and moves on them.
With tractors of class 3, one wedge is attached to the front hitch of the tractor, while the unit forms two half-rolls. To connect the wedge in front of the tractor, the serial equipment used for hitching the bulldozer onto the tractor DT-75M is used.

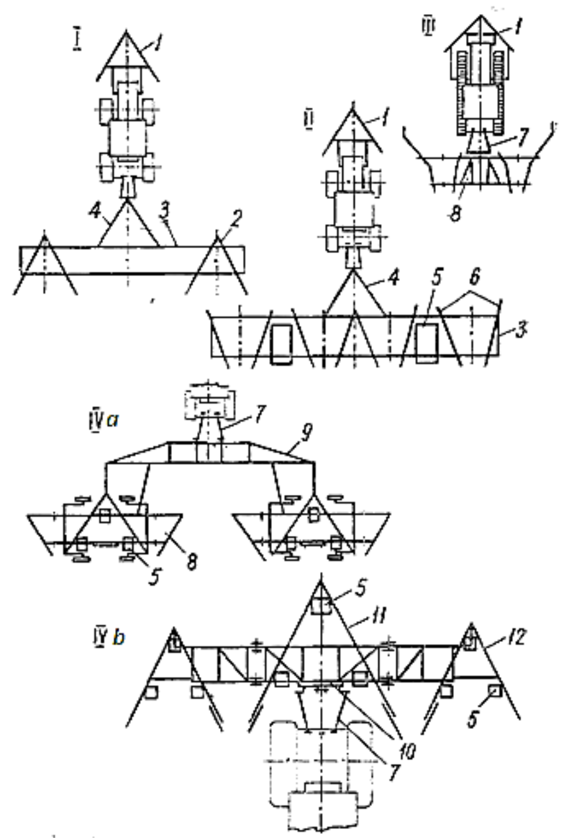


Fig. 135. Schemes of snowploughs: 1 - front wedge for cleaning snow before propulsion; 2 - angle piece for the formation of rolls and half-rolls; 3 - frame-farm; 4 - hitch frame; 5 - supporting ski; 6 - dumps; 7 - mechanism of hitching the tractor; 8 - side section; 9 - hitch coupler; 10 - automatic coupler; 11 - wedge of the middle section; 12 - additional blade

Scheme II. The unit includes a dump wedge 1, hung in front of the tractor of traction class 5, and a trailed tool. The width of the cannon is 14,06 m, in one pass it forms four rolls.
The trailing part of the unit has a rigid frame-truss 3, consisting of four equal parts, connected together by means of flanges and bolts. To the frame are rigidly fastened four ring-dump snowplows. The cannon used snow blades of SVU-2,6, elongated in front. Due to this, the working grip of each rider is increased from 2,6 to 2,75 m. Each rigger is equipped with four slides (as in SVU-2,6) to control the thickness of the protective layer of snow.
The frame-truss is supported by two skis 5, which have a considerable area of the supporting surface. Suspension of supporting skis to the frame is parallelogram. For lifting the implement on skis, hydraulic cylinders are used (2 for each ski). The skis are installed in the intervals between the ridge and go along the untouched snow and must seal it. On skis the machine rises on transport wheels; the front wedge rises on skis and moves on them.
For moving to long distances, the trailing part of the snowplough is equipped with two pneumatic wheels and an additional lowering 4 fixed to the end of the frame. On wheels the machine rises with the help of two hydraulic cylinders.
The wedge attached to the front of the tractor using push rods serves, as in Scheme 1, to clean the snow in front of the tractor propulsors. Working wedge 3,5 m. The wedge is raised with the help of two remote hydraulic cylinders. To clean the snow, one front wedge is used.
The aggregation of the machine with tractors of class 3 in this version has not been decided.
Scheme III. The aggregate with a traction class 5 tractor consists of a wedge, hung to the front of the tractor, two modules, a frame-truss connecting the modules and hitch tongue.

The front wedge 1 has two heaps, connected at the front and rigidly fixed to the frame. Two pneumatic wheels are attached to the frame of the front wedge. In work, they serve as a support for the back of the wedge. With the help of the same wheels and a remote hydraulic cylinder, the wedge rises to the transport position. In front of the wedge rests on a ski with screw adjustment of the thickness of the protective layer of snow.

The module of the trailed part consists of two left and two right dumps, in which the angle of crumbling from the nose to the wings increases. Behind the dumps go into the forming chamber, the exit window of which has a triangular shape.

The front parts of the adjacent module piles are connected and fixed to the frame. The extreme snowplough piles are connected to the frame of the modules by articulation and can be lifted upwards with the help of hydraulic cylinders to reduce the working width of the unit to 4,4 m. The articulated fastening of the extreme dumps provides for lifting them when snow accumulates. Two cross-sectional areas of the exit windows do not change in the two middle formations of the snow trap.

The unit of the two modules has two front and two rear skis. Suspension of each ski is parallelogram. With the help of these mechanisms and four hydraulic cylinders, the implement rises on skis during bends and for transportation. The supporting surface of the front skis attached to the frame-farm is installed below the blade blades of the dumps for the thickness of the protective layer of snow left by the piles. Back skis go along the cleared dumps of the trail.

The frame-farm of the snow trap consists of three sections rigidly connected to each other. The wedge is connected to the frame-truss hinge and is transferred to the position of long-distance transport by means of a tractor. They transport the car for long distances on skis in the transverse direction.

For working with tractors of class 3, one snowplough module is used. The front wedge is assembled from two internal dumps of the module. These dumps are rigidly connected to each other in the bow and a bar in the middle part. The wedge is attached to the tractor by means of beams. To fix the pushing beams to the DT-75 tractor and to raise the wedge, the serial equipment used for hitching the bulldozer onto this tractor is used. To the mechanism of hitching the tractor is attached frame, on which are fixed two sealing chambers and side dumps. In one pass the machine forms two rollers. The road from the snow is cleared of the front wedge.

Scheme IV. A tool for a Class 5 tractor consists of two hinge-sectional modules 8 and a hinged coupling 9 (Scheme IV a), and according to scheme IV*b*, the gun consists of one module.

Modules (scheme IV*b*) consist of middle and two side sections, pivotally attached to the middle. The working organs of the snowflake are cylindrical dumps placed at an angle of $\gamma \cong 30°$ to the direction of travel. In the middle section, two dumps form a wedge, which is rigidly connected to the frame. On the side sections are attached one blade so that together with the wedge heaps they form two ridgers forming snow rollers. The transverse area of the exit window of the pit is regulated by additional dump extensions. When working in shallow snow, extensions extend, the exit window narrows and the roller forms higher, and when working on deep snow, the wings are retracted, the width of the exit window increases, which prevents the car from clogging with snow.

The middle wedge rests on three skis 5, adjustable in height relative to the plane of the blade blades. This ensures the necessary thickness of the protective layer of snow.

To use the module in an aggregate with a class 5 tractor in an attached version, it isequipped with an automatic coupler 10 (Scheme IV*b*).

Due to the hinge attachment of the side sections, the relief is copied by a wide-sweeping machine in a transversely vertical plane. To reduce the machine width in the transport, the side sections are lifted by means of lever mechanisms and hydraulic cylinders. In the hinged version, the snow is transported by the lifted mechanism of the linkage of the tractor 7, with folded in the transverse-vertical plane and fixed side sections. In the trailed version (two modules + coupling), each module is transported on four wheels, which are rigidly attached to the frames of the side sections. In operation, the wheels are placed above the snow and do not interfere with the formation of snow rolls. When the machine is moved to the transport position, the side sections are raised by hydraulic cylinders; At the same time, the wheels fall, which, leaning against the soil, raise all the snow to a given height.

The hinge-section hitch 7 consists of the middle part, attached to the tractor hitch mechanism, and side openings. The module is attached to the coupler openings using a swipe and a stretcher. In work, the hitch is based on skis. When the unit is moved to the long-distance transport position, the coupler openings are lifted by hydraulic cylinders in the transverse-vertical plane and fixed.
To attach the middle section of the module in front of the tractor DT-75, equipment is used to attach the bulldozer to this tractor. For this purpose, snow is additionally equipped with pushing rods.
The test conditions that were carried out in the Tselinograd region were as follows: the field was processed in autumn by flat-saw machines; the snow cover had a relatively stable thickness of 20-25 sm; the snow was redeposited by the wind and had a density of 0,23-0,26 g/sm^3; structure of snow grainy; the air temperature during the test period fluctuated sharply from -5° to -30°C. Such conditions of tests of snowfalls can be considered typical for the second snow retention in the thickness of the snow cover.
When testing the snowfall, performed in accordance with scheme 1, its inoperability was revealed, because the trajectory of the trailing part of the unit in the horizontal plane is unstable. Because of the deflection of the angles from the rectilinear motion, the snow half-wains formed by the front wedge were not shaped to the full roll by the rear modules, that is, the technological process of formation of the rolls was broken. In addition, according to this version, it was noted that the unit has a large mass of 6000 kg; The rigid frame of the trailing part of the snowplough and the rigid attachment of the angles to it on a large span (about 14 m) complicate the copying of the field relief in a transversely vertical plane; the front gon (wedge) is supported by skis during transportation, so it is difficult to transport the unit over long distances, and it is not possible to cross the railroads. Based on these shortcomings, this option was removed from the snowploughs.
Snow Schemes II worked more steadily, although here there were fluctuations in the trailing part of the snowfall in the horizontal plane. The unit is maneuverable - it can be turned on skis and transported from the field to the field. Long distance travel on wheels is reliable. During the tests, it was established that on skiing the trailing part of the unit slips from unevenness (frozen ground clumps, compacted snow crust), while the opposite end of it rises and the conditions for forming a complete snow roll are generally worsened. This is explained by the fact that there is no copying of the field in a transversely vertical plane by the terrain aggregate, since the trailing part of the snowplough has a rigid frame on a large span (about 14 m) and a rigid fixing of the ridge piles to the frame. Two middle dumps of the trailed part of the unit do not participate in the formation of snow rolls, as they follow the track of the front wedge cleared of snow. The depth of immersion of supporting skis is unstable due to the different state of snow on the field; it is possible to "float" the unit on skis when driving on a firm snow crust. On hard snow, the drivers are blocked, since there is no possibility of cleaning the machine from snow, i.e. the exit windows of the drivers have a constant cross-sectional area. The role of large skis, like snow compressors in this zone at a width of 15 m, is minimal - the unit leaves only two shallow strips of 1 m wide.
Trailing part of the unit has a large mass - 7000 kg. In this construction, the question of aggregation with tractors of class 3 has not been solved.
In the case of a snowfall made in accordance with scheme III in the trailed variant, a stable course in the horizontal plane is noted, a stable height of snow rolls (h = 53 sm). The car did not clog on loose snow. The articulated connection of the modules of the side dumps to the frames ensured their raising and the transmission of snow through the side forming chambers. However, on compacted snow, it is possible to block the machine with snow, especially the two middle forming chambers, which have constant cross-sectional areas of the exit windows.
The unit, raised on skis, is maneuverable - it turns and can move on them for short distances.
When checking the unit in the field, it was noted that the middle wedge of the trailed part (two dumps) is practically not involved in the formation of snow rolls, since it follows the track of the front wedge cleared of snow. A large angle of setting the dumps to the direction of motion $\gamma = 35°$ causes intensive crumbling of the snow layer. The module of this snowplough, hung on a class 3 tractor, satisfactorily complied with the requirements for technological processes of snow roll formation and road cleaning.
Testing the unit of two hinge-sectional modules and a hinged coupling (Scheme IV a) showed that it can not work on a snow cover of 25 sm or more in thickness. Such an aggregate was supposed to

be used on the first snow retention, when the thickness of the snow cover on the average does not exceed 12 - 15 sm.
In deep snow, the tractor's propulsors had low coupling qualities, and therefore the hook force was insufficient to move the unit at a uniform speed. The unit was unstable in the horizontal plane. Remote hydraulic cylinders with hydraulic hoses, spaced at great distances from the tractor, did not work because of the solidification of oil in them, so it was impossible to lift the modules on the wheels, and also to raise the side sections when it was required to clean the snow-clogged machine. On this basis, the trailing version of the snowplough, carried out according to scheme IV a, was removed from the tests.
In the hinged version of the snow (SVSh-10), performed in accordance with the scheme IV b, formed snow rolls of height h_{aver} = 57 sm, which meets the agrotechnical requirements. This unit is free to move and turn, the tractor can transport the machine lifted by the linkage mechanism, on any roads at high speed, which is of great importance when working in harsh conditions. The unit worked satisfactorily on cleaning roads from snow drifts.
Effective snowfalls were used to experiment with the accumulation of snow in the fields (Table 30) and determine their energy intensity (Table 31). As a control, a serial snowploughSVU-2,6 were adopted. Both experimental snowfalls satisfy the agrotechnical (initial) requirements imposed on the process of forming snow rolls. The largest resistivity per 1 m of capture is observed in the snowshoe, made according to Scheme III, and the smallest - for the serial snowploughSVU-2,6. This is since the snowfall according to Scheme III is followed by the first grinding of the snow, and then the compaction in the decoration chambers.

Table 30 - Parameters of snow rolls formed by experimental and serial snowfalls, and the thickness of snow accumulated by the end of winter (OPA VNIIZH, 1981)

Snow-plough	Height of snow roll for laying expe-riments, m	Width at base, m	Thickness of protective layer of snow, sm	Distance between rollers, m	The thickness of the accumulated snow cover at the end of winter (2nd of March), m
Scheme III (hinged)	0,53	1,14	5,8	3,60	3,53
Scheme IV b (hinged)	0,57	1,26	3,8	4,50	3,60
СВУ-2,6 (control)	0,36	0,98	4,4	3,70	2,65

Table 31 - Specific resistance of experimental and serial snowshares (APC VNIIZH, 1981)

Snowplough	Operational speed, m/s	Specific resistance per 1 m of grip, N/m
Scheme III (hinged)	1,95	1262,0
	2,31	1405,0
	2,47	1775,6
Scheme IV b (hinged)	1,68	1186,7
	1,92	1211,0
	2,34	1213,6
	2,49	1315,7
СВУ-2,6 (control)	1,75	930,7
	2,13	907,7
	2,55	957,5
	2,80	1180,7

Analysis of the results of comparative tests showed the following.
1. It is inappropriate to develop a single snow to tractors of different traction classes, as the mass of the unit sharply increases, and during the reconfiguration part of this mass is not used; it is practically impossible to carry out readjustment in the harsh weather and climatic conditions of winter; an extended network of oil lines contributes to the thickening of oil in them and the failure of the

hydraulic system; The design of joining the front wedge to the tractor for snow clearing before its propulsors becomes more complicated; transportation of long wedges on the front wedge on skis (or wheels) is unsafe and difficult when moving ditches, roads, etc.
2. All the above-mentioned difficulties of constructive and operational nature, as well as the formation of a snow roll, are satisfactorily solved when creating hinged (semi-mounted) snow-ploughs-swaths, maximally unified among themselves for tractors of Class 3 and 5.
Subsequent preliminary tests of experimental snowfalls confirmed these provisions and were further refined; Snowfall scheme III has not changed much; in the snowfall according to scheme IV b, additional dumps were removed on the side sections, and dumps equal to the dumps of the middle wedge were placed on the side sections to increase the width of the capture of the snowflake.

5.3.2 Stability of the unit consisting of a tractor K-701and a hinged wide-grab snack-rake

When developing a snow-plow-rake to the tractor K-701 according to the scheme IV b, it becomes necessary to check the stability of the unit when moving over long distances. Due to the need to ensure the normal loading of the tractor engine and the formation of snow rollers of a given height, the width of the capture of the snow must be at least 8-10 m, and its mass is 2500-3100 kg.
When the side sections are raised in the transport position, the vertical coordinate of the center of gravity (CG) from the reference surface increases, which may reduce the stability of the unit.
The diagram of the aggregate consisting of the tractor K-701 and the hinged sectional snowshoe in the transport position, as well as the accepted designations of the gravity and coordinates of the tractor's CG and snowfall are shown in Fig. 136, and their numerical values in Tables 32 and 33. Using the method described in [124] and the data of these tables necessary for calculating [125], we find that the coefficient of longitudinal stability of the aggregate (0,189) is much lower than the allowable (0,4). The load on the front axle of the tractor wheels lies in the range 71475-75625 N, on the rear 90375-94525 N. The limiting values of the angles characterizing the longitudinal and transverse static stability of the aggregate are respectively $48^0 43'$ and $38^0 43'$.
However, these data should be regarded as tentative, since in their determination, the limiting values of the angles are not taken into account, the magnitude of which is due to the slip of retarded propellers along the snow surface on the slopes and can be much smaller than those obtained. The need is to determine the limiting angles of longitudinal and transverse slip of the tractor K-701 when working on snow-covered slopes is obvious. When determining the angles, it is necessary to take into account that the frame of the tractor K-701 is articulated.

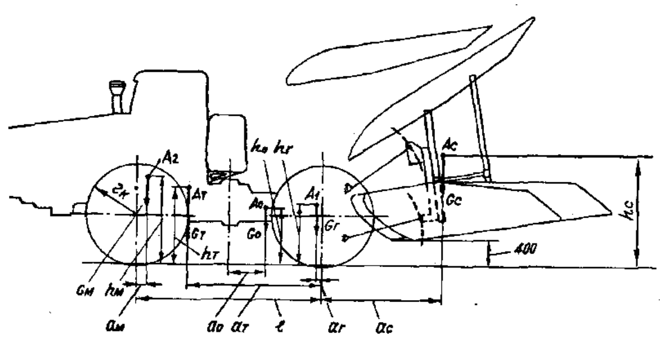


Fig. 136. The design diagram of the unit consisting of the tractor "Kirovets"
and hinged hinge-sectional snowplough

Table 32 - Coordinates of the center of gravity of the mounted hinged sectional snowplough and the forces acting on it

Options	Notation	Value of parameters
Gravity of snowplough, H	G_c	28000
Coordinates of snowplough CT hanging on tractor hitch, m:		
transport position of the snowplough:		
- longitudinal from the rear axle of the tractor	a_c	1,80
-vertical against the support surface	h_c	1,80
the operational position of the snowplough (not shown in Fig. 136):		
-longitudinal against suspension axis	a'_c	0,50
-vertical against suspension axis	h'_c	0,76

Table 33 - Active forces and coordinates of the center of gravity of the tractor K-701 and the unit K-701 + hinged snowplough

Options	Notation	Value of parameters	
		for K-701	for K-701 + hinged snowplough
Gravity of the tractor (unit), H:			
- general	$G_т$	135000	166000
- motor section of the tractor	$G_м$	89500	89500
- cargo section of the tractor	$G_г$	42500	73500
- the central hinge of the tractor	G_0	3000	3000
Coordinates of the center of gravity, tractor (unit), m:			
- longitudinal axis of the wheels of the tractor's load section	$a_т$	2,160	2,160
vertical against the support surface - motor section of the tractor	$h_т$	1,220	1,220
- longitudinal axis of the wheels of the tractor's motor section	$a_м$	0,168	0,168
- vertical against the support surface cargo section of the tractor	$h_м$	1,390	1,390
- longitudinal axis of the wheels of the tractor's load section	$a_г$	0,077	
- vertical against the support surface the central hinge of the tractor	$h_г$	0,890	Variables that depend on the design of the snowplough(a'_cи h'_c)
- longitudinal from the vertical axis of the hinge of the tractor	a_0	0,726	0,726
- vertical against the support surface	h_0	0,850	0,850
Longitudinal tractor base, m	l	3,200	3,200
Track of tractor support wheels, m	b	2,115	2,115
Radius of the tractor's drive wheels, m	$r_к$	0,800	0,800
Distance from the vertical axis of the central hinge of the tractor, m:			
- to the axis of the wheels of the motor section of the tractor	$l_м$	1,600	1,600
- to the axis of the wheels of the load section of the tractor	$l_г$	1,600	1,600

Scheme of the tractor with articulated frame allows to improve the maneuverability of the unit provided that the load distribution on the wheels and the stability of the tractor are correctly dis-

tributed. Since when the frame is folded, there is a significant change in the position of the tractor's CG, the determination of stability for this type of aggregate is decisive. In this case, the orientation (position) of the unit on the inclined plane (slope) should be taken into account.
An analytical solution to the problem of dynamic stability of an aggregate, the tractor of which has a hinged-articulated frame, is associated with mathematical difficulties. In addition, a tractor with a raised snowfall, which reduces the stability of the unit, usually moves at low speeds, so that the effect of dynamic loads is not decisive. In separate works, only the static stability of the tractor is considered. The main provisions of these works were used by us to determine the stability of the aggregate, consisting of a tractor K-701 and a hinged snowplough.
For the unit, the tractor of which has an articulated frame, it is necessary to distinguish the following signs of stability loss: detachment from the slope plane of one of the tractor wheels; irreversible loss of stability and the overturning of one of the sections of the tractor, the rest stops of the other; irreversible loss of stability of the whole unit and its overturning. Since the safety of work on the slope when one of the wheels is torn off can not be guaranteed, this sign of loss of stability is regarded as critical [126].
The stability criterion is taken to be the slope angle γ at which the stability of the aggregate is disturbed in the position in question at the slope characterized by the orientation angle β (Fig. 137)) and when the tractor sections are folded by an angle α (the angle between the longitudinal axes of the sections) [126].

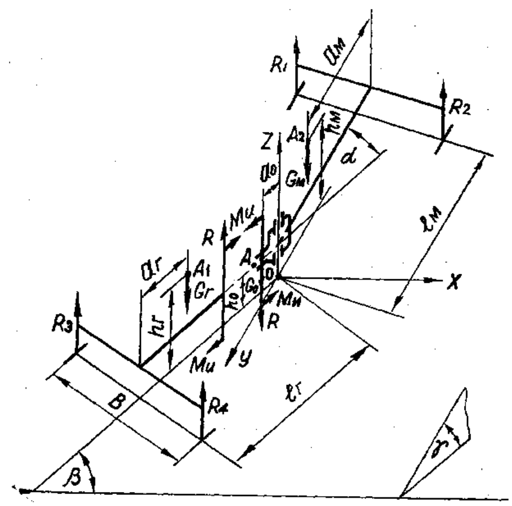


Fig. 137. To the calculation of the stability of the aggregate, the tractor of whichhas articulated frame

The axes of coordinates are chosen as follows: their origin is on the reference plane at the projection point on its axis of the vertical hinge; the axis OX is parallel to the line of intersection of the reference plane with the horizontal plane; axis $OУ$ - perpendicular to the horizontal.
The action of one section on another is replaced by a vertical reaction R and a reactive bending moment $M_{и}$ acting in a plane perpendicular to the supporting surface passing through the projection of the longitudinal axis of the load section onto the support section. The pointsA_0, A_1, A_2 denote the CG corresponding to the horizontal hinge, cargo and motor sections of the tractor; R_1, R_2, R_3, R_4- reactions of the supporting surface to the wheels.
To derive an analytical expression for the dependence of the stability criterion $\gamma_{кр}$ on α and β, the equations of equilibrium of forces and moments that affect each section of the tractor in projections on the coordinate axes are compiled under the assumptions: the wheels are rigid and rest on an undeformable surface; gaps in the joints of parts and the friction torque in the horizontal hinge are not taken into account.
Taking into account the accepted designations (see Fig.137, Table 33) and accepted assumptions, a system of equations for the equilibrium of forces and moments of the motor and cargo sections of the unit with respect to the selected axes coordinates is compiled:

$$\left.\begin{array}{c}
R_4 y_4 - G_r y_r + R_3 y_3 + M_и \cdot \dfrac{\sin\beta \cdot \cos\gamma}{\sqrt{\cos^2\beta + \sin^2\beta\cos^2\gamma}} + R y_R = 0; \\
R + R_3 + R_4 - G_r = 0; \\
R_3 x_3 + R_4 x_4 + M_и \cdot \frac{\cos\beta}{\sqrt{\cos^2\beta + \sin^2\beta\cos^2\gamma}} - G_r x_r + R x_R = 0; \\
-R + R_1 + R_2 - G_м - G_0 = 0; \\
-M_и \cdot \dfrac{\cos\beta}{\sqrt{\cos^2\beta + \sin^2\beta\cos^2\gamma}} - R x_R + G_м x_м - R_1 x_1 - R_2 x_2 - G_0 x_0 = 0; \\
-M_и \cdot \dfrac{\sin\beta \cdot \cos\gamma}{\sqrt{\cos^2\beta + \sin^2\beta\cos^2\gamma}} - R_2 y_2 + G_м y_м - R_1 y_1 - R y_R - G_0 y_0 = 0,
\end{array}\right\} \quad (238)$$

where:

$x_1 = l_м \cos(\alpha+\beta) - \dfrac{b}{2}\sin(\alpha+\beta)$; $\quad y_1 = [l_м \sin(\alpha+\beta) + \frac{b}{2}\cos(\alpha+\beta)]\cos\gamma$;

$x_2 = l_м \cos(\alpha+\beta) + \dfrac{b}{2}\sin(\alpha+\beta)$; $\quad y_2 = [l_м \sin(\alpha+\beta) - \dfrac{b}{2}\cos(\alpha+\beta)]\cos\gamma$;

$x_3 = l_r \cos\beta + \dfrac{b}{2}\sin\beta$; $\quad y_3 = (l_r \sin\beta - \dfrac{b}{2}\cos\beta)\cos\gamma$;

$x_4 = l_r \cos\beta - \dfrac{b}{2}\sin\beta$; $\quad y_4 = (l_r \sin\beta + \frac{b}{2}\cos\beta)\cos\gamma$;

$x_м = (l_м - a_м)\cos(\alpha+\beta)$; $\quad y_м = (l_м - a_м)\sin(\alpha+\beta)\cos\gamma - h_м \sin\gamma$;

$x_R = a_r \cos\beta$; $x_0 = a_0 \cdot \cos\beta$; $a_R = \alpha_0$; $\quad y_r = (l_r - a_r)\sin\beta \cdot \cos\gamma + h_r \sin\gamma$;

$$y_R = l_r \sin\beta \cdot \cos\gamma;$$
$$y_0 = a_0 \sin\beta \cdot \cos\gamma + h_0 \sin\gamma$$

The solution of the system (238) can be represented in the form

$$R_i = A_i + B_i \mathrm{tg}\gamma_i, \quad (239)$$

where: i = 1,2,3,4 - numbers of the tractor wheels (see Fig. 137). The coefficients A_i and B_i for specific values of i are determined by the formulas:

$$A_{1,2} = \frac{1}{b(l_м + l_r\cos\alpha)}\left\{G_м a_м l_м \left[\frac{b}{2a_м}\left(1 + \frac{l_r}{l_м}\cos\alpha\right) - \frac{b}{2l_м}\left(1 - \frac{G_r a_r}{G_м a_м}\cos\alpha\right) \mp \left(\frac{l_r}{l_м} + \frac{G_r a_r}{G_м a_м}\right)\sin\alpha\right] + \right.$$
$$\left. + G_0(l_r - a_0)\cdot\left(\frac{b}{2}\cos\alpha \mp l_м \sin\alpha\right)\right\}; \quad (240)$$

$$B_{1,2} = \pm\frac{1}{b(l_М+l_r cosa)}\Big\{(G_М h_М + G_0 h_0)\Big[\big(l_М \cos(\alpha+\beta) \pm \frac{b}{2} sin(\alpha+\beta) + +l_r cos\beta\Big] \pm G_r h_r \Big(\frac{b}{2}cos\alpha \mp \mp l_М sin\alpha\Big) sin\beta\Big\}; \qquad (241)$$

$$A_{3,4} = \frac{1}{2}\Big\{G_М\Big[\frac{G_r}{G_М}\Big(1+\frac{l_r}{l_М}cos\alpha\Big)l_М + a_М\Big(1-\frac{G_r a_r}{G_М a_М}cos\alpha\Big)\Big] + G_0(l_М + a_0 cos\alpha)\Big\}\cdot \cdot\frac{1}{l_М + l_r cos\alpha}; \tag{242}$$

$$B_{3,4} = \mp\Big\{\frac{G_r h_r}{b}\Big(1+\frac{l_r}{l_М}cos\alpha\Big)l_М cos\beta \mp G_М h_М\Big[\Big(1+\frac{G_0 h_0}{G_М h_М}\Big)sin(\alpha+\beta) + \frac{G_r h_r}{G_М h_М}cos\alpha\cdot sin\beta\Big]\Big\}\cdot \cdot\frac{1}{l_М + l_r cos\alpha}; \tag{243}$$

In formulas (240), (241), (242) and (243), the upper signs "plus" and "minus" are used in calculating the coefficients A and B for odd wheels (i= 1,3), and the lower ones - even (i =2, 4).
The condition for maintaining the stability of the aggregate is written in the form $R_i \geq 0$. Equating the reaction on the wheels to zero, we write down for each wheel the critical state by detaching it from the support surface

$$R_i = A_i + B_i \mathrm{tg}\gamma_i, \tag{244}$$

Solving (244), taking into account the expressions for determining A_i and B_i with respect to γ_i, we obtain

$$\gamma_{1,2} = arc\ \mathrm{tg}\left\{\pm\frac{G_М a_М l_М\Big[\frac{b}{2a_М}\Big(1+\frac{l_r}{l_М}cos\alpha\Big) - \frac{b}{2l_М}\Big(1-\frac{G_r a_r}{G_М a_М}cos\alpha\Big) \mp \mp\Big(\frac{l_r}{l_М}+\frac{G_r a_r}{G_М a_М}\Big)\sin\alpha\Big] + G_0(l_r - a_0)\Big(\frac{b}{2}\cos\alpha \mp l_М \sin\alpha\Big)}{(G_М h_М + G_0 h_0)\cdot\Big[\big(l_М\cos(\alpha+\beta) \pm \frac{b}{2}(\sin(\alpha+\beta) + +l_r cos\beta\big] \pm G_r h_r\Big(\frac{b}{2}cos\alpha \mp l_М \sin\alpha\Big)sin\beta}\right\}; \tag{245}$$

$$\gamma_{3,4} = arc\ \mathrm{tg}\left\{\pm\frac{G_М\Big[\frac{G_r}{G_М}\Big(1+\frac{l_r}{l_М}cos\alpha\Big)l_М + a_М\Big(1-\frac{G_r a_r}{G_М a_М}\cdot cos\alpha\Big)\Big] + + G_0(l_М + a_0 cosa)}{2\frac{G_r h_r}{b}\Big(1+\frac{l_r}{l_М}cos\alpha\Big)\cdot l_М\cos\beta \mp G_М h_М\cdot \cdot\Big[\Big(1+\frac{G_0 h_0}{G_М h_М}\Big)\cdot\sin(\alpha+\beta) + \frac{G_r a_r}{G_М h_М}cos\alpha\cdot sin\beta\Big]}\right\}. \tag{246}$$

These dependencies make it possible to determine the slope angle at which one of its wheels starts tearing away from the support surface, both for the tractor and for the unit, depending on the coordinates of the center of gravity and the orientation angle of the unit on the slope. In this case, the load section of the tractor and the hinged snow are considered as a single whole, i.e. with a common gravity and the coordinates of its application.
For a given width of capture of a hinged snowfall, its material capacity is determined mainly on the basis of the provision of the required strength and to a lesser extent depends on the structural scheme. When developing the design of a mounted snowplough to a tractor of the Kirovets type according to scheme IVb (see Fig. 135), it is possible to vary the coordinates of the center of gravity of the snowfall within a wider range, and consequently, the coordinates of the center of gravity of the tractor's load section with the hanging machine, i.e., the coordinates are $a_r и h_r$.
Using the dependences (245) and (246) and the data required for the calculation from Table 33 the changes in the critical angle of the slope $\gamma_{кр}$ for the aggregate are determined as a function of the

coordinates $a_{г}$ and $h_{г}$, the values of which vary, respectively, within 0,8-1,3 m and 0,8-1,4 m. When calculating the angle of folding of sections of the tractor K-701 was assumed to be minimal (α = 0) and maximum (α = 35°), and the orientation angle of the aggregate on the slope β varied from 0 to 360° with an interval of 10°.

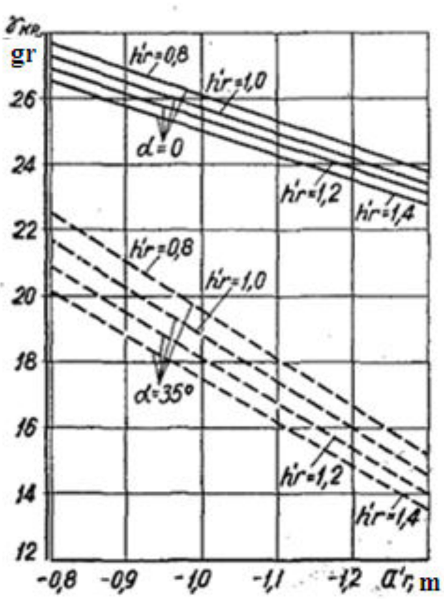


Fig. 138. The influence of the common coordinates of the CG of the tractor's load section and the hinged snowplough (a_r' and h_r') the critical angle of the slope ($\gamma_{кр}$)

Analysis of the results of calculations carried out on a computer shows (Fig. 138) that with an increase in the coordinates a_r (absolute value) and h_r, the angle $\gamma_{кр}$ decreases, and at a maximum angle of folding of the tractor frame α = 35°, the decrease in γ_{kr} occurs more intensively. Thus, with increasing a_r from 0,8 to 1,3 m and h_r = 1,4 m at α = 0, $\gamma_{кр}$ decreases from 26°30' to 22°42', and at α = 35° it decreases from 20°6' up to 13°24'. Removing the common center of gravity of the tractor's load section with the snow from the axle of the tractor's load section (a_r) towards the snowfall affects more the critical slope angle than the change in the overall vertical coordinate of the center of gravity of the snowplough and the load section (h_r). Therefore, when developing the design of a hinged snowfall according to the adopted scheme, it is necessary to place the suspension axis as close to its center of gravity, i.e., closer to the bow part of the wedge of the central section, as far as the solution angle of the blades in the plan allows. In this case, it should be borne in mind that when lifting the side sections of the snowfall in a transverse-vertical plane, they do not touch the tractor cab and do not reduce the view to the tractor driver.
The hinged sectional snowshoe to the tractor K-701 weighs 3100 kg, and the longitudinal coordinate of the center of gravity of its ax relative to the axis of the tractor's rear wheels in the transport position of the unit with the raised side sections and the vertical coordinate h_c from the reference surface is -1,8 m. Then The total mass of the cargo section of the tractor with a suspended snowplough is estimated at 7,350 kg, and the coordinates of the center of gravity a_r' and h_r' are respectively 1,3 and 1,4 m. Substituting these data into formulas (245) and (246), and also from Table. 33 the influence of the orientation angle β of the tractor without a snow trap and with it at a maximum α = 35° and the minimum α = 0 angles of folding of the tractor sections is determined, by the magnitude of the critical angle of the slope $\gamma_{кр}$ (Fig. 139).

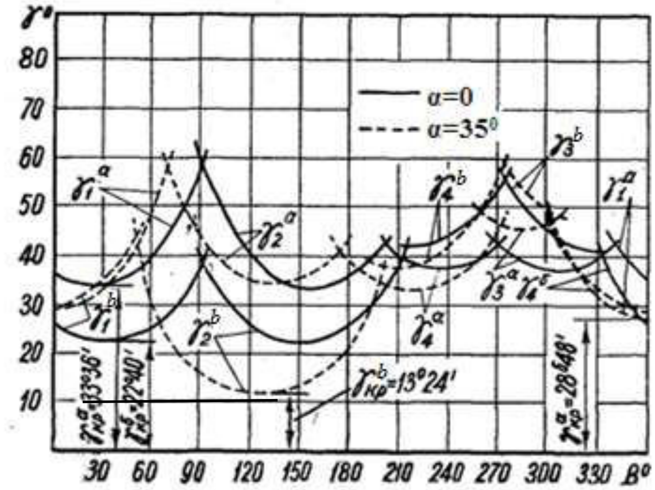


Fig. 139. The limiting values of the angles of the slope (γ°) of the tractor K-701 γ_i^a and an aggregate consisting of K-701 + snowplough (γ_4^b), dependingon the orientation angle and the folding of tractor sections

From the analysis of these data it follows that the critical angle of the slope $\gamma_{кр}$ for the tractor K-701 without the folding of its sections (α = 0) is 33°36', and with the maximum insertion of them (α = 35°) - decreases to 28°48'. Mounted on the tractor and raised in the transport position of snowmobiles reduces the stability of such an aggregate. For uncharacteristic operating conditions of the unit, i.e. when the tractor sections are completely folded up to 35° and the unit is positioned on the slope at an angle of approximately 40°, the critical angle of the slope decreases to 13^{0}24'. Without a snow tractor attached to the tractor K-701, the critical angle of the slope at α = 35° is approximately 30°.

For the most characteristic operating conditions, that is, at small angles of folding the tractor sections and placing the aggregate approximately across the slope ($\beta \cong 20°$ and $\beta \cong 180°$), the critical angle of the slope lies within 20° - 25°.

5.3.3 About the loading of traction motors on snow retention

In most cases, the tractor "Kirovets" is aggregated with only one snowploughSVU-2,6A. This is to some extent explained by the absence of a special coupling, with which it would be possible to compile aggregates of two or three SVU-2,6A snowshoes, as well as the complexity of operation of such an aggregate at low air temperatures and high wind speeds, especially when driving rolls "windows" with snow. The main reason for K-701 aggregation of only one SVU -2,6A is that with the increase in the thickness of the snow cover, the maximum tractive effort and traction power of the tractor, as indicated above, sharply decrease. This decrease is explained by large losses of power for self-movement and slippage of the propellers, which is confirmed by the energy evaluation of the snow retention unit consisting of the tractor K-701 and one SVU-2,6A, or K-701 and special coupling and two or three SVU-2,6A [127]. So, at speeds of 797 to 11,7 km/h, snow cover thickness of 20 - 38 sm and snow density of 0,180-0,310 g/sm^3, the power consumed to overcome the resistance of snows of SVU-2,6A is 23,65- 45,16% of the effective engine power, self-movement 31,32 - 54,00%, on slipping 6,00 - 33,40% and for mechanical losses 11,00 - 13,00% (Fig. 140).

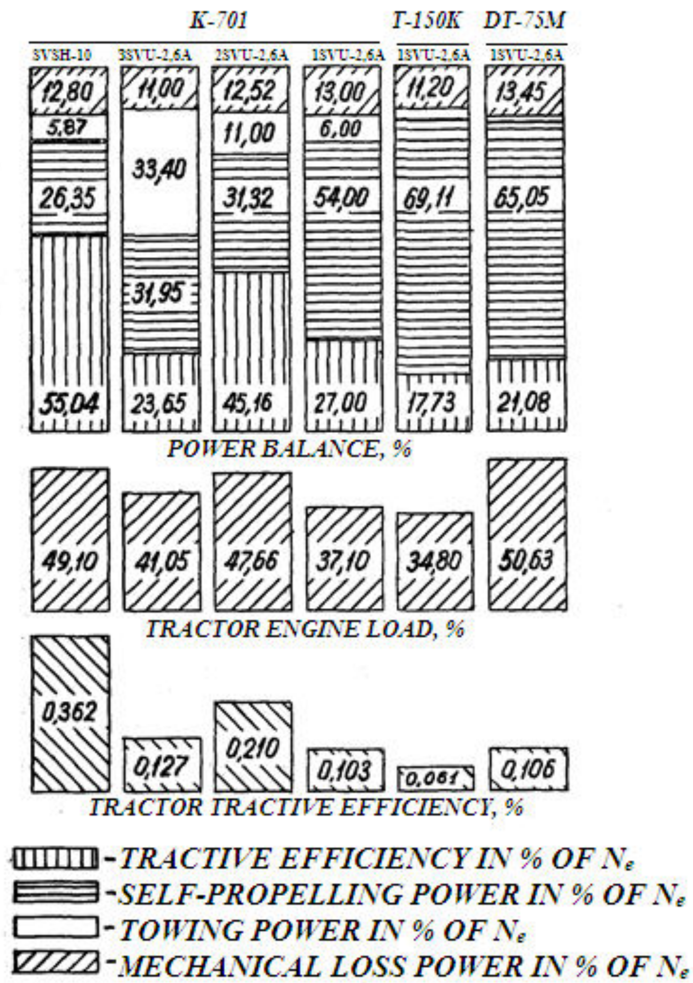


Fig. 140. Energy performance of snowploughs during snow retention

Thus, self-movement and slippage of the tractor K-701 when working with 1-3 snows of SVU-2,6A expends 42,32-65,35% of the engine power, and the efficiency is very low – 0,103-0,210.
The design of the hinged hinge-sectional snow bag to the tractors "Kirovets", made according to the scheme IV*b*, allows dramatically improving the power parameters of the unit due to snow cleaning from under the wheels of the tractor. Thus, under the same conditions and operating conditions as for the K-701 with SVU-2,6A snows, the aggregate consisting of the K-701 and the hinged sectional snowshoe, the capacity to overcome the resistance of the snowfall was 55,04% from the effective power of the engine, to self-movement – 26,35, to slippage – 5,87 and to mechanical losses – 12,80%.
The application of a hinged snowplough, implemented in accordance with the scheme IV b, with the tractor K-701 allows, in comparison with the SVU-2,6A, to increase the use of traction power in 1,22-2,33 times, to reduce power losses on slipping and self-movement in 1,31 - 2,03 times and to increase the tractive efficiency to 0,382, i.e., 1,82 - 3,71 times [127].
Energy performance during operation of the K-701 with the middle section of the hinged sectional snowshoe on clearing the on-farm roads from snow is practically the same with the same indicators when special road-cleaning machines are used in this operation (Fig. 141).

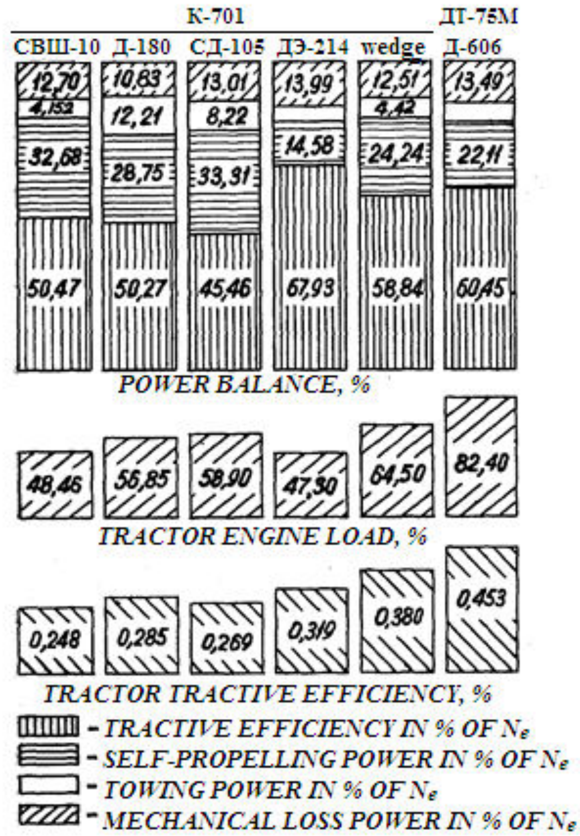


Fig. 141 Energy performance of the middle section of a hinged snowplough and special road-cleaning machines for clearing roads from snow

According to the IIA, the load factor of the K-701 tractor engine on snow retention with a hinged snowfall was 9,5 m wide and was within 40-60% (on average 49,1%), which experts consider insufficient. Such a belief is based on the available normative data on the loading of tractors' engines on plowing (78-94%) or on sowing and planting (85-96%). Data on the value of the rational load of the engine K-701 on snow retention are not available to date. In connection with this, we tried to determine the optimal value of the load factor from the following considerations.

It is known that the permissible load factor of a tractor engine depends more on the nature of the technological process performed by the agricultural unit [34,128]. Changes in traction resistance of the snowfall, superimposed on continuous changes in the resistance to the movement of the tractor, lead to unstable operation of the unit and affect the degree of utilization of engine power. The latter, under unsteady loads M_{en}can be determined by the formula

$$N_{en} = N_{er}\left(1 + q'\varepsilon_p n_{dv} - \frac{q''M_{cc}\delta_c}{V_{\sum a}\cdot m}\right), \quad (247)$$

where: N_{er} - engine power for stand test (calculated);q', q'' - are the experimental coefficients characterizing the engine design, ε_P- is the degree of unevenness of the regulator, n_{dv}- is the average engine speed of the crankshaft, M_{cc}- is the average moment of resistance on the motor shaft, $V_{\sum a}$ - is the reduced moment of inertia of the unit, δ_c - is the degree of unevenness the moment of resistance on the motor shaft, m - is the frequency of the change of the moment of resistance on the motor shaft.

This equation shows that the higher the average load and the degree of unevenness of the resistance, the lower the developed engine power. The expression in parentheses of the equation is taken as the correction factor $a$to the value of the calculated power [34, 128], that is,

$$N_{en} = N_{er} \cdot a. \tag{248}$$

The value of the coefficient a is found experimentally for various conditions and shows the reduction in engine power due to the unsteady operating mode, i.e.,
$a< 1$.
The degree of unevenness of the resistance on the motor shaft (δ_c) is determined mainly by the degree of unevenness of the traction resistance of the agricultural machine. Under normal conditions, small oscillations of the traction resistance of the agricultural machine will be overcome due to the kinetic energy of the unit. Large fluctuations of the traction resistance of agricultural machines can be overcome only due to some reserve of power (engine torque). Therefore, the second correction factor is introduced into the formula (248) b<1 [126], т. е.$N_{en} = N_{er} \cdot a \cdot b$. The coefficient b shows how much the real power of the N_{en} engine can be used with variations in the traction resistance of the agricultural machine. The value of b depends on the value of the coefficient of adaptability of the engine $k_{dv}$and the degree of unevenness of the traction resistance δ_p or its standard deviation from the averageσ_p. To find the value of b, the dependences [128]

$$b = \frac{k_{dv}}{1 + 0{,}5\delta_c} \quad \text{если } \frac{\delta_c}{2} > k_{dv} - 1; \tag{249}$$

$$b = \frac{k_{dv}}{1 + \delta_c / P_{aver}} \quad \text{если } \frac{\delta_c}{2} < k_{dv} - 1. \tag{250}$$

Condition (249) is possible when plowing dry or stony soils and in other similar cases. The nature of the oscillations of the resistance on the motor shaft δ_c is different than the oscillation of the resistance of the working parts of the machines $\delta_p$and σ_p. However, it is also recommended to take into account $\delta_c \cong \delta_p$ and $\sigma_c \cong \sigma_p$ for the calculation, since there are areas that are more unfavorable in unevenness than the average in the field. To determine δ_p offered the dependence [34]:

$$\delta_p = (P_{max} - P_{min})/P_{aver} \, , \tag{251}$$

where:$P_{max}, P_{min}, P_{aver}$ - accordingly the maximum, minimum and average value of traction resistance of an agricultural machine.
The degree of unevenness of the traction resistance of the snowfall depends not only on the parameters of its working organs and operating modes, but also on the operating conditions-varying snow cover thickness, snow density and field background. In this case, the background of the field can have a decisive influence on both the magnitude of the traction resistance and the degree of unevenness. When the SVU-2,6A snowfall was dynamo-measured on various background fields, it was found that the deviations of the mean maximum resistance values from the total average value were within the limits of: 10-40% in the fields of winter crops and perennial grasses, and 80-170% on uneven ground [120]. In the latter case, the large unevenness of the traction resistance is due to the fact that the share of the snowplough meets frozen ridges and bumps of soil in its path. To determine the maximum, minimum and average values of the traction resistance of the hinged sectional snowplough to the tractor K-701, its strain gauging was performed on the fields of the OPC VNIIZH (Tselinograd region). Background - the field after the wheat, processed in the autumn flat-cut to a depth of 12 - 14sm. The thickness of the snow cover varied from 5 to 40 sm (average 22,4 sm), and snow density from 0,109 to 0,400 g/sm^3 (mean 0,300 g/sm^3). The air temperature is 22°C, the wind speed is 10 - 12 m/s.
The analysis of the obtained data showed that, with an increase in the speed of the unit from 4,39 to 9,47 km/h, the maximum forces in the left and right lower links of the linkage mechanism of the tractor K-701 lie, respectively, in the range: 38,32 - 57,48 kN and 23,03-75,01 kN, and the minimum effort is 1,67-2,49 kN and 2,30-4,60 kN. In this case, the average values of the forces in the left and right lower links of the linkage lie respectively in the range 19,98-29,80 kN and 13,16-39,48

kN. According to these data and formula (251) for a hinged snowfall, the degree of unevenness of the traction resistance is determined δ_{p}, which is within the range 1,70-1,83.
The coefficient of adaptability of the engine characterizes its ability to overcome temporary overloads due to the reserve of engine torque and is in relation to [128]

$$k_{dv} = M_{dm}/M_{dp}, \tag{252}$$

where: M_{dm} - maximum engine torque;
M_{dp}- is the torque corresponding to the maximum effective power and normal engine speed.
Based on the results of the control braking on the MIS of the ЯМЗ-240B engines, which have worked for 100-800 hours, the coefficient of their adaptability lies in the range 1,217-1,270 (aver k_{dv}= 1,242).
The value of the coefficient b for the hinged snowfall, found by the formula (249), since$\frac{\delta_{\mathrm{c}}}{2} \cong \frac{\delta_{\mathrm{p}}}{2} >$ k_{dv}-1 lies within the range of 0,647-0,671. Consequently, $N_{en} = N_{er} \cdot a$ (0,647-0,671). If $a = 1$ is assumed (ideal case, although it is always < 1), then for the mounted snowfall the maximum rational load of the K-701 tractor engine will be approximately 60-70%, which is clearly lower than the standard adopted at the MIS. The values of the load factors of the engine of the tractor K-701, obtained at the MIS, when they are mounted with a hinged snowfall made according to the scheme IVb, are close to our calculated ones.

5.3.4. Pressure in the hydraulic cylinders controlling the turning of the tractor K-701 when working with a hinged wide-spread snowplough

When developing a wide-spread hinged snowman to the tractor K-701 according to the scheme IV b, there is a concern that the hydraulic cylinders controlling the tractor's turning can cause pressure exceeding the permissible value. If the pressure is close to the maximum, it can be assumed that this will adversely affect the wear and durability of the entire control mechanism, including the hinge connecting the cargo and motor sections of the tractor. The test to determine the pressure in the hydraulic cylinders for controlling the turning of the tractor K-701 during the assembly of the hinged wide-spread snowplough was carried out [129] under the same conditions as for the determination of forces in the lower links of the weigher mechanism [127].
To measure the pressure in the steering hydraulic cylinders, pressure sensors TDD-200 were installed in them. To measure the angle of the folding of the motor and cargo sections of the K-701, a special sensor was installed in the articulation points. The process of changing the pressure in the hydraulic cylinders by turning and the angle of folding the sections of the tractor K-701 were recorded on the tape with the help of the H-700 oscilloscope. Power, amplifying and balancing blocks, as well as an oscilloscope were installed in the tractor cab.
The change in the pressure in the steering cylinders and the angle of the folding of the sections were fixed both in place (without movement), with buried snowfall, and during the movement of the unit, i.e. when performing a technological operation (snow guard). The test results showed the following. The maximum pressure in the pressure cavities of the hydraulic cylinders controlling the turning of the tractor K-701 in place with a buried snow trap when the sections are folded up to 32^0 and back to zero does not exceed 6,2 MPa (Fig. 142).

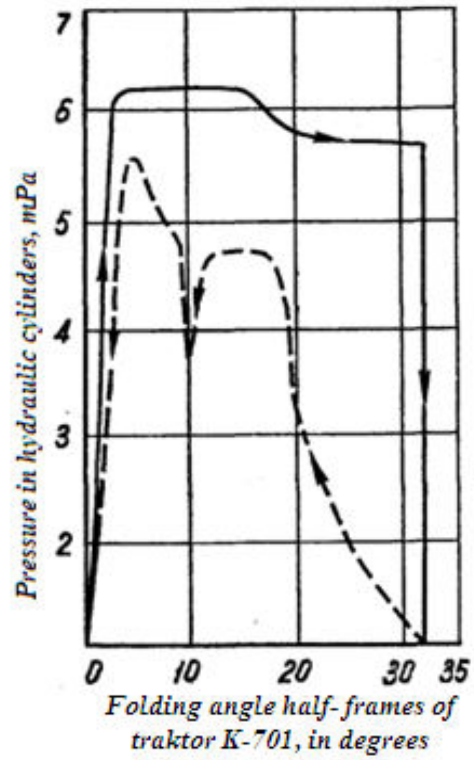


Fig. 142. The pressure in the hydraulic cylinders controlling the rotation of the tractor K-701 when the section is folded up to 32°C (→) and back (← ––) with the recessed hinged snowfall in place (average of 5 replicates)

This case is the most unfavorable and not typical for the operation of such an aggregate. It can occur only in the case of complete slippage of the unit (stopping) when overcoming snow drifts of great depth or when crossing ravines, ditches, etc.
During the operation of such an aggregate on snow retention at speeds of 4,39 to 9,47 km/h, the average pressure in the hydraulic cylinders controlling the rotation of the tractor K-701 varied between 0,47-1,65 MPa, and the maximum 2,40-5,66 MPa (Table 34).

Table 34 - Pressure in the hydraulic cylinders controlling the rotation of the tractor K-701 when the hinged hinge-sectional snowplough is mounted

Indicators	Traveling speed, km / h					
	4,39	4,72	4,86	7,49	8,57	9,47
Pressure in the hydraulic cylinders of tractor turning control, MPa						
a) in the right:						
maximum	3,198	5,058	4,058	3,944	4,428	1,968
moderate	0,861	1,476	1,551	1,022	0,990	0,615
b) in the left:						
maximum	3,150	3,150	2,400	4,950	4,200	4,500
moderate	0,753	0,469	0,629	0,825	0,900	1,650

The nature of the pressure changes in the hydraulic cylinders controlling the rotation of the tractor K-701 in these cases is shown in Fig. 143. Thus, at angles of folding sections up to 35° and vice versa, the maximum pressure in the hydraulic steering cylinders does not exceed 5 MPa.

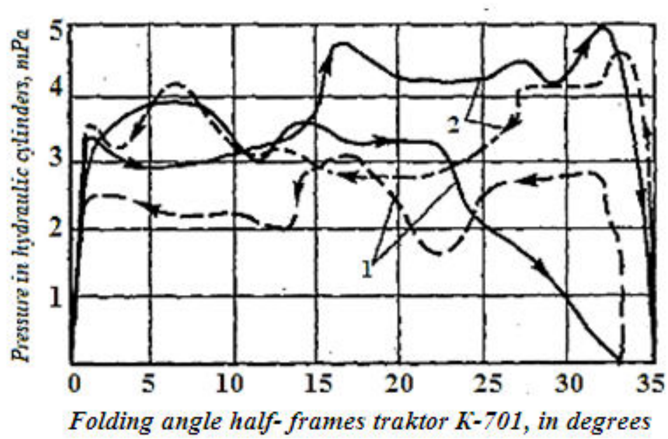


Fig. 143. Pressure in the hydraulic cylinders controlling the turning of the tractor K-701 while folding its section on snow retention with a hinged snowplough up to 35° (→) and back (← ‒‒))
(speed of movement – 4,72 km/h;1 and 2 are repetitions)

Thus, even in the most unfavorable and uncharacteristic operating conditions, i.e. when the K-701 tractor's cargo and motor sections are folded in place with a recessed hinged wide-spread snow-plough,

the maximum pressure in the hydraulic cylinders is 38% less than the maximum permissible (the bypass valve is adjusted to 10 MPa). When carrying out the same technological process of snow retention with a hinged wide-swivel hinge sectional snowshoe, the pressure in the hydraulic control cylinders of the turn K-701 is less than the maximum permissible by 50%.

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