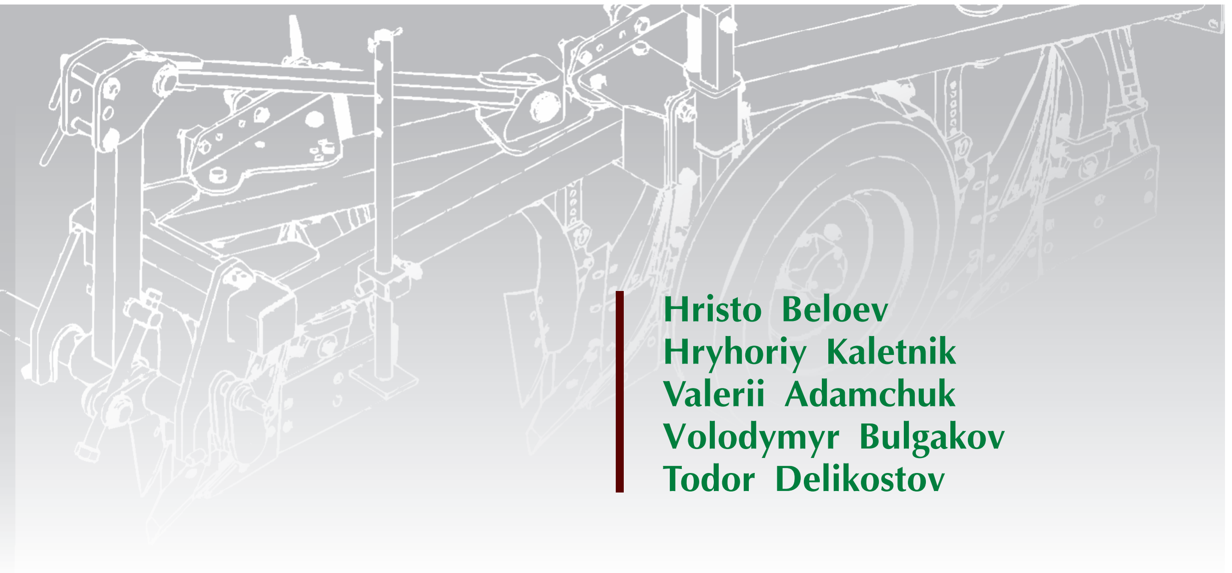

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THEORETICAL BASIS AGGREGATION OF PLOWS

Monograph

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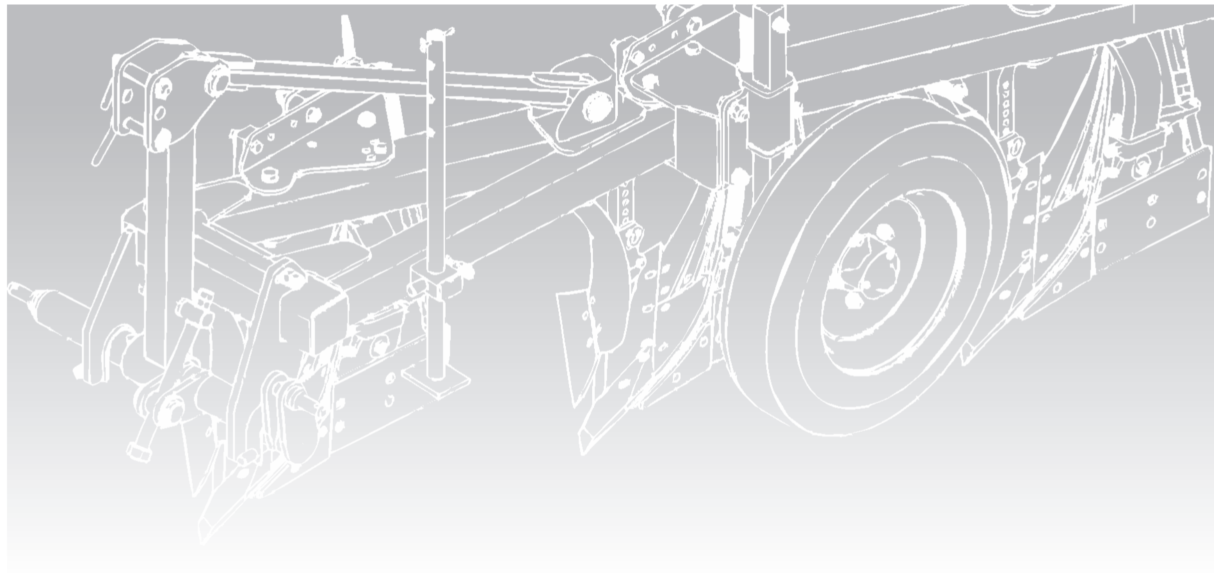

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The monograph covers the theoretical basis for combining plows with tractors of the classical and integral layout, as well as with modular power packs of a variable traction class. Separate consideration is given to the aggregation of front-mounted plows as part of arable machine-tractor units operating under the "push-pull" scheme, as well as their combined hanging options. The monograph is intended for researchers, specialists, master degree students, as well as postgraduates and engineers in the field of "Agroengineering".

The monograph was reviewed and recommended for publication by the Academic Council of the Vinnytsa National Agrarian University (Minutes No. 1 of 27.08.2020); Scientific Council of the National Scientific Centre "Institute of Mechanisation and Electrification of Agriculture" of the National Academy of Agrarian Sciences of Ukraine (Minutes No. 17 of 22.04.2021); The Department Council of Department of Agricultural Machinery, Angel Kanchev University in Ruse (Minutes No.8 of 16.09.2021).

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CONTENTS

INTRODUCTION

Mechanical soil tillage not only creates the most favourable conditions for crop growth and development, but also ensures continuous improvement of soil properties and fertility.

Despite the great variety of tillage tools, the plow has been, is and will remain one of the main working tools for deep tillage in the foreseeable future.

However, despite its relatively simple design, this tool is practically the only one that, under the same soil conditions, in the same working mode, when aggregated with different means of power (tractors) can create (and usually creates) different traction resistance.

This result is due to the ratio between the width of the power vehicle's undercarriage and the working width of the plowing tool in question. To ensure high traction and power output, minimum fuel consumption and good process performance the plow's working width should be greater than the width of the undercarriage of the wheeled or caterpillar tractor used to ensure high traction and energy efficiency, minimum fuel consumption and good workability. However, this requirement is often not met, resulting in asymmetrical plow aggregation with all the possible negative consequences that this can entail. The latter can only be prevented by an in-depth knowledge of the characteristics of the combination of plows with a particular power vehicle. This applies, in particular, to the fundamentally new advanced modular power packs of the variable traction class, whose parameters enable plows to be attached not only symmetrically, but even with an energy-saving left-hand lateral displacement.

The lack of the necessary aggregation patterns for front-mounted plows is currently holding back the introduction of future push-pull plows.

The results of the research undertaken under the direction and with the direct participation of the authors of this monograph address these very problems.

CHAPTER 1
SPECIAL FEATURES OF THE PLOW'S AGGREGATION

1.1. Aggregation schemes for mounted and semi-mounted plows

Mounted and semi-mounted plows from various manufacturers are mainly connected to the aggregating tractor via couplings. The most common is a coupling consisting of beams and struts. The PLN-5-35 plow, shown in Fig. 1.1 is equipped with a coupling device consisting of beams 12, with pins 13, struts 11, with a connection lug 9. It is these elements that ensure the plow is securely connected to the tractor during plowing and transport operations.

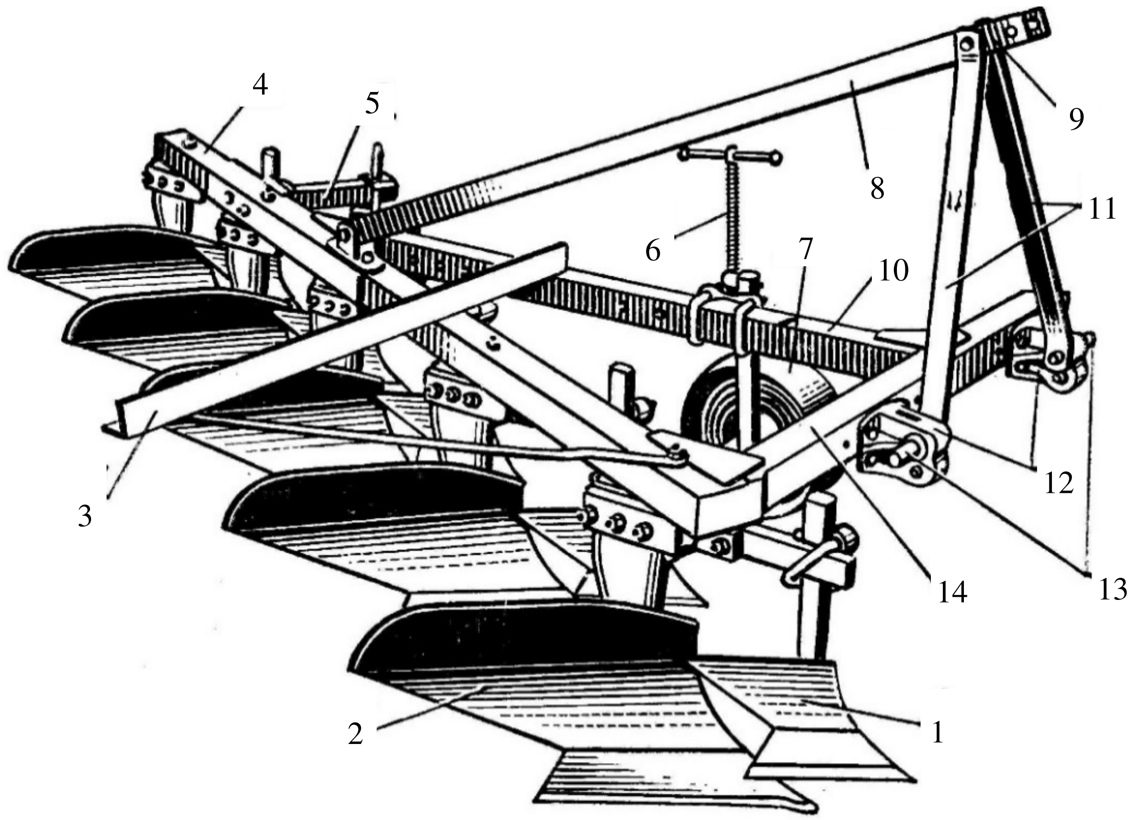


Fig. 1.1 – Plow PLN-5-35:
1 – share; 2 – moldboard (plow body); 3 – harrow hitch; 4 – stiffening beam; 5 – disc coulter; 6 – screw to adjust plowing depth; 7 – support wheel; 8 – strut; 9 – connection lug; 10 – longitudinal beam; 11 – linkage strut; 12 – beams; 13 – connecting pins; 14 – frame crossbar

Recently, an automatic coupler has been widely used, with its lock 2 secured to the plough frame 1 and the coupling triangle 3 attached to the rear aggregation of the mounting tractor. (Fig. 1.2).

The beams on the cross beam of the plow frame can be moved laterally and locked in one position or another. This is necessary in order to properly aggregate it with one or the other power vehicle.

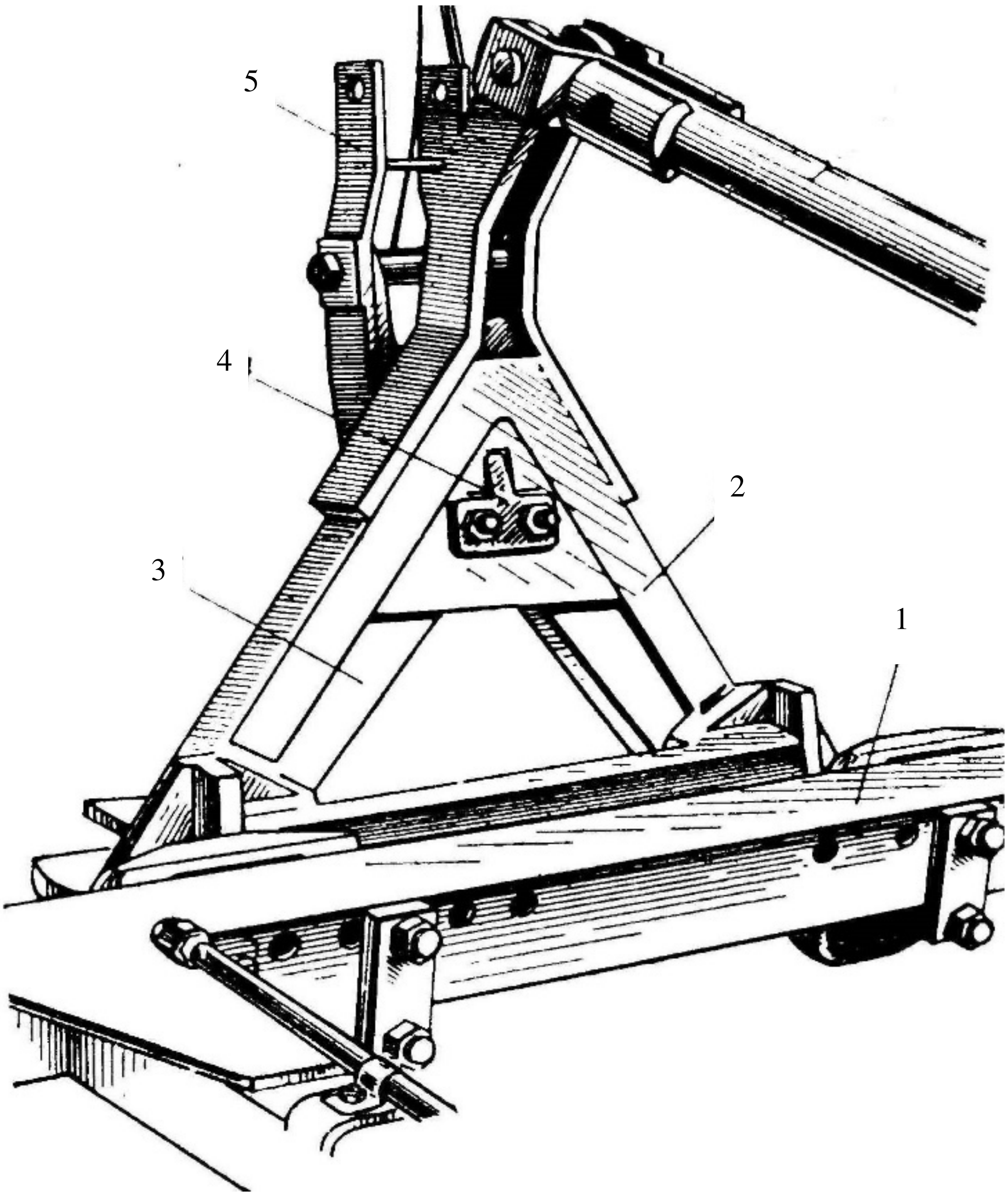


Fig. 1.2 – Automatic coupling device of the plow:
1 – plow frame; 2 – coupling lock; 3 – coupling triangle;
4 – locking lug; 5 – connecting lug

On some plows, the automatic coupler lock is also mounted with the ability to move it laterally across the frame (see Fig. 1.2).

Correct coupling of the tractor with the plow is defined as a position that ensures the least possible drag resistance when the plow is moving steadily in the horizontal plane.

As you know, the stability of the plowing tool in the horizontal plane is provided by the field boards, which are in contact with the furrow wall under the influence of the transverse component of soil resistance. It should be emphasized that the pressure of the field boards produces an additional friction force which increases the plow's traction resistance. According to studies conducted, this increase can average up to 17 % [1].

In addition, the soil of the furrow wall itself is partially deformed in the transverse direction, causing the plow's own oscillating movements in the horizontal plane. And the nature of these oscillations in relation to the trajectory oscillations of the tractor is quite independent. Therefore, even if the power vehicle moves absolutely straight, the trajectory of the furrow wall after the plowing machine has passed will be distorted to some degree.

During plowing, the operator influences the tractor's steering controls by following the trajectory of the previous open furrow. The reaction of the latter is to change course by turning its frame accordingly.

To make sure that this transition process does not affect the stability of the plowing tools, the design of the tractor's rear linkage must ensure (within certain limits, of course) that they can turn independently in the horizontal plane. For this purpose, the lower beams of the rear aggregation of the power vehicle are not positioned in parallel, but at a certain angle of convergence. The two attachment points for these beams and the attachment point of the central beam to the tractor express the essence of the three-point rear linkage adjustment.

Until recently, the view was that because of the much slower response time of a caterpillar tractor's rear linkage, the angle of convergence of the lower beams in the horizontal plane should be greater than that of a wheeled tractor. This was achieved by attaching these beams to the frame of the power vehicle at practically the same point. This results in a two-point linkage arrangement for the rear linkage of the tractor.

Since a three-point linkage scheme is used to work with other agricultural machines and tools, the rear linkage of some power vehicles (e.g. tractors: HTZ-100, HTZ-160, HTZ-170, HTZ-180, T-150) allows their reconfiguration from one scheme to another.

The layout of the connection between the plow and the aggregating tractor determines the traction and energy performance, productivity, straight-line stability, steerability and the quality of the plowing machine's performance. This is because it is structurally difficult to achieve an optimum ratio between the plow's working width and the tractor's track. The result is a momentum that tries to turn the plowing machine horizontally. Depending on how well the plowing tool's coupling pattern is selected, the torque will be higher or lower and the performance of the machine will be worse or better. Let us analyze these issues in more detail.

With a two-point attachment, the angle ε of the plow is almost equal to the angle β of the left and right lower links of the rear linkage of the power vehicle (Fig. 1.3, a).

The three-point attachment has $\varepsilon < \beta$, because in this case the rotation angles of the lower links of the tractor rear linkage are not actually equal to each other (Fig. 1.3, b). With sufficient accuracy the value of angle ε can be found according to the following expression [2]:

$$\varepsilon = (1 - l_T . r^{-1}) . \beta \, , \qquad (1.1)$$

where l_T and r – parameters of the tractor rear linkage shown in Fig. 1.3, b.

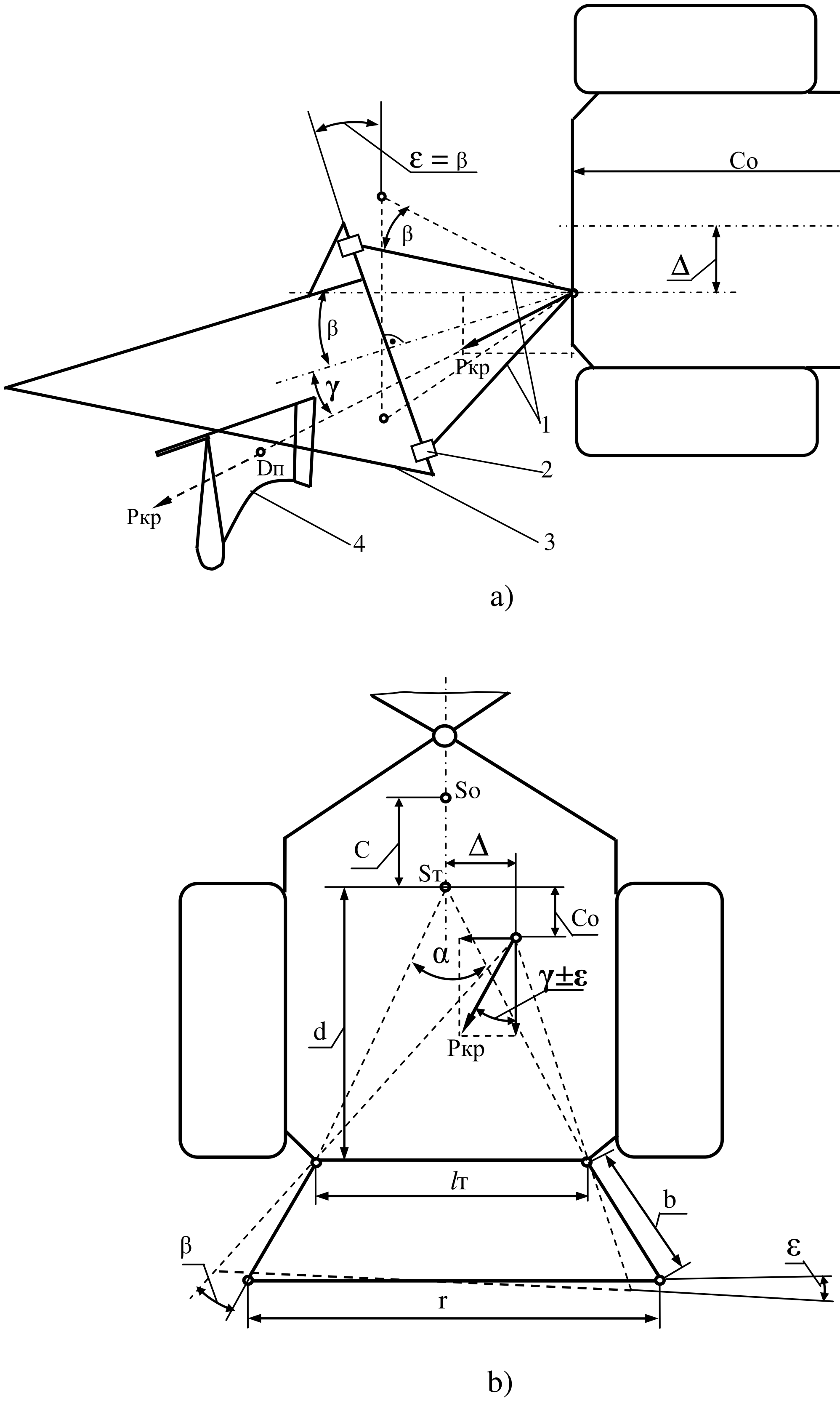


Fig. 1.3 – two-point (a) and three-point (b) schemes of attachment of mounted plow to tractor: 1 – lower links of tractor rear linkage; 2 – plow bolster; 3 – plow frame; 4 – “equivalent” body

It follows that the two-point linkage design generally provides greater angular mobility of the plow, relative to the aggregating tractor. However, even with the three-point linkage scheme, they can provide such mutual angular rotation, when the rotations of the frame of the power vehicle, which arise under the influence of the tractor operator, will not violate the stable movement of the plowing tool in the horizontal plane. This demands that the angle α of convergence of the tractor rear linkage lower links meet the following requirement:

$$\alpha = 2.\arccos[d.(r - l_T).l_T^{-1}.b] \geq 0{,}38 \text{ rad. } (22^o)\,, \tag{1.2}$$

where d – distance from the axis passing through the attachment points of the lower links to the instantaneous "center of rotation" (point S_T, Fig. 1.3, b) of the tractor rear linkage;

b – length of the lower links of the power vehicle rear linkage.

Moreover, at the moment of time when the angle β will take zero value, the fulfillment of condition (1.2) will ensure the coincidence of the instantaneous "center of rotation" of the rear linkage of the power vehicle with the point lying on the axis, passing through its "center of mass" (point S "O", Fig. 1.3, b). This helps to minimize the M_P torque, affecting the tractor through the plow, when it is connected in this way.

Let us further consider how the value of M_P is influenced by fluctuations in some parameters during operation of the plowing unit. In general, the unfolding torque can be expressed as follows:

– when adjusting the rear linkage of the power vehicle according to the two-point scheme (see Fig.1.3, a):

$$M_{p2} = P_{kr}.[\Delta.\cos(\gamma \pm \beta) - C_o.\sin(\gamma \pm \beta)]\,; \tag{1.3}$$

– according to the three-point connection scheme of the plowing tool (Fig.1.3, b):

$$M_{p2} = P_{kr}.[\Delta.\cos(\gamma + \varepsilon) \pm (C_o + C).\sin(\varepsilon)]\,;$$

or according to (1.1):

$$M_{p3} = P_{kr}.\left\{\Delta.\cos\left[\gamma \pm \left(1 - l_T.r^{-1}\right).\beta\right] \pm \left(C_o + C\right).\sin\left[\gamma \pm \left(1 - l_T.r^{-1}\right).\beta\right]\right\},$$

where P_{kr} – plow resistance;

Δ – transverse displacement of the point of application of the drag force of the tool, relative to the longitudinal axis of symmetry of the power vehicle;

γ – the angle between the direction of the power vehicle's pulling force and the longitudinal axis of symmetry of the plow;

C_o – distance in longitudinal direction from the hook load point either to the "center of mass" of the power vehicle (for a two-point linkage), or to the instantaneous "center of rotation" of its rear linkage (for a three-point linkage);

C – the distance between the "center of mass" and the "center of rotation" of the rear hitch of the aggregating tractor.

The sign "–" before the second summand of the last expression is taken when the change of angles γ and β during the work of the plowing unit is in antiphase. In the case of in-phase change, the sign "+" is taken.

In the second variant, as you can see, the value of the torque affecting the power vehicle on the plow side is greater. In order to consider the more stressed state of operation of the plow – tractor unit for further analysis, we assume:

$$M_{p3} = P_{kr}.\left\{\Delta.\cos\left[\gamma \pm \left(1 - l_T.r^{-1}\right).\beta\right] + \left(C_o + C\right).\sin\left[\gamma \pm \left(1 - l_T.r^{-1}\right).\beta\right]\right\} \qquad (1.4)$$

From the analysis of expressions (1.3) and (1.4) it follows that one of the conditions for M_{p2} and M_{p3} to be equal to zero is that Δ and angles γ and β are equal to zero.

Angle γ will be equal to zero only when the plow's "resistance centre" (point D_P, Fig.1.3, a) is located in the longitudinal-vertical symmetry plane, which cannot always be achieved.

The rotation angle β of the lower links of the rear linkage changes according to the random function law and takes the zero value as an instantaneous value when fluctuating from positive to negative value and vice versa.

Since the pulling force is always greater than zero when working with the plow, it is possible to avoid the appearance of the torque, as follows from expressions (1.3) and (1.4), only if the following conditions are simultaneously met:

$$\Delta = 0, \tag{1.5}$$

$$C_o = 0, \tag{1.6}$$

$$C = 0. \tag{1.7}$$

In the case of two-point connection of the plow to the aggregating tractor, condition (1.5) can be fulfilled, but condition (1.6), as it can be seen from Fig.1.3, b, cannot. In the case of three-point connection of the plow, conditions (1.5) and (1.6) are functions of angles β of plow rotation and convergence α of the lower links of the power vehicle's rear linkage. With sufficient practical accuracy, the values of Δ and C_o can be determined by the following analytical expressions proposed by us:

$$\Delta = \frac{l_T.\beta}{2.\tan\left(\alpha.2^{-1}\right)}, \tag{1.8}$$

$$C_o = \frac{l_T}{2.\tan\left(\alpha.2^{-1}\right)}.\left[1 - \cot an\left(\alpha.2^{-1} + \beta\right).\tan\left(\alpha.2^{-1} + \beta\right)\right] \tag{1.9}$$

As follows from the analysis of expressions (1.8) and (1.9), only at zero value of the angle β will values Δ and C_o be equal to zero.

Concerning condition (1.7), it can be fulfilled only if there is a coincidence of coordinates of the tractor's “center of mass” and the momentary “center of rotation” of the rear linkage unit. However, the effect of angle β on the formation of the expanding momentum remains when conditions (1.8) and (1.9) are satisfied.

Let us consider the nature of changes in the torque acting on the tractor T-150K when it is coupled with a mounted plow type PLN-5-35, whose traction resistance is at the level of 30 kN. The design parameters of the rear linkage of this power vehicle are as follows: l_T = 0,58 m; b = 0,88 m; r = 0,91 m; d = 1,5 m.

We emphasize that the coordinate S_T (Fig.1.3, b) of the instantaneous “center of rotation” of the rear linkage of the T-150K tractor does not coincide with the coordinate of its “center of mass” (point S_o) at the value $C \approx$ 0,8 m. In order to achieve their alignment in one point and obtain a minimum value of the torque, it is necessary to increase (if technically possible) the value of the parameter l_T within the condition (1.2).

Next, let us consider symmetrical plow aggregation, when for a two-point link of the tractor's rear linkage $\Delta = \gamma = 0$, and for a three-point one – $\gamma = 0$.

In this case, when the turning angle β of the lower links of the rear linkage of the power vehicle increases, the torques that act on the side of the plowing tool increase as well. With the two-point attachment scheme, this process is more intense than with the three-point scheme (Fig. 1.4).

Asymmetric aggregation of the plow with the three-point attachment of the rear linkage of the aggregating tractor takes place when angle $\gamma \neq 0$.

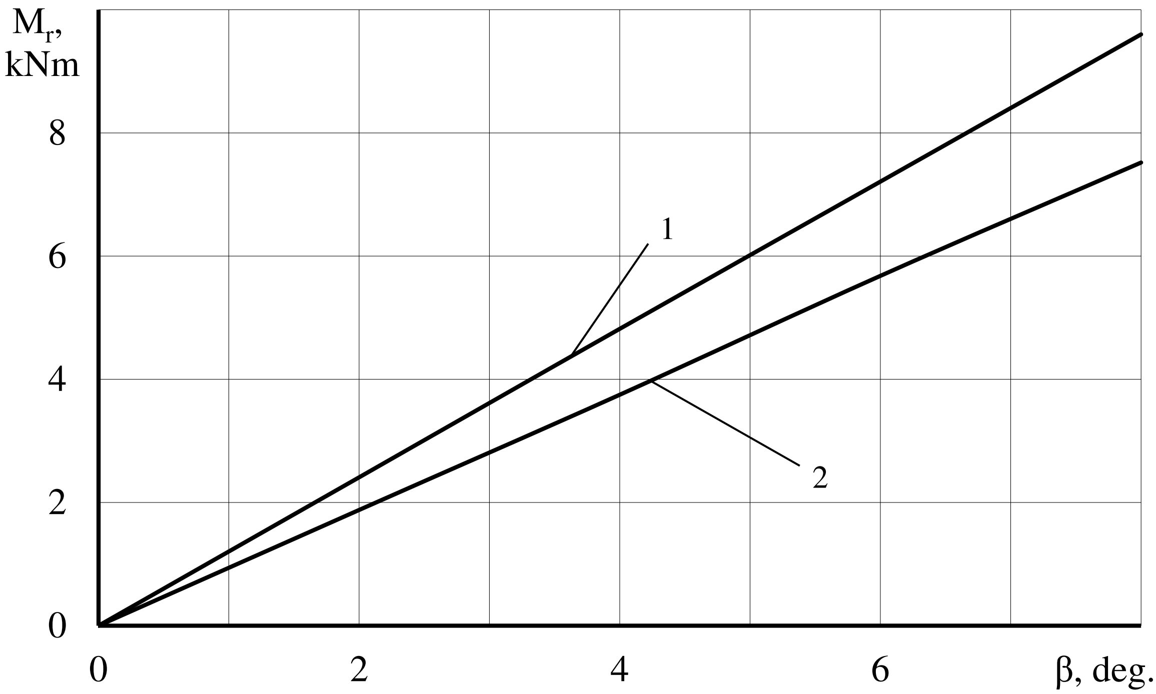


Fig. 1.4 – Dependence of the torque on the angle β for two-point (1) and three-point (2) schemes of rear linkage to the tractor

For a two-point attachment, such aggregation of the plowing tool is possible when the rear linkage of the power vehicle is set to one of the following two options:

1) $\gamma \neq 0, \quad \Delta = 0;$

2) $\gamma \neq 0, \quad \Delta \neq 0.$

In the first variant of the two-point linkage setup, not only is the torque significantly greater than in the three-point linkage, but it is also characterized by a significantly greater intensity of growth with an increase in the angle γ (Fig. 1.5).

In the second version of the two-point linkage, the torque at $\Delta \neq 0$ is almost twice as high as the torque with the three-point linkage.

As the right side lateral displacement Δ of the lower linkage convergence point of the rear linkage increases, the torque decreases (Fig. 1.6). At Δ = 0,21 m its value is equal to the torque acting on a tractor with a three-point attachment of a plowing tool.

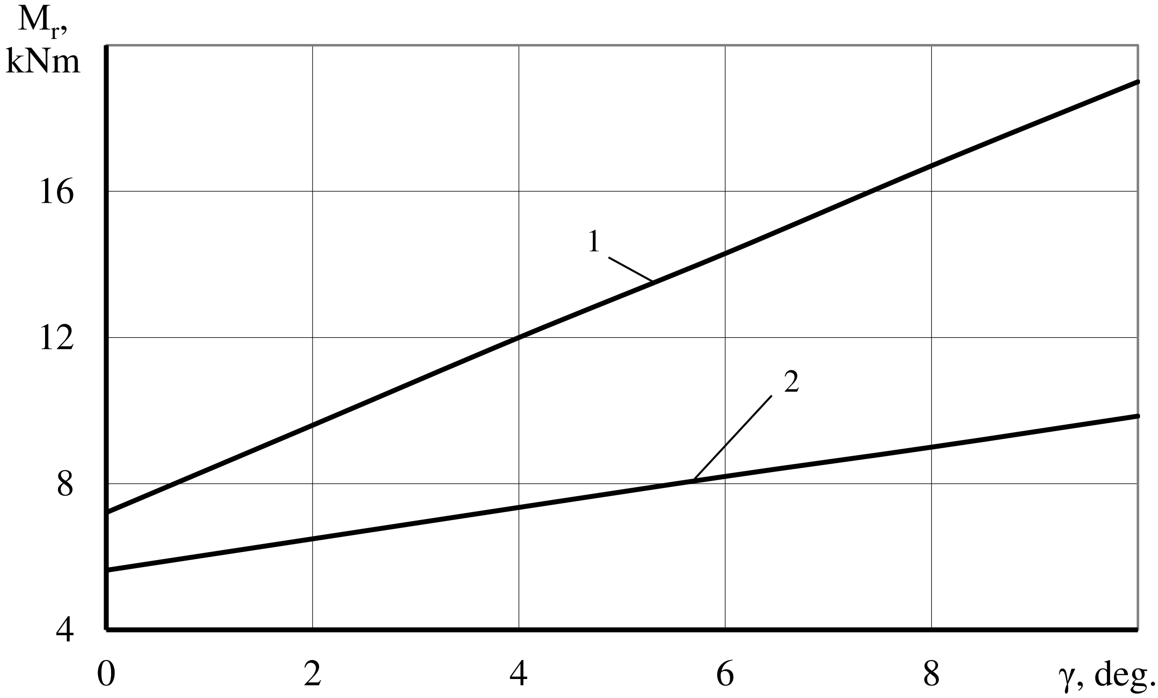


Fig. 1.5 – Dependence of the torque on the angle γ for two-point (1) and three-point (2) rear linkage of the aggregating tractor (β = const = 6°)

The above analysis shows that even if the "center of mass" of the tractor does not coincide with the "center of rotation" of the rear linkage, the torque acting on it from the plow, attached to the three-point attachment, is less than that for the two-point attachment of the unit. From this point of view, the first of them (i.e., the three-point linkage scheme) is preferable, provided that the requirement (1.2) is met.

However, to address the issue of abandoning the two-point linkage pattern of rear linkage units on aggregate tractors, we must consider the following: until recently, it was postulated that the least drag resistance of a plow could be obtained when it was symmetrically connected to the aggregating tractor. The following requirements should be met for this to be true [1]:

$$B_T = b_k.(n+1) - 2.A - b\,, \tag{1.10}$$

$$B_{\dot{T}} = b_k.(n+1) + b\,, \tag{1.11}$$

where B_T, $B_{\dot{T}}$ – the track of the power vehicle when moving its starboard propellers outside the furrow and in the furrow respectively (in this case the movement of caterpillar tractors with a plow is only outside the furrow);

b_k – working width of the plow body;

n – number of plowing tool bodies;

A – distance from the furrow to the outer edge of the propeller;

b – tractor propeller width.

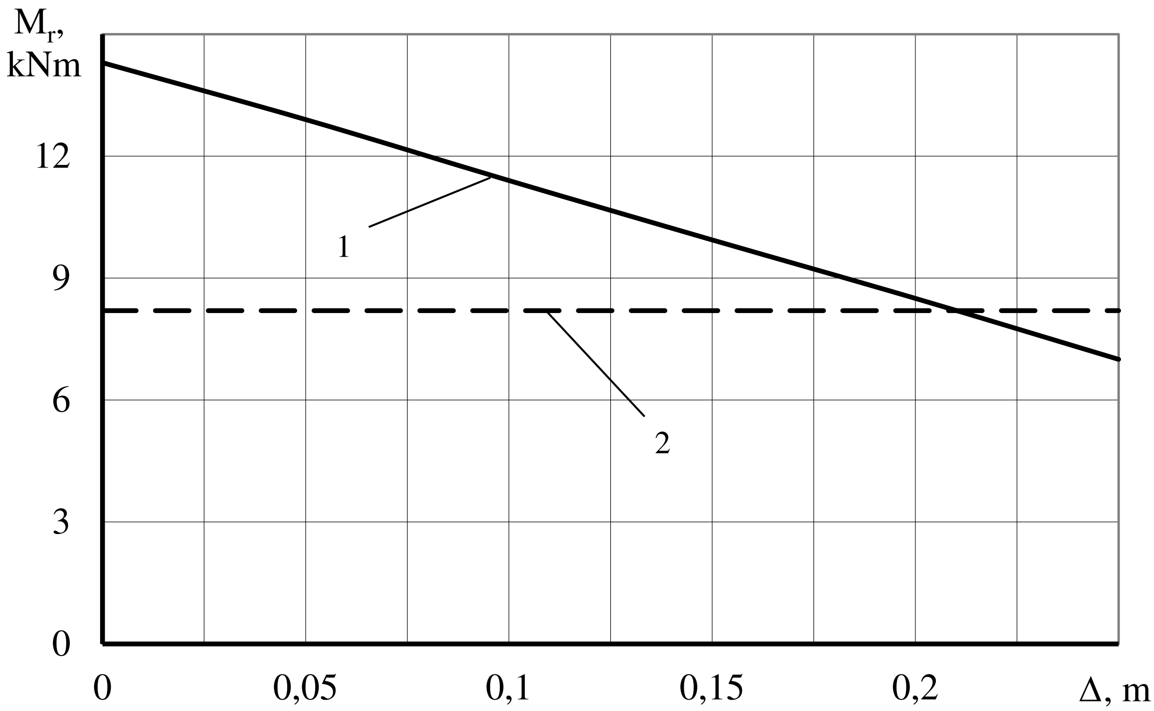


Fig. 1.6 – Dependence of torque on the value of Δ for two-point (1) and three-point (2) rear linkage of the aggregating tractor: (γ = 6°, β = 6°)

If the actual value B_D of the power vehicle is larger than the desired B_T, the plow is not attached symmetrically, but with a right-side lateral displacement e_p, which is found by this expression:

$$e_p = (B_D - b_T).2^{-1}. \qquad (1.12)$$

With the three-point attachment of the plow with the tractor's rear linkage, this displacement can only be achieved by moving the plow relative to its own link, i.e. the rods (position 2 in Fig. 1.3). This results in an angle γ between the longitudinal axis of symmetry of the plow and the line of traction. The presence of the latter, in turn, causes not only an increase in the torque (which has already been discussed above), but also leads to the lateral (transverse) component of the traction force, which forms an additional frictional force of the field boards against the furrow wall.

With the two-point linkage, the right side lateral displacement of a plowing tool is possible both as described above and by appropriately moving the lower links of the tractor rear linkage by a specified amount (Δ, Fig. 1.3, a).

Since in this case the value of angle γ does not change, the traction resistance of the plow remains almost constant, and the torque it creates tends to decrease (see Fig. 1.6).

Based on the above, we can state the following: If under conditions (1.10) or (1.11) parameters of the running gear of the tractor do not allow to connect the plow symmetrically, it is better to use a two-point hitch scheme of the rear linkage of the power vehicle. In this case, the right side lateral displacement of the plowing tool should be primarily carried out by moving the lower links of the tractor linkage. And only when this displacement proves insufficient, the frame of the plowing tool should be moved relative to its own attachment pins.

If, however, the parameters of the undercarriage of the aggregating tractor allow for symmetrical coupling of the plow, the power vehicle can have only one attachment scheme of the rear linkage – three-point attachment. Especially since, unlike the two-point scheme, there is no need to readjust it when the tractor works with other attached agricultural machines or tools.

Taking into account expression (1.10), expression (1.12) can be presented in a more expanded form:

$$e_p = \frac{B_D + 2A + b - b_k\left(n+1\right)}{2} \qquad (1.13)$$

It follows from the analysis of expression (1.13) that the value of transverse displacement of the “center of resistance” of the plow depends on the distance A between the outer edge of the propeller and the furrow wall, which is determined by the stability of the rectilinear motion of the power vehicle. The more resistant the latter is to the effect of external disturbances, the smaller the value A may be, and vice versa.

Literary sources emphasize that, depending on the physical and mechanical properties of the soil and the stability of movement of a particular power vehicle in the plow-tractor unit, the distance A may vary within 10 ... 29 cm [3]. A greater distance makes it more difficult for the tractor operator to track the furrow edge of the previous machine pass, which increases the unevenness of the machine's working width. At a lower value of A there is a risk of sliding of the power vehicle by the right thrusters in the furrow, which negatively affects the quality performance of the plow-tractor unit.

That is why for tractor T-150K, for example, when working with plows PLN-5-35 or PLP-6-35, the specified distance A in practice is taken close to the value of the plowing depth. On average, it may be 27 cm, which is close to the recommended value for this distance [3].

Since the width of the body of these plows is $b_k = 35$ cm, and the width of the tractor tire is $b = 54$ cm, then, as follows from expression (1.10), the value of the power vehicle track, must equal:

– aggregated with plow PLN-5-35 ($n = 5$) plow – 102 cm;

– aggregated with plow PLP-6-35 (n = 6) plow – 137 cm.

Since the T-150K tractor has the minimum actual value of the track B_D = 168 cm, then in both variants you need a right-sided lateral displacement of the "center of resistance" of the plowing tool. And if you consider the above analysis, this means that when aggregating with plows, it is desirable to set the rear linkage of the T-150K tractor by the two-point linkage scheme.

Unlike a wheeled tractor, a crawler tractor (e.g., T-150 tractor) is more resistant to the action of the torque during straight-line movement. Aggregated with the same plows that are aggregated with T-150K tractor, it can move, as practice shows, without sliding into a furrow at a distance from its wall to the edge of the track, equal to 10 ... 12 cm. This is explained by the fact that the T-150 tractor tracks have a much larger bearing surface area (mainly due to their length) than those of the wheeled tractor.

If we consider that this power vehicle has the parameter b equal to 43 cm, then for A = 10 cm we will have:

B_T = 147 cm – when working with PLN-5-35,

B_T = 182 cm – when working with PLP-6-35.

Since the T-150 crawler tractor has the actual value of the track equal to B_D = 147 cm, then in the first variant we get symmetric attachment of the plowing tool, and in the second – aggregation with left-side transverse displacement of its "center of resistance" by 17,5 cm.

Hence, a caterpillar tractor type T-150 can have a three-point rear attachment for a plow.

1.2. Mathematical model of a plowing unit on the basis of a conventional tractor

1.2.1. General provisions and assumptions made when modelling the movement of a plowing unit in the horizontal plane

The functioning of a plowing unit can be considered as a response to input controlling and disturbing influences. The plowing machine-tractor unit's response to controlling influences is characterized by its controllability, and to disturbing influences – by its stability of motion.

The above approach to the scheme of functioning of a plowing-tractor unit defines it as a dynamic system that converts input variables into output variables in the process of movement. Since the main external factor causing the course deviation of the implement tractor is the angle β between the longitudinal axes of symmetry of the power vehicle and the plow, it is advisable to consider it as an input perturbation.

The input control action is the rotation angle α of the aggregating tractor steering wheels.

The output variables are parameters that set the trajectory of the plowing-tractor unit: tractor course angle φ and the lateral displacement X_{S_T} of its centre mass.

The nature of operation of such a dynamic system during the development of controlling and disturbing influences depends on its dynamic characteristics, which (in this case) are determined by the parameters of the plowing aggregate. Therefore, the correct choice of these parameters, based on an assessment of their impact on the controllability and stability of the plowing machine-tractor unit movement, means the establishment of such dynamic characteristics of the system that provide it with the optimal transformation of the input disturbances.

The transformational properties of one or another dynamical system can be expressed by some operator. The latter is essentially an algorithm for converting input variables into output ones and is therefore a common and defining characteristic of a dynamic system for synthesizing and optimizing the parameters of a plowing unit.

The basic operators for dynamic systems are transfer functions and frequency characteristics. However, to determine them, a system of appropriate differential expressions linking output variables with input influences is required, that is, a mathematical model of the process under study.

A plowing machine-tractor unit is a rather complex dynamic system. This complexity is explained by its multidimensionality, the high order of the differential expressions, and the presence of nonlinear relationships between its individual coordinates. At this stage of the study of the dynamics of the unit in question, it is advisable to consider its somewhat simplified scheme in the form of a linear model. A.B. Lurie, PhD emphasized that "... such idealization of the system in many cases is quite effective for complex agricultural units and their control systems, the dynamics of which are not yet studied sufficiently. Linearization of the dynamic system gives an opportunity to physically comprehend the obtained result and to accumulate design experience" [4].

When solving synthesis problems, many researchers use amplitude-frequency and phase-frequency characteristics as the operator. Given that the controllability and stability of motion of the plowing-tractor unit is supposed to be estimated by the accuracy and speed of its performance of controlling and disturbing influences, it is necessary to determine the amplitude-frequency and phase-frequency characteristics of both influences relative to the parameters that ensure satisfactory tracking of the power vehicle trajectory of the previous passage of the plowing – tractor unit.

Such parameters can be the lateral displacement of the tractor's steering wheels and its course angle.

It should be emphasized that even in the linear interpretation, the mathematical model of a plowing unit is a system of complex differential equations. In order to simplify them, we accept the following assumptions:

– the surface of the field is horizontal, and there will be no tilt and trim of the machine;

– the tractor is considered as a solid body with a longitudinal plane of symmetry passing through its center of mass;

– fluctuations in the plow's traction resistance have no significant effect on the forward speed of the plow, so it is assumed to be constant;

– lateral interaction of the pneumatic tires of the aggregating tractor wheels with the surface of the agricultural background is considered within the framework of the “lateral pulling away” hypothesis;

– due to insignificant values, gyroscopic and stabilizing moments of the pneumatic tires of the aggregate tractor and moments of their torsion, relative to the vertical axis are not taken into account;

– the angles of departure of the pneumatic tires of the tractor wheels located on one geometrical axis, as well as the lateral forces acting on them, will be considered to be small;

– the steering angles of the steering wheels of the aggregate tractor will be sufficiently small, and therefore equal to each other.

Let us justify each of these assumptions. Thus, the validity of the first assumption can be explained by the levelling of the macro-relief of the fields on most of the arable land in Europe.

The second assumption is based on the fact that aggregate tractors have a frame structure that is inherently stiff and rigid. At the same time for the majority of energy vehicles the transverse deviation of the coordinate of their center of mass from the longitudinal axis of symmetry does not exceed 2,5 cm.

The coefficient of variation V is quite often used as an indicator of the variability of a particular parameter. Variability is considered insignificant if V does not exceed 10 %, average – if it is higher than 10 %, but less than 20 %, and significant – at 20 % or more. During many years of experimental research, we have found that with almost negligible variability in the traction resistance of the plow, the coefficient of variation of forward speed of plowing-tractor unit does not exceed 6%. The relative error in determining the average velocity is less than 1 %. Thus, the third assumption in this case is quite correct.

To represent the lateral interaction of a particular thruster with the deformed surface, the "lateral withdrawal" hypothesis is most commonly used, both in its linear and nonlinear interpretations. In this case, to determine the lateral horizontal reactions at the contact points of the running wheels of the tractor aggregate with the rolling surface (lateral forces) the coefficients of resistance to lateral withdrawal of pneumatic wheel tires are used. It is well known that the application of the nonlinear interpretation of this hypothesis is justified only in the study of the phenomenon of tire retraction on hard surfaces (asphalt, concrete, etc.). When driving on a soft background, which prevails during operation of a plowing unit machine, the hypothesis of "lateral drift" in a linear interpretation gives quite satisfactory results and is a sufficient basis for adopting the fourth assumption.

The validity of the fifth assumption is justified by the low angular velocity of rotation in the horizontal plane of all links of the plowing unit (including the steering wheels of the tractor), as well as by the relatively high hardness of the power vehicle tires when they twist relative to the vertical axis.

The sixth assumption stems from the formulation of research tasks, providing for the analysis of the movement of a plowing unit along the trajectory of small curvature and small angular oscillations of the links (tractor and plow) in the transverse plane. In addition, the vertical load acting on the tire plays a major role in many of the parameters that affect the angle of retraction of a particular tire. The operating weight of wheeled tractors of the HTZ-120/160 family, for example, is distributed on their left and right sides approximately equally (the difference does not exceed 7 %). Hence, it is reasonable to assume that during the working motion in operation, the angles of retraction of tires located on the same geometric axis are almost the same. Since the lateral forces acting on these tires are also equal (according to the theory of "sideways departure"), each axle of the aggregating tractor can be conventionally represented by an "equivalent" wheel. This assumption is quite widely used in similar studies.

The validity of the seventh assumption is justified by the fact that during the movement of a plowing unit in operation, the angle of rotation of the steered wheels of the power vehicle, as shown by experimental studies, does not exceed 0,175 rad (10°). The kinematics of the steering trapezium of wheeled tractors of the classical layout is such that the difference in the angles of rotation of their left and right steering wheels can be practically neglected in this case.

1.2.2. Differential equations of motion of a tractor with a classic layout in the horizontal plane

To compose the differential equations of motion of the tractor it is necessary, first of all, to develop its mathematical model. Let us build an equivalent scheme of plane-parallel motion of a wheeled tractor of classical layout. Given the assumptions outlined in the previous paragraph, we assume that the tractor in operation performs translational uniform motion at speed V_o relative to a stationary plane $X_1O_1Y_1$ (Fig. 1.7).

During the technological process of plowing, under the influence of random factors, it deviates from its original position and gets additional speed – the relative motion of the power vehicle in the plane XOY begins. In this case, the plane $X_TS_TY_T$ connected with the center of mass of the tractor, rotates in the plane XOY around a vertical axis S_TZ, which passes through point S_T.

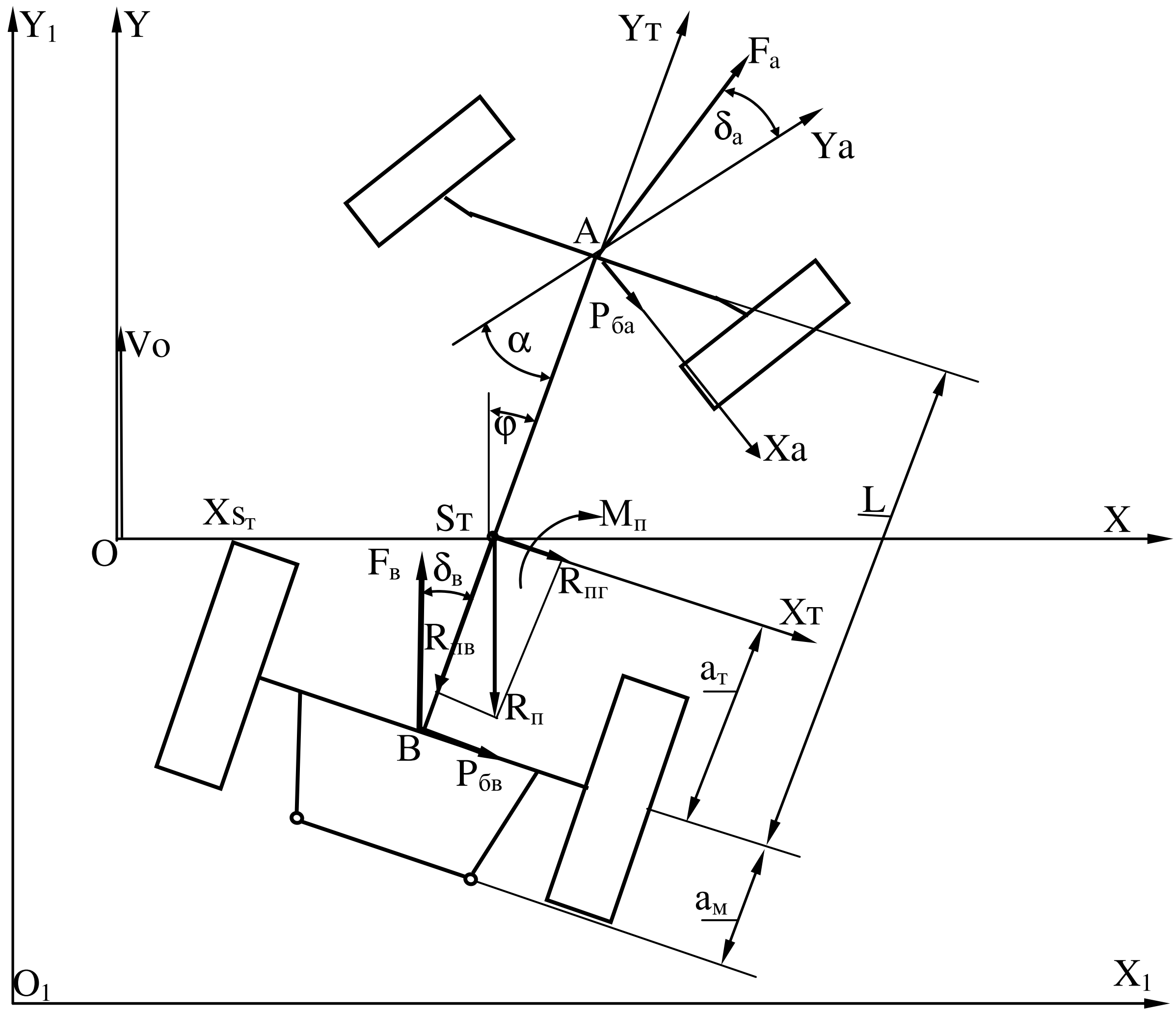


Fig. 1.7 – Diagram of forces acting on the tractor in the horizontal plane

The measure of this rotation is the angle φ, formed by the longitudinal axis of symmetry S_TY_T of a power vehicle with an axis OY.

During the relative motion of this tractor its center of mass (point S_T) moves along the axis OX, which is characterized by the change of abscissa $X_{ST} = O_{ST}$.

Thus, the tractor has two degrees of freedom in relation to the plane XOY, to which correspond two generalized coordinates: angle φ and abscissa X_{ST}.

We will consider the forces applied to the tractor. The external forces applied to the power vehicle during its plane-parallel motion include (Fig. 1.7):

– driving force $\overline{F}_a$ of the front wheels, which is exerted in point A and forms an input angle δ_a with the longitudinal axis Y_a;

– driving force $\overline{F}_b$ of the rear wheels, which is exerted in point B and forms an input angle δ_b with the longitudinal axis of symmetry of the tractor;

– lateral forces $\overline{P}_{la}$ and $\overline{P}_{lb}$, exerted in points A and B, respectively;

– main vector $\overline{R}_{\Pi}$ (consisting of $\overline{R}_{pg}$ and $\overline{R}_{pv}$), adjusted to the tractor's center of mass (point S_T) and the main moment M_P of the forces, acting on the plow.

The kinetic power of the tractor T_T related to plane XOY can be obtained with the following expression:

$$T_T = \frac{M_T.V_{S_T}^2 + J_{S_T}.\omega_T^2}{2},$$

where M_T – tractor mass;

V_{S_T} – linear velocity of the tractor's center of mass in the plane XOY;

J_{S_T} – moment of inertia of the tractor relative to the vertical axis S_TZ;

ω_T – angular velocity of the tractor turning around the axis S_TZ.

Concerning the modules of the specified linear and angular velocities we can write:

$$V_{S_T} = \dot{X}_{S_T}; \quad \omega_T = \dot{\varphi}.$$

Taking this into account we get the value of the kinetic power of the tractor:

$$T_T = \frac{M_T.\dot{X}_{S_T}^2 + J_{S_T}.\dot{\varphi}^2}{2}.$$

Differential equations of tractor oscillations at its plane-parallel motion will be made in the Lagrange form of the 2nd kind:

$$\frac{d}{dt}\left(\frac{\partial T_T}{\partial \dot{q}_i}\right) - \frac{\partial T_T}{\partial q_i} = Q_i, \tag{1.14}$$

where q_i, Q_i – generalized coordinates and forces, respectively, ($i = \overline{1,2}$);

Since the kinetic power depends only on the velocity and does not depend on the generalized coordinate, then:

$$\frac{\partial T_T}{\partial q_i} = 0,\ i = 1,\ 2\ .$$

The partial derivatives of the velocities of the generalized coordinates will be:

$$\frac{\partial T_T}{\partial \dot{X}_{S_T}} = M_T.\dot{X}_{S_T},$$

$$\frac{\partial T_T}{\partial \dot{\varphi}} = J_{S_T}.\dot{\varphi}\ .$$

The partial derivatives over time will be:

$$\frac{\partial}{\partial t}\left(\frac{\partial T_T}{\partial \dot{X}_{S_T}}\right) = M_T.\ddot{X}_{S_T},$$

$$\frac{\partial}{\partial t}\left(\frac{\partial T_T}{\partial \dot{\varphi}}\right) = J_{S_T}.\ddot{\varphi}.$$

Since in this case the dynamic system has two degrees of freedom, and therefore, two generalized coordinates φ and X_{ST}, expression (1.14) will be represented as a system of two differential equations of the following form:

$$\left.\begin{aligned} & M_T.\ddot{X}_{S_T} = Q_{X_T}, \\ & J_{S_T}.\ddot{\varphi} = Q_{\varphi}. \end{aligned}\right\} \tag{1.15}$$

To determine the generalized force Q_{X_T} let us give a boost ΔX_{S_T} corresponding to the generalized coordinate. The dynamic system will move to the right by the value of ΔX_{S_T}. During the translational motion of a solid body, the work is performed by the main vector of forces. The generalized force is equal to the projection of the main vector of external forces applied to the tractor in the direction of the axis OX:

$$\begin{aligned} Q_{X_T} = {} & P_{la}.\cos(\varphi+\alpha) + F_a.\sin(\varphi+\alpha-\delta_a) + R_{\Pi_\Gamma}.\cos\varphi + \\ & + F_b \sin(\varphi-\delta_b) - R_{\Pi_B}.\sin\varphi + P_{lb}.\cos\varphi \end{aligned},$$

where α – rotation angle of the steering wheels of the aggregating tractor.

Since for small angles the value of their cosine is approximately equal to one, and the sine is equal to the angle itself, the value of the generalized force Q_{X_T} will be equal to:

$$Q_{X_T} = P_{la} + P_{lb} + F_a.(\varphi+\alpha-\delta_a) + F_b.(\varphi-\delta_b) + R_{\Pi_\Gamma} - R_{\Pi_B}.\varphi,$$

or

$$Q_{X_T} = P_{la} + P_{lb} + F_a.(\alpha-\delta_a) - F_b.\delta_b + R_{\Pi_\Gamma} + (F_a + F_b - R_{\Pi_B}).\varphi.$$

Considering that $F_a + F_b - R_{\Pi_B} = 0$, we end up with:

$$Q_{X_T} = P_{la} + P_{lb} + F_a.(\alpha-\delta_a) - F_b.\delta_b + R_{\Pi_\Gamma}. \tag{1.16}$$

To obtain the generalized force Q_φ we turn the tractor, as a solid body, relative to the axis S_TZ at an angle φ. The desired generalized force Q_φ will be equal to the main moment M_Π of all forces relative to point S_T:

$$Q_\varphi = P_{la}(L-a_T).\cos\alpha + F_a(L-a_T).\sin(\alpha-\delta_a) - P_{lb}.a_T + F_b.a_T.\sin\delta_b + M_\Pi,$$

where L, a_T – design parameters, the nature of which is clear from Fig. 1.7.

Given the small value of the angles α and β we finally get:

$$Q_\varphi = P_{la}(L-a_T) + F_a(L-a_T)(\alpha-\delta_a) - P_{lb}.a_T + F_b.a_T.\delta_b + M_\Pi. \tag{1.17}$$

According to the "lateral withdrawal" hypothesis, the lateral forces P_{la} and P_{lb} can be determined by the following well-known formulas:

$$P_{la} = k_a.\delta_a; \quad P_{lb} = k_b.\delta_b,$$

where k_a and k_b – resistance coefficients of front and rear pneumatic tractor tires, respectively, kN·(rad)$^{-1}$;

δ_a and δ_b – the tire angles of the front and rear tractor wheels, respectively.

To obtain the angles of departure δ_a and δ_b let us first determine the components of the velocity vectors of points A and B, which correspond to the middle projections of the front and rear tractor axles on the horizontal plane (see Fig. 1.7). We obtain:

$$\overline{V}_A = \overline{V}_O + \overline{V}_{S_T} + \overline{V}_{AS_T}; \; \overline{V}_B = \overline{V}_O + \overline{V}_{S_T} + \overline{V}_{BS_T}. \tag{1.18}$$

According to the module, the components of the expressions (1.18) are:

$$\left|\overline{V}_O\right| = V_O; \qquad \left|\overline{V}_{S_T}\right| = \dot{X}_{S_T}; \qquad \left|\overline{V}_{AS_T}\right| = (L-a_T).\dot{\varphi}; \qquad \left|\overline{V}_{BS_T}\right| = a_T.\dot{\varphi}. \tag{1.19}$$

The geometrical sums of expressions (1.18) are shown in Fig. 1.8 and Fig. 1.9, respectively. The angle of retraction δ_a of the tractor front wheels can be obtained, using this well-known equation:

$$\tan\delta_a \approx \delta_a = \frac{V_{XA}}{V_{YA}},$$

where V_{XA} and V_{YA} – the projection of the absolute velocity vector $\bar{V}_A$ on point A of axes AX_A and AY_A (see Fig. 1.7).

From the diagram of Fig. 1.8, taking into account the analytical relationships (1.19) and the small size of the angles α, φ and δ_a it follows that:

$$V_{XA} = -V_O.(\varphi + \alpha) + \dot{X}_{S_T} + (L - a_T).\dot{\varphi}, \tag{1.20}$$

$$V_{YA} = V_O + \dot{X}_{S_T}.(\alpha + \varphi) - (L - a_T).\dot{\varphi}.\alpha .$$

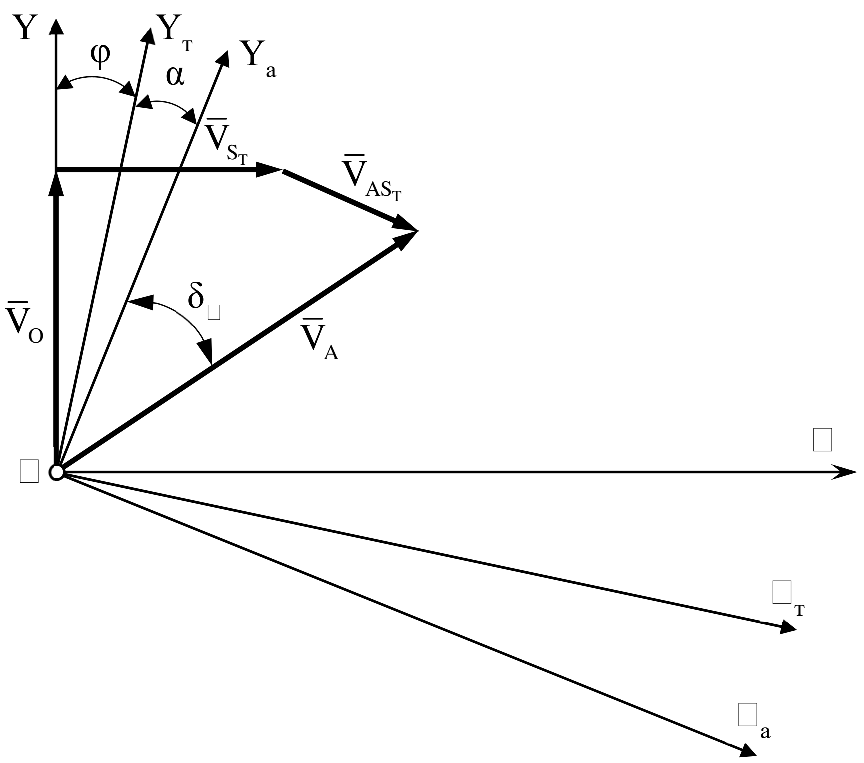


Fig. 1.8 – Vector sum of the linear velocity of the tractor front axle centre (point A, Fig. 1.7)

Since the product of small values gives even smaller results, the last two components of the second equation of expressions (1.20) can be neglected. That is, we can consider that $V_{YA} \approx V_O$. Then as a result of this we will have:

$$\tan\delta_a \approx \delta_a = \frac{V_{XA}}{V_{YA}} = \frac{\dot{X}_{S_T} + (L - a_T).\dot{\varphi}}{V_O} - \varphi - \alpha .$$

From the diagram of Fig. 1.8, taking into account the analytical relationships (1.19) and the small size of the angles α, φ and δ_a it follows that:

$$V_{XA} = -V_O.(\varphi + \alpha) + \dot{X}_{S_T} + (L - a_T).\dot{\varphi}, \tag{1.20}$$

$$V_{YA} = V_O + \dot{X}_{S_T}.(\alpha + \varphi) - (L - a_T).\dot{\varphi}.\alpha .$$

Since the product of small values gives even smaller results, the last two components of the second equation of expressions (1.20) can be neglected. That is, we can consider that $V_{YA} \approx V_O$. Then as a result of this we will have:

$$\tan\delta_a \approx \delta_a = \frac{V_{XA}}{V_{YA}} = \frac{\dot{X}_{S_T} + (L - a_T).\dot{\varphi}}{V_O} - \varphi - \alpha .$$

We can also obtain the angle of retraction δ_b of the rear tractor thrusters by projecting the vector of absolute velocity $\bar{V}_B$ in the centre of the rear axes AX_A and AY_A (Fig. 1.9). Taking into account the analytical relationships (1.19) and the small size of the angles α, φ and δ_b the required projections will be equal to:

$$V_{XB} = -V_O.\varphi + \dot{X}_{S_T} - a_T.\dot{\varphi},$$

$$V_{YB} = V_O + \dot{X}_{S_T}.\varphi + a_T.\dot{\varphi}.\alpha \approx V_O$$

As a result, we can write an equation for finding the angle of retraction δ_b of the tractor rear wheels:

$$\tan\delta_b \approx \delta_b = \frac{V_{XB}}{V_{YB}} = \frac{\dot{X}_{S_T} - a_T.\dot{\varphi}}{V_O} - \varphi .$$

By finding the angles of retraction δ_a and δ_b and taking into account the negative direction of their deposition, we can express the lateral forces P_{la} and P_{lb}. We have:

$$P_{la} = k_a.\left[\frac{-X_{S_T} - (L - a_T).\dot{\varphi}}{V_O} + \varphi + \alpha\right], \tag{1.21}$$

$$P_{lb} = k_b.\left[\frac{-X_{S_T} + a_T.\dot{\varphi}}{V_O} + \varphi\right].$$

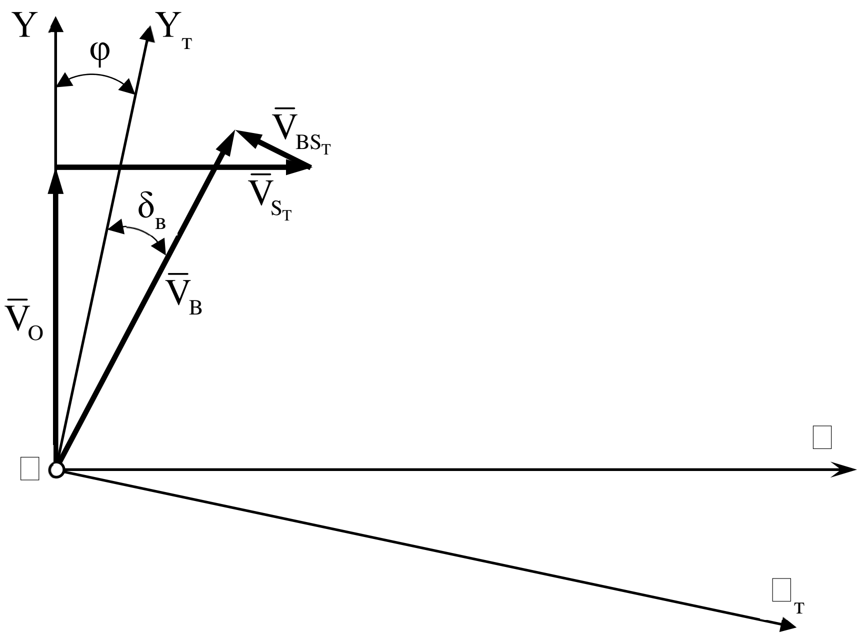


Fig. 1.9 – Vector sum of the linear velocity of the tractor rear axle center (point B, Fig. 1.7)

Taking into account expression (1.21) and substituting the values of generalized forces, according to expressions (1.16) and (1.17), into system (1.15), we obtain the following system of differential equations of tractor motion in the horizontal plane:

$$\left.\begin{aligned} &A_{11}.\ddot{X}_{S_T} + A_{12}.\dot{X}_{S_T} + A_{13}.\dot{\varphi} + A_{14}.\varphi = f_{11}.\alpha + f_{12}, \\ &A_{21}.\ddot{\varphi} + A_{22}.\dot{\varphi} + A_{23}.\varphi + A_{24}.\dot{X}_{S_T} = f_{21}.\alpha + f_{22}, \end{aligned}\right\} \tag{1.22}$$

where $A_{11} = M_T$,

$$A_{12} = \frac{k_a + k_b - F_a - F_b}{V_O},$$

$$A_{13} = \frac{(k_a - F_a).(L - a_T) + (F_b - k_b).a_T}{V_O},$$

$$A_{14} = F_a + F_b - k_a - k_b,$$

$$A_{21} = J_{S_T},$$

$$A_{22} = \frac{(k_a - F_a).(L - a_T)^2 + (k_b - F_b).a_T^2}{V_O},$$

$$A_{23} = -A_{13}.V_O,$$

$$A_{24} = A_{13},$$

$$f_{11} = k_a,$$

$$f_{12} = R_{P_d}$$

$$f_{21} = (L - a_T).k_a,$$

$$f_{22} = M_P.$$

The values of force $R_{П_Г}$ and moment $M_П$, of the system of expressions (1.22) can be obtained by looking at Fig. 1.3 and Fig. 1.7. From them, taking into account the expressions (1.1), (1.3) and (1.4) and the small size of the angles γ and β, we can see that:

– for the two-point rear linkage setting diagram of the tractor:

$$P_{Pd_r} = P_{kr}.(\gamma \pm \beta),$$

$$M_P = -P_{kr}.(a_T + a_M).(\gamma \pm \beta);$$

– for the three-point rear linkage setting diagram of the tractor:

$$P_{Pd} = P_{kr}.\left[\gamma \pm \left(1 - \frac{l_T}{r}\right)\beta\right],$$

$$M_P = P_{kr}.(C_o + C).\left[\gamma \pm \left(1 - \frac{l_T}{r}\right).\beta\right].$$

Ultimately, the mathematical model of a plowing machine-tractor unit based on a wheeled power vehicle of the classical layout in differential form is as follows:

$$\left.\begin{aligned} &A_{11}.\ddot{X}_{S_T} + A_{12}.\dot{X}_{S_T} + A_{13}.\dot{\varphi} + A_{14}.\varphi = f_{11}.\alpha + f_{12}.\beta + f_{13}, \\ &A_{21}.\ddot{\varphi} + A_{22}.\dot{\varphi} + A_{23}.\varphi + A_{24}.\dot{X}_{S_T} = f_{21}.\alpha + f_{22}.\beta + f_{23}, \end{aligned}\right\} \qquad (1.23)$$

where $A_{11} = M_T$,

$$A_{12} = \frac{k_a + k_b - F_a - F_b}{V_O},$$

$$A_{13} = \frac{(k_a - F_a).(L - a_T) + (F_b - k_b).a_T}{V_O},$$

$$A_{14} = F_a + F_b - k_a - k_b,$$

$$A_{21} = J_{S_T},$$

$$A_{22} = \frac{(k_a - F_a).(L - a_T)^2 + (k_b - F_b).a_T^2}{V_O},$$

$$A_{23} = -A_{13}.V_O,$$

$$A_{24} = A_{13},$$

$$f_{11} = k_a,$$

$f_{12} = P_{kr}$, – for the two-point rear linkage adjustment system of the tractor;

$f_{12} = P_{kr}.\left(1 - \frac{l_T}{r}\right)$, – for the three-point rear linkage adjustment system of the tractor,

$$f_{13} = P_{kr}.\gamma,$$

$$f_{21} = (L - a_T).k_a,$$

$f_{22} = -P_{kr}.a_T + a_M$, – for a two-point linkage system,

$f_{22} = P_{kr}.(C_o + C).\left(1 - \frac{l_T}{r}\right)$, – for a three-point linkage system,

$f_{23} = -P_{kr}.(a_T + a_M).\gamma$, – for a two-point linkage system,

$f_{23} = -P_{kr}.(C_o + C).\gamma$, – for a three-point linkage system.

By applying the substitution $\frac{d}{dt} = s$, we can obtain the mathematical model of the plowing aggregate in operation form written as follows:

$$\left.\begin{aligned} K_{11}.X_{S_T}(s) + K_{12}.\varphi(s) = F_{11}.\alpha(s) + F_{12}.\beta(s) + F_{13}.l(s), \\ K_{21}.X_{S_T}(s) + K_{22}.\varphi(s) = F_{21}.\alpha(s) + F_{22}.\beta(s) + F_{23}.l(s), \end{aligned}\right\} \tag{1.24}$$

where $K_{11} = A_{11}.s^2 + A_{12}.s$, $\quad K_{12} = A_{13}.s + A_{14}$,

$$K_{21} = A_{24}.s\,, \qquad K_{22} = A_{21}.s^2 + A_{22}.s + A_{23}\,,$$

$$F_{11} = f_{11}, \qquad F_{12} = f_{12}\,,$$

$$F_{13} = f_{13}\,, \qquad F_{21} = f_{21}\,,$$

$$F_{22} = f_{22}\,, \qquad F_{23} = f_{23}\,.$$

In the mathematical model (1.24) the input quantities are the angle α of the tractor steering wheels (steering action) and the angle) α and angle β of the plow rotation in the horizontal plane (disturbing influence). All parameters included in the coefficients F_{13} and F_{23} act as staggered effects.

The output quantities in this mathematical model are the transverse displacement X_{S_T} center of mass of the tractor and its course angle φ.

1.3. Differential equations of tractor motion with articulated frame

In contrast to the tractor with the classical configuration, the power vehicle with articulated frame controls the direction of movement by turning its half-frames by a predetermined angle α (Fig. 1.10). Since in rectilinear motion the value of this angle is relatively small (up to 8°), when compiling the differential equations of motion, we will immediately take into account that $\sin\alpha \approx \alpha$, a $\cos\alpha \approx 1$. In this case, we accept the same assumptions that were outlined and justified in paragraph 1.2.1.

For the plowing machine under consideration, the "center of mass" of the tractor is brought to the centre of the axle of the rear half-frame (point S_T or B, Fig. 1.10). The expressions for the generalized forces Q_{X_T} and Q_φ, which specify, respectively, the linear displacements X_{S_T} and the course angle φ, have the following form:

$$Q_{X_T} = P_{lb}.\cos\varphi + R_{\Pi_\Gamma}.\cos\varphi + F_b.\sin(\varphi - \delta_b) - R_{\Pi_B}.\sin\varphi +$$
$$+P_{la}.\cos(\varphi + a) + F_a.\sin(\alpha - \delta_a + \varphi),$$

$$Q_\varphi = P_{la}.\cos\alpha.(l.\cos\alpha + l) + F_a.\sin(\alpha - \delta_a).(l.\cos\alpha + l) + M_\Pi,$$

or

$$Q_{X_T} \approx P_{lb} + R_{\Pi_\Gamma} + F_b.\varphi - F_b.\delta_b - R_{\Pi_B}.\varphi + P_{la} + F_a.\alpha - F_a.\delta_a + F_a.\varphi,$$

$$Q_\varphi = P_{la}.2l + F_a.(\alpha - \delta_a).2l + M_\Pi,$$

or

$$Q_{X_T} \approx R_{\Pi_\Gamma} + (F_b + F_a - R_{\Pi_B}).\varphi + k_b.\delta_b - F_b.\delta_b + k_a.\delta_a - F_a.\delta_a + F_a.\alpha$$

$$Q_\varphi = k_a.\delta_a.L + F_a.\alpha.L - F_a.\delta_a.L + M_\Pi.$$

Taking into account that $F_b + F_a - R_{\Pi_B} = 0$, we will end up with:

$$Q_{X_T} = (k_a - F_a).\delta_a + (k_b - F_b).\delta_b + F_a.\alpha + R_{\Pi_\Gamma}, \tag{1.25}$$

$$Q_\varphi = \delta_a.(k_a - F_a).L + F_a.L.\alpha + M_\Pi, \tag{1.26}$$

where $L = l + l = 2l$ – the base of the tractor.

To find the angles of retraction δ_a and δ_b for a given power vehicle, we will also determine the components of the velocity vectors of points *A* and *B*, which correspond to the projections of the tractor front and rear axle centers on the horizontal plane (see Fig. 1.10). We have:

$$\overline{V}_A = \overline{V}_O + \overline{V}_{S_T} + \overline{V}_{AS_T} + V_{A\pi}; \qquad \overline{V}_B = \overline{V}_O + \overline{V}_{S_T}. \tag{1.27}$$

According to the module, the components in both expressions (1.27), will be equal:

$$\left|\bar{V}_O\right| = V_O\,; \qquad \left|\bar{V}_{S_T}\right| = \dot{X}_{S_T}\,; \qquad \left|\bar{V}_{AS_T}\right| = L.\dot{\varphi}\,; \qquad \left|\bar{V}_{A\pi}\right| = l.\dot{\alpha}\,. \tag{1.28}$$

The geometric sums of expressions (1.27) are shown in Fig. 1.11 and Fig. 1.12, respectively. The angle of retraction δ_a of the tractor's front wheel steering can be obtained by the following expression:

$$\tan\delta_a \approx \delta_a = \frac{V_{XA}}{V_{YA}},$$

where V_{XA} , V_{YA} – projections of the absolute velocity vector $\bar{V}_A$ in point A of axes AX_a and AY_a, respectively (see Fig. 1.11).

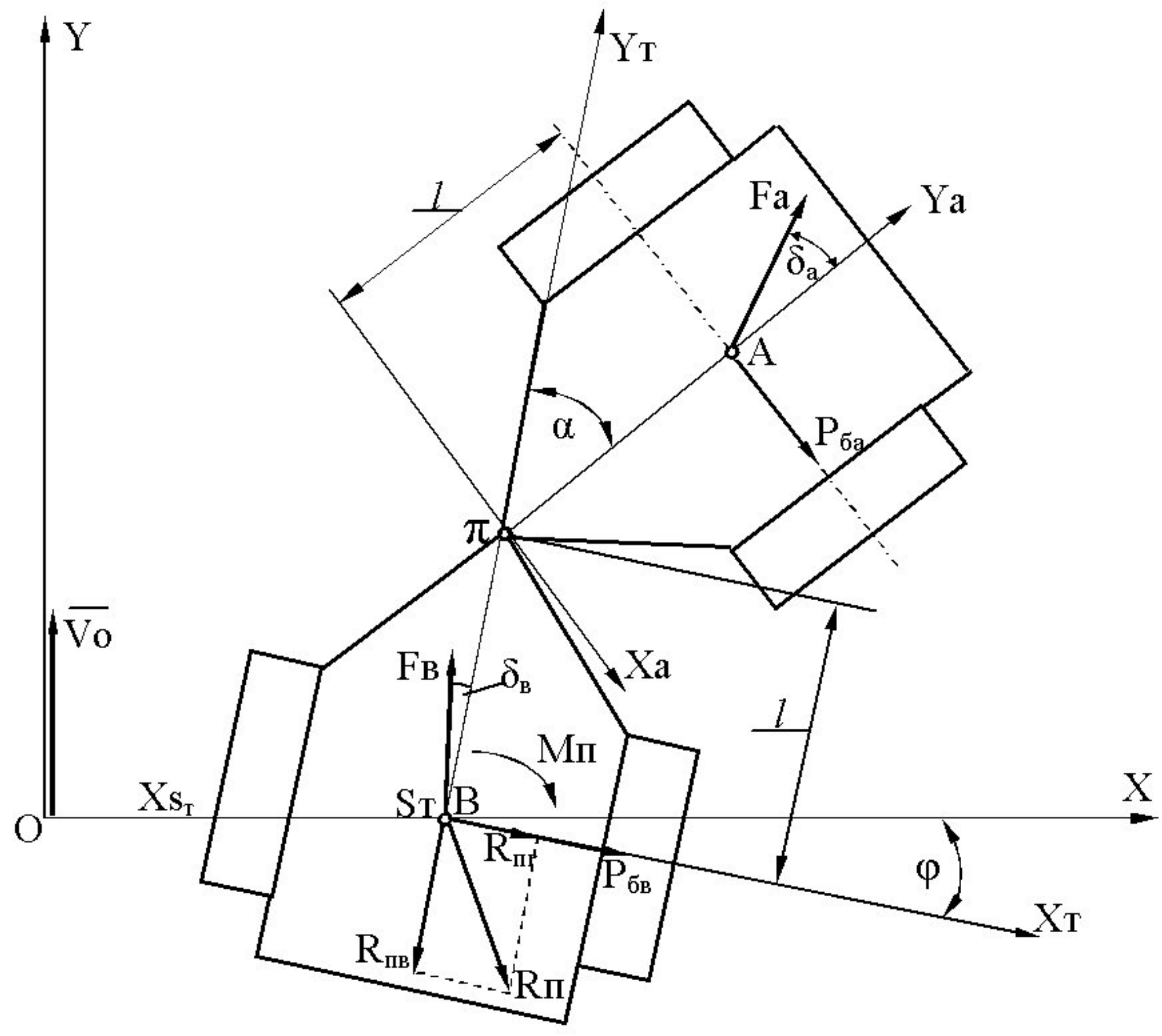


Fig. 1.10 – Diagram of forces acting in the horizontal plane
on a tractor with articulated frame

By projecting the velocity vectors shown in Fig. 1.11 onto the axis AX_a we have:

$$V_{XA} = -V_O.\sin(\varphi + \alpha) + \dot{X}_{S_T}.\cos\varphi + L.\dot{\varphi}.\cos\alpha + l.\dot{\alpha} \ ,$$

or, taking into consideration the small size of angles φ and α, we will have:

$$V_{XA} = -V_O.(\varphi + \alpha) + \dot{X}_{S_T} + L.\dot{\varphi} + l.\dot{\alpha} \ .$$

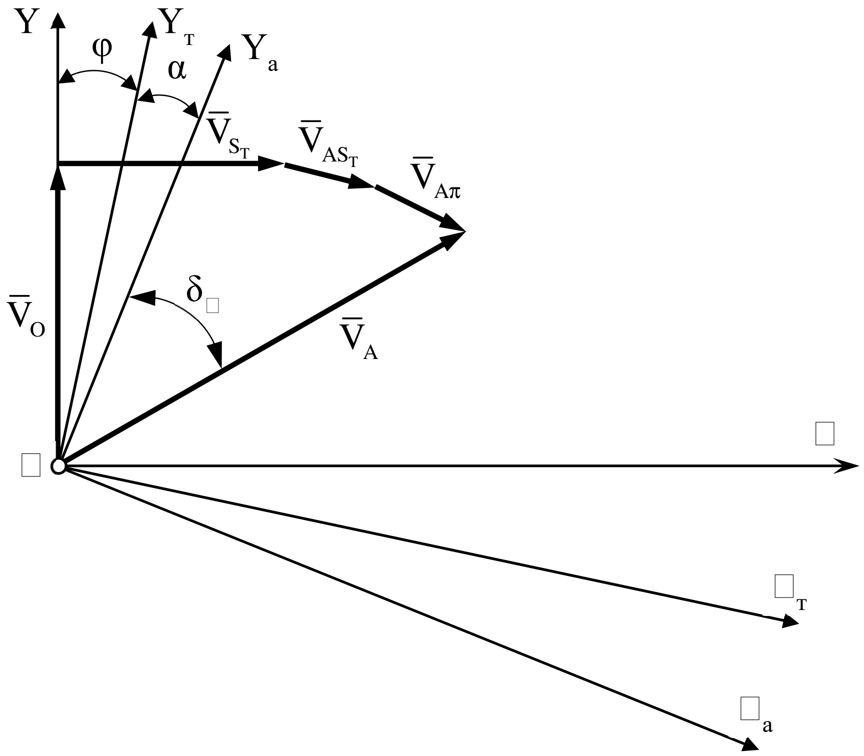


Fig. 1.11 – Vector sum of the linear velocity of the tractor front axle centre (point *A*, Fig. 1.10)

Similarly, projecting the above velocity vectors on the axis AY_a, we will have:

$$V_{YA} = V_O.\cos(\varphi + \alpha) + \dot{X}_{S_T}.\sin\varphi + L.\dot{\varphi}.\sin\alpha \ ,$$

or, taking into consideration the small size of angles φ and α, we will have:

$$V_{YA} = V_O + \dot{X}_{S_T}.\varphi + L.\dot{\varphi}.\alpha \ .$$

Since the product of small quantities gives a value of an even higher order of smallness, the last two terms in the last obtained expression can be neglected. Then finally we will have an expression for the projection of velocities V_{YA} of axle AY_a in this simplified way:

$$V_{YA} \approx V_O .$$

The obtained values of velocity projections can be substituted into the equation for determining tabδ_a, and obtain the value of the angle of retraction δ_a, respectively. We have:

$$\tan\delta_a \approx \delta_a = \frac{V_{XA}}{V_{YA}} = \frac{-(\dot{X}_{S_T} + L.\dot{\varphi} + l.\dot{\alpha})}{V_O} + \varphi + \alpha . \qquad (1.29)$$

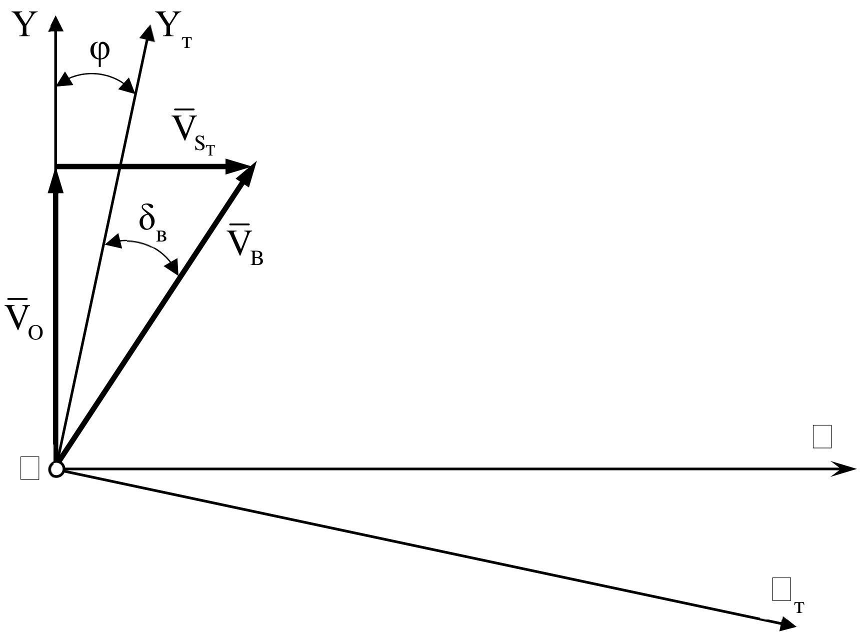


Fig. 1.12 – Vector sum of the linear velocity of the tractor rear axle centre (point B, Fig. 1.10)

The angle of retraction δ_b of the rear tractor thrusters can be obtained by projecting the absolute velocity vector $\overline{V}_B$ in the centre of the rear tractor axle of AY_T and AX_T (Fig. 1.12). The required projections will be equal to:

$$V_{XB} = -V_O.\sin\varphi + \dot{X}_{S_T}.\cos\varphi\,,$$

and

$$V_{YB} = V_O.\cos\varphi + \dot{X}_{S_T}.\sin\varphi\,,$$

or, taking into consideration the extremely small values of angle φ, we have:

$$V_{XB} = -V_O.\varphi + \dot{X}_{S_T}\,, \qquad V_{YB} = V_O + \dot{X}_{S_T}.\varphi \approx V_O\,.$$

Given these expressions, it is also possible to substitute them in a similar expression to determine $\tan\delta_b$, and obtain the angle of retraction δ_b, respectively. As a result, we can write:

$$\tan\delta_b \approx \delta_b = \frac{V_{XB}}{V_{YB}} = \varphi - \frac{\dot{X}_{S_T}}{V_O}\,. \tag{1.30}$$

Substituting the values of the forces according to the expressions (1.25) and (1.26) and the angles of retraction, according to expressions (1.29) and (1.30) in the system (1.15), taking into account the previously given values of the force R_{Pg} and the torque M_P, we will obtain the following differential equations of movement in the horizontal plane of the plowing aggregate on the basis of an articulated frame tractor:

$$\left.\begin{aligned} &A_{11}.X_{S_T} + A_{12}.X_{S_T} + A_{13}.\varphi + A_{14}.\varphi = f_{11}.\alpha + f_{12}.\alpha + f_{13}.\beta + f_{14},\\ &A_{21}.\varphi + A_{22}.\varphi + A_{23}.\varphi + A_{24}.X_{S_T} = f_{21}.\alpha + f_{22}.\alpha + f_{23}.\beta + f_{24}, \end{aligned}\right\} \tag{1.31}$$

where $A_{11} = M_T$,

$$A_{12} = \frac{k_a + k_b - F_a - F_b}{V_O},$$

$$A_{13} = (k_a - F_a).\frac{L}{V_O},$$

$$A_{14} = F_a + F_b - k_a - k_b,$$

$$A_{21} = j_{S_T},$$

$$A_{22} = (k_a - F_a).\frac{L^2}{V_O},$$

$$A_{23} = -A_{13}.V_O,$$

$$A_{24} = A_{13},$$

$$f_{11} = l.\frac{F_a - k_a}{V_O},$$

$$f_{12} = k_a,$$

$f_{13} = P_{kr}$ – for two-point attachment systems,

$f_{13} = P_{kr}.\left(1 - \frac{l_T}{r}\right)$ – for three-point attachment systems,

$$f_{14} = P_{kr}.\gamma,$$

$$f_{21} = L.l.\frac{F_a - k_a}{V_O},$$

$$f_{22} = L.k_a,$$

$f_{23} = -P_{kr}.(a_T + a_M)$ – for two-point attachment systems,

$f_{23} = P_{kr}.(C_o + C).\left(1 - \frac{l_T}{r}\right)$ – for three-point attachment systems,

$f_{24} = P_{kr}.(a_T + a_M).\gamma$ – for two-point attachment systems,

$f_{24} = P_{kr}.(C_o + C).\gamma$ – for three-point attachment systems.

In the operator form of notation, the system (1.31) will take the form:

$$\left.\begin{aligned} K_{11}.X_{S_T}(s) + K_{12}.\varphi(s) = F_{11}.\alpha(s) + F_{12}.\beta(s) + F_{13}.l(s), \\ K_{21}.X_{S_T}(s) + K_{22}.\varphi(s) = F_{21}.\alpha(s) + F_{22}.\beta(s) + F_{23}.l(s), \end{aligned}\right\} \tag{1.32}$$

where $K_{11} = A_{11}.s^2 + A_{12}.s$, $\qquad K_{12} = A_{13}.s + A_{14}$,

$K_{21} = A_{24}.s$, $\qquad K_{22} = A_{21}.s^2 + A_{22}.s + A_{23}$,

$F_{11} = f_{11}.s + f_{12}$, $\qquad F_{12} = f_{13}$,

$F_{13} = f_{14}$, $\qquad F_{21} = f_{21}.s + f_{22}$,

$F_{22} = f_{23}$, $\qquad F_{23} = f_{24}$.

1.4. Analysis of controllability and stability of a plowing machine based on a classic tractor configuration

The controllability of the motion of a plowing machine-tractor unit can be evaluated by means of the corresponding amplitude-frequency and phase-frequency characteristics. The latter are known to be calculated using the necessary transfer function. In this case, it is the function $W^{\alpha}(s)$, which describes the change of course angle φ of the power vehicle with a change of rotation angle α of its steering wheels. There is reason to believe that this function is the ratio of the two determinants:

$$W^{\alpha}(s) = \frac{D^{\alpha}(s)}{D(s)},$$

where $D(s)=\begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix}; D^{\alpha}(s)=\begin{vmatrix} K_{11} & F_{11} \\ K_{21} & F_{21} \end{vmatrix}$.

The coefficients of these determinants are taken from the system of differential equations (1.24).

After exposing the above determinants and performing the appropriate transformations, we obtain:

$$W^{\alpha}(s)=\frac{C_1.s+C_o}{s.(B_2.s^2+B_1.s+B_o)}, \tag{1.33}$$

where $C_1=A_{11}.f_{21}$,

$C_o=A_{12}.f_{21}-A_{24}.f_{11}$,

$B_2=A_{11}.A_{21}$,

$B_1=A_{12}.A_{21}+A_{11}.A_{22}$,

$B_o=A_{12}.A_{22}+A_{11}.A_{23}-A_{13}.A_{24}$.

Calculations of amplitude-frequency and phase-frequency characteristics for the considered dynamic system were performed according to the developed program on a PC at the following values, which are the components of the coefficients of the system of equations (1.32): M_T = 81000 kg; J_{S_T} = 195 kN·m·s^2; a_T = 1,0 m; L = 2,86 m; F_a = 12 kN; F_b = 20 kN; k_a = 140 kN·rad^{-1}; k_b = 200 kN·rad^{-1}.

It should be noted that the dynamical system under study is a tracking system. Therefore, when it works out the control action, the amplitude-frequency response in the operating frequency range, which is 0 … 3 s^{-1} [2], must be equal to one, and the phase-frequency response must be equal to zero [5].

Analysis of mathematical modeling data shows that an increase in speed of a plowing machine-tractor unit leads to an increase in amplitude-frequency response (Fig. 1.13).

At the speed of V_o = 3 m·s^{-1} and oscillation frequency rotation angle α of the tractor's steering wheels that are smaller than 0,75 s^{-1}, there is an over-regulation of the controlling influence in general.

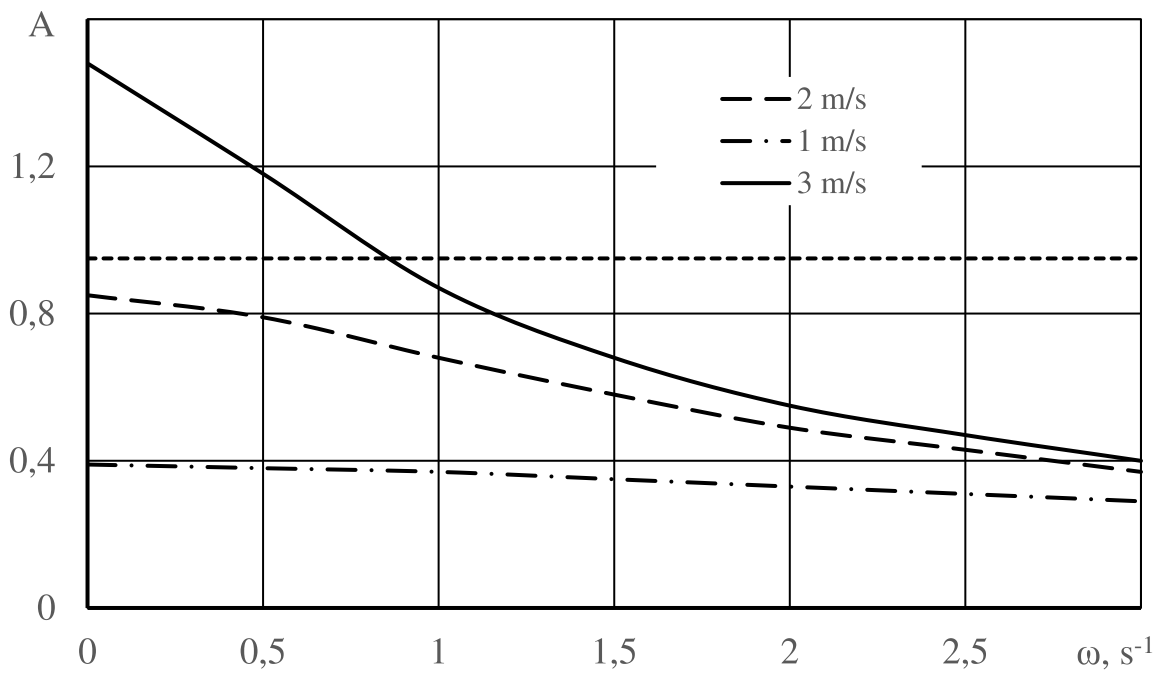


Fig. 1.13 – Amplitude-frequency response of a plowing machine and tractor unit to the control action α at different speeds of working movement

Thus, there is every reason to assert that in order to ensure satisfactory controllability of the movement of a plowing machine-tractor unit at a speed of approximately up to 2,2 m·s^{-1}, oscillation frequency of the rotation angle α of the tractor steering wheels should be smaller as the value of the velocity V_o is greater. When the speed of the working movement of a plowing tractor unit exceeds 2,2 m·s^{-1}, each value of the latter corresponds to such a value ω, at which the amplitude-frequency response of the workout of the dynamic system under consideration becomes necessarily useful, i.e., equal to one. Thus, for a speed of $V_o = 3$ m·s^{-1} we have $\omega = 0{,}75$ s^{-1} (see Fig. 1.13).

As follows from the analysis of the phase-frequency response of the plowing machine to the control action, the greater the speed value V_o, the smaller the lag (Fig. 1.14). It is quite another matter when the difference between the obtained phase-frequency characteristics is insignificant and generally appears at frequencies of the controlling influence greater than 0,5 s^{-1}.

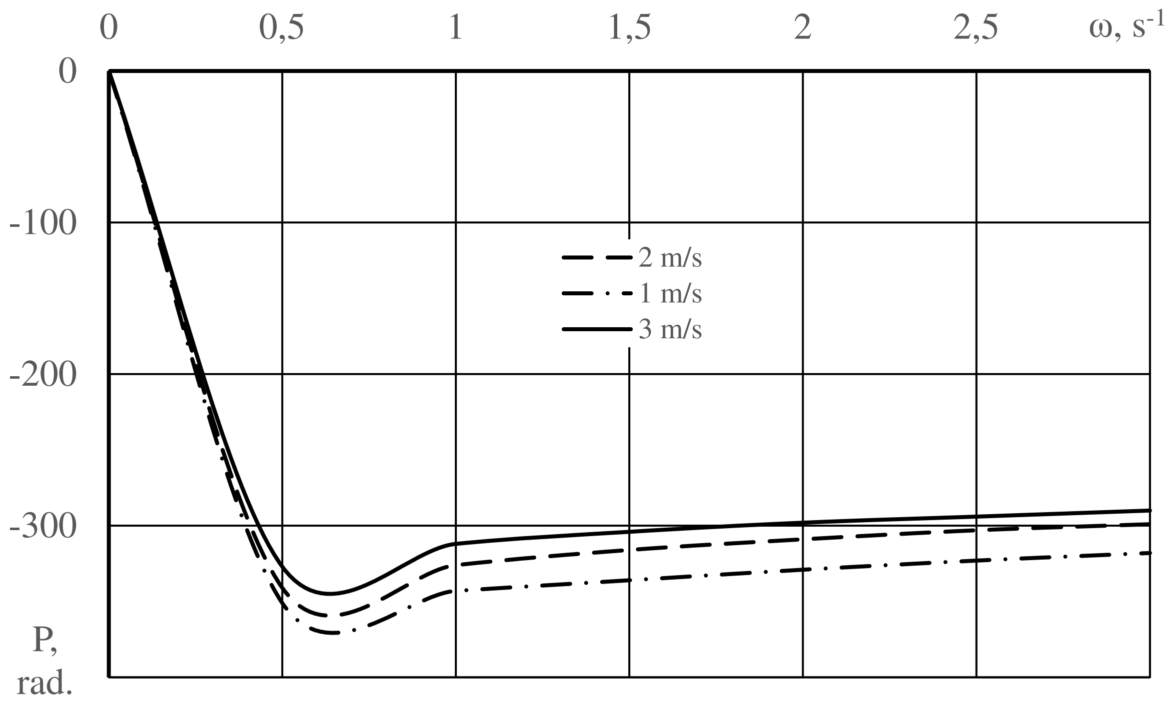


Fig. 1.14 – Phase-frequency response of a plowing machine and tractor unit to the control action α at different operating speeds

The highest value of phase shift (375° = 6,54 rad) приходится на частоту примерно 0,625 s^{-1} (see Fig. 1.14). The lag of the reaction of the arable machine to the control action is maximal and amounts to 10,5 s.

The influence of the coefficient of rear wheel steering resistance on the steerability of the tractor can be considered tangible at the frequencies of steering action less than 0,75 s^{-1} (Fig. 1.15).

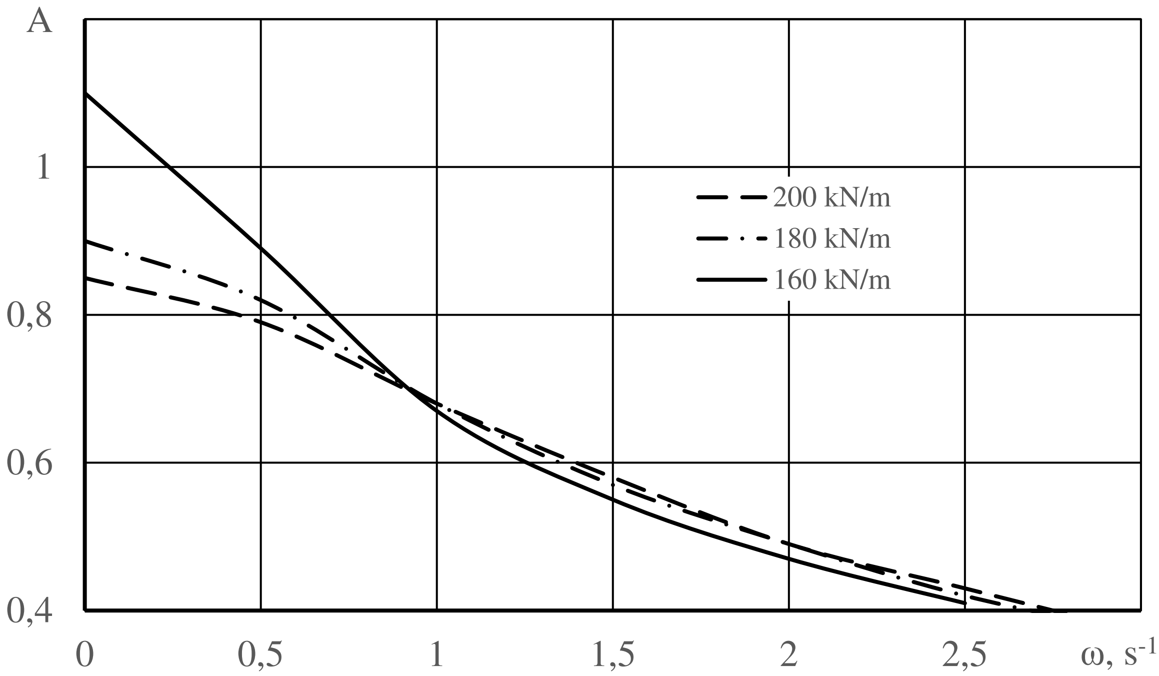


Fig. 1.15 – Amplitude-frequency response of the plowing unit to the control action α for different values of the coefficient k_b of the tractor's rear tire slip resistance

In the range of $\omega = 0 \ldots 0{,}75$ s^{-1} the decrease of value k_b leads to the desired increase in the amplitude-frequency response.

The influence of the resistance coefficient of the tractor front tires on the controllability of the plowing machine-tractor unit in qualitative terms is the opposite: the increase k_a of 120 to 160 kN.rad^{-1} causes a lowering of the amplitude-frequency response. In quantitative terms, this influence is so small that, from a practical point of view, it can be neglected.

The transfer function $W^{\beta}(s)$, which reflects the degree of change in the course angle φ of the power vehicle from the value of the angle β, is the ratio of the following two determinants:

$$W^{\beta}(s) = \frac{D^{\beta}(s)}{D(s)},$$

where $D(s)=\begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix}; D^{\beta}(s)=\begin{vmatrix} K_{11} & F_{11} \\ K_{21} & F_{21} \end{vmatrix}$.

The coefficients in these determinants are also taken from the system of equations (1.24).

After finding the above determinants and performing the appropriate transformations, we obtain:

$$W^{\beta}(s)=\left(C_1.s+C_o\right).\left[s.\left(B_2.s^2+B_1.s+B_o\right)\right]^{-1} \tag{1.34}$$

where $C_1=A_{11}.f_{22}$,

$$C_o=A_{12}.f_{22}-A_{24}.f_{12}.$$

Concerning coefficients B_o, B_1 and B_2, they are defined in the same way as in the equation (1.33).

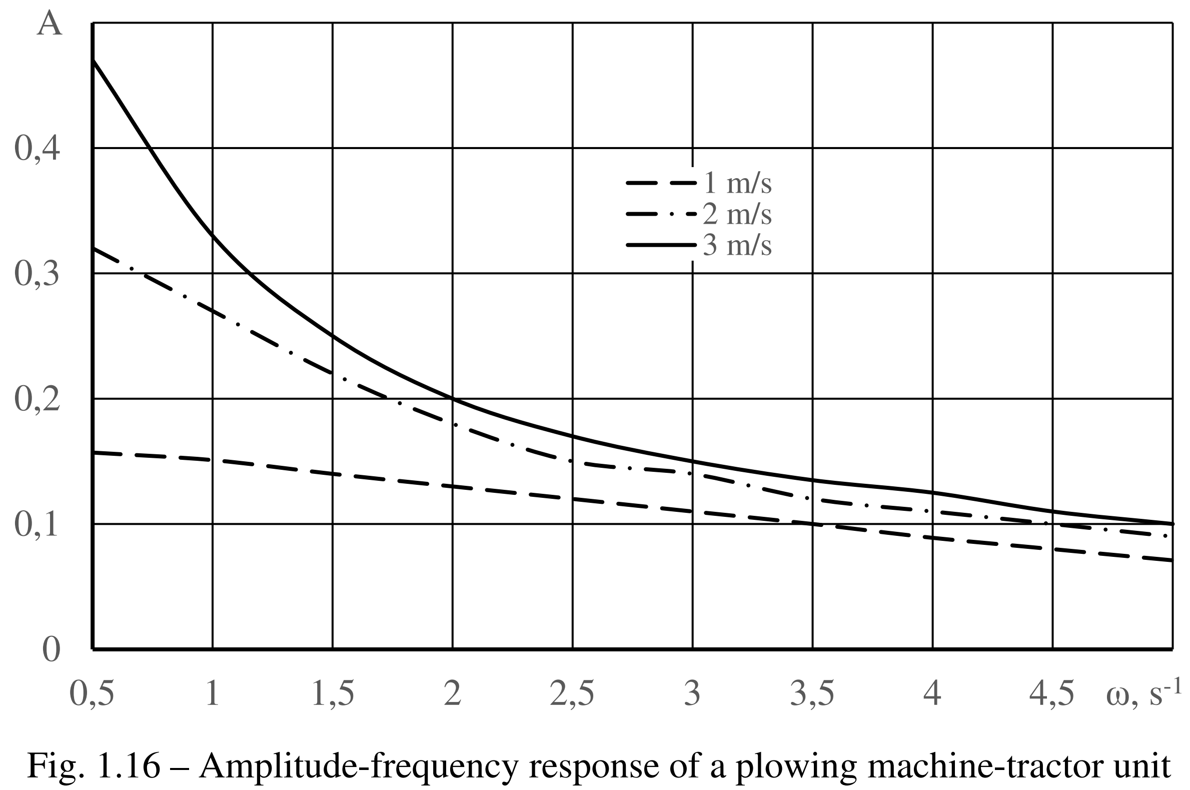


Fig. 1.16 – Amplitude-frequency response of a plowing machine-tractor unit oscillation of rotation angle β at different speeds of the working movement

When working out the investigated dynamic system of perturbing influences, the amplitude-frequency response in the working range of frequencies (0,5 … 5 s^{-1}) should be equal to zero, and the phase-frequency response tends to infinity [5].

As further analysis shows, the amplitude-frequency response of the disturbance in the form of plow oscillations in the horizontal plane grows undesirably when the speed of the plowing unit increases (Fig. 1.16). It is true that it is more or less perceptible only at low frequencies of angle fluctuations β (0,5 … 1,5 s^{-1}).

It should be emphasized that the response of the unit to the disturbing influence is less intense too, but it also (undesirably) decreases (Fig. 1.17).

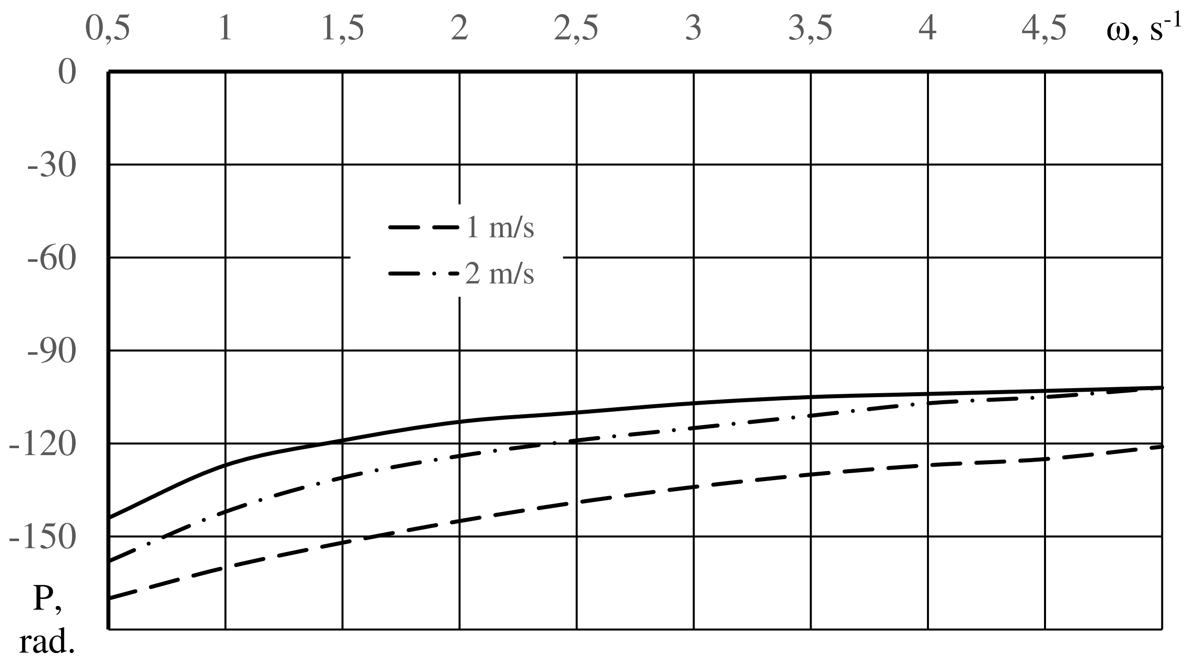


Fig. 1.17 – Phase-frequency response of a plowing machine and tractor oscillation of the rotation angle β at different speeds of the working movement

Another thing is that the magnitudes of undesirable increase in the amplitude-frequency response and decrease in phase-frequency response are so small in principle that they cannot act as a limit to increase the productivity of a plowing machine by increasing the speed of its working movement.

Another disturbing factor in the operation of a plowing machine is the fluctuation of the traction resistance of the plow. The transfer function that describes the character of changes in the course angle φ of the tractor under the influence of vibrations P_{kr}, is a polynomial (1.34) with:

$$C_1 = A_{11} . f_{23},$$

$$C_o = A_{12} . f_{23} - A_{24} . f_{13}.$$

The analysis of the amplitude-frequency characteristics obtained shows that they grow undesirably with the increasing speed of the plowing machine-tractor unit (Fig. 1.18).

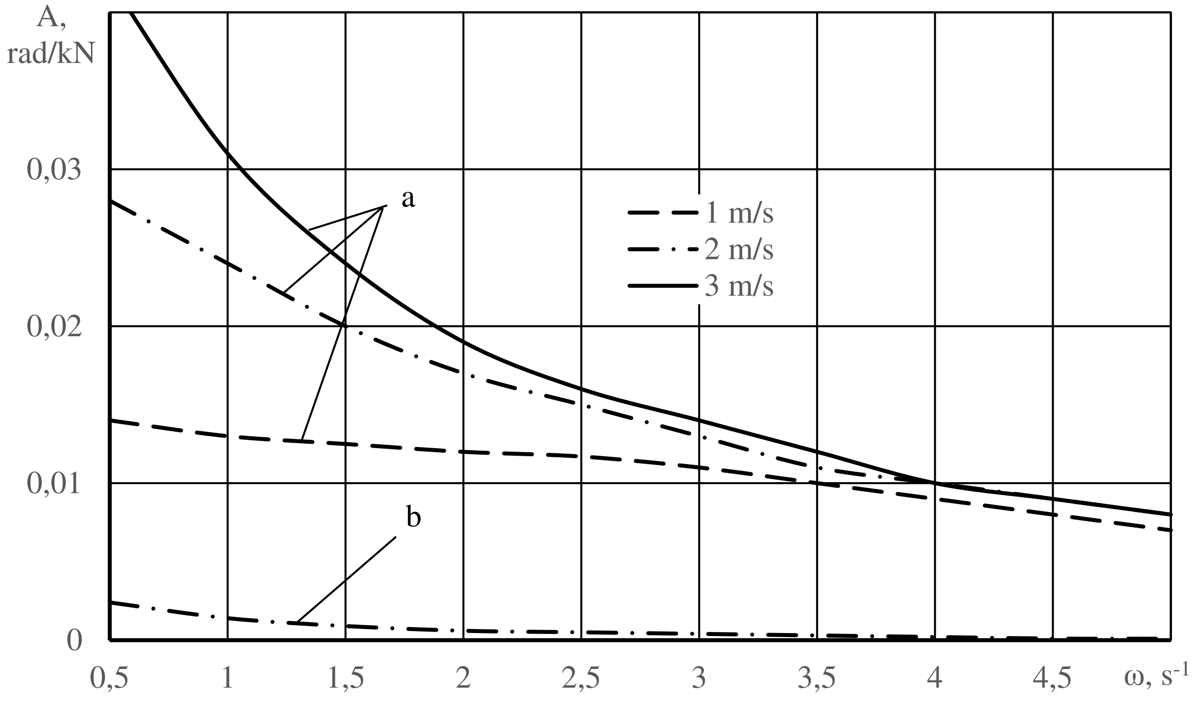


Fig. 1.18 – Amplitude-frequency response of traction resistance oscillations of a plowing machine-tractor unit P_{kr} at different travel speeds and rear linkage settings with two-point (a) and three-point (b) linkage schemes

The rear linkage setting scheme of the tractor has a significant influence in this case. To prove this, let us carry out the following analysis. Let the standard of fluctuations of the traction resistance of the plow be ±4 kN and be carried out with a frequency of 1 s^{-1}. Then, when the plowing machine moves at a speed of 2 $m \cdot s^{-1}$ and the adjustment of the rear attachment of the aggregating tractor is according to the two-point scheme (Fig. 1.18 a), the amplitude of oscillation of the course angle β of the latter will be 0,025 $rad \cdot kN^{-1} \cdot 4$ кN = 0,1 rad or 5,7°. At the same time, when adjusting the rear linkage of the power vehicle by the three-point linkage scheme, the amplitude of oscillations of its course angle will be more than ten times smaller (Fig. 1.18, b). However, at the same time the reaction time of the tractor to the disturbing influence will be significantly reduced. So, if at the same speed of the tractor (2 $m \cdot s^{-1}$) and oscillation frequency of the traction resistance of the plow 1 s^{-1}, the rear linkage of the power vehicle is set by the two-point linkage scheme, the phase shift will be 330 deg, i.e. 5,7 rad (Fig. 1.19 a). This means that the lag of the power vehicle's response to the action of the perturbation is equal to 5,7 s.

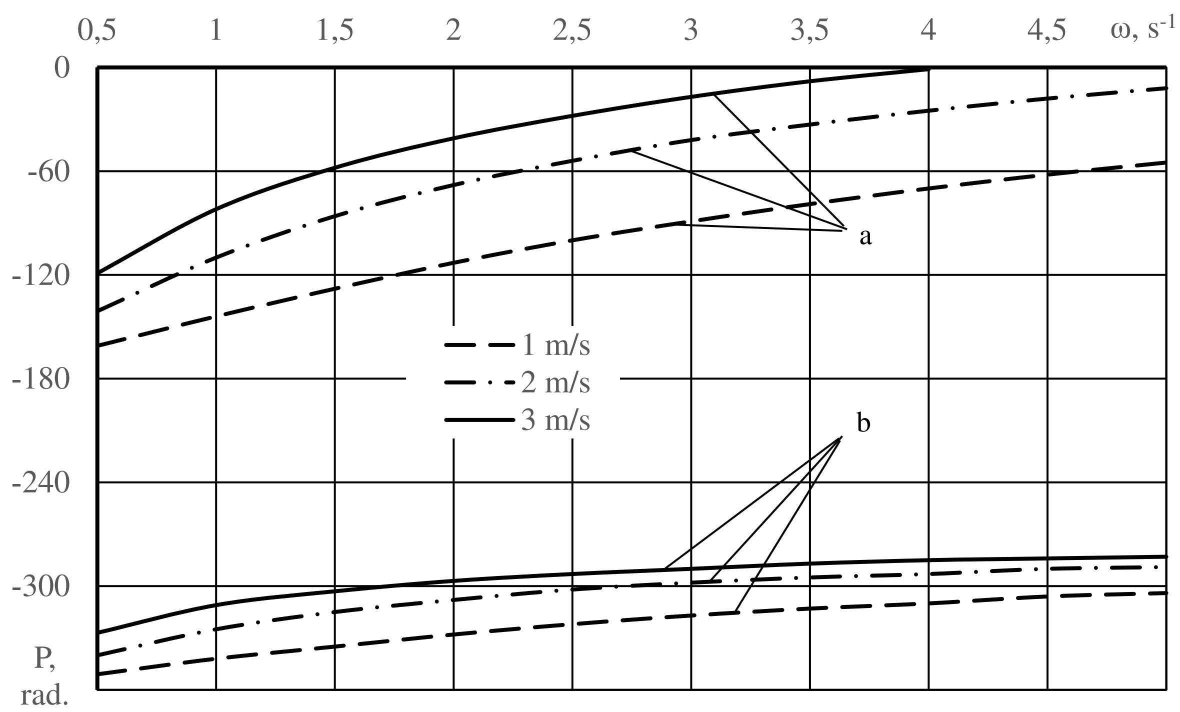


Fig. 1.19 – Phase-frequency response of plow's traction resistance oscillations of a plowing machine-tractor unit P_{kr} at different working speeds and rear linkage settings with two-point (a) and three-point (b) linkage schemes

When, under the same conditions, the tractor rear linkage is adjusted according to the three-point linkage scheme, the phase shift will be reduced to 109°, i.e. to 1.9 rad. (Fig. 1.19 b), and the response lag of the power vehicle to the disturbance will decrease to 1.9 s.

However, since the amplitude is more important than the speed of perturbation reproduction when working out the dynamic system, the above analysis unambiguously points to the advantages of the three-point setup of the tractor rear linkage when it works with a plow. Naturally, it is necessary to take into account the basic provisions stated by us in the first paragraph of this section.

1.5. Analysis of controllability and motion stability of an arable machine-tractor unit based on an integral tractor

Unlike the previous version of a plowing machine-tractor unit, the numerator of the transfer function describing the response of such a unit to the control action is a polynomial of second power:

$$W^{\alpha}(s) = \frac{C_2.s^2 + C_1.s + C_o}{s.(B_2.s^2 + B_1.s + B_o)},$$

where $C_2 = A_{11}.f_{21}$,

$$C_1 = A_{12}.f_{21} + A_{11}.f_{22} - A_{24}.f_{11},$$

$$C_o = A_{12}.f_{22} - A_{24}.f_{12},$$

$$B_2 = A_{11}.A_{21},$$

$$B_1 = A_{12}.A_{21} + A_{11}.A_{22},$$

$$B_o = A_{12}.A_{22} + A_{11}.A_{23} - A_{13}.A_{24}.$$

In this case, the steerability of a plowing machine depends significantly on its speed only in the frequency range of 0 ... 0,75 s^{-1} (Fig. 1.20). Outside of this range, the difference between the amplitude-frequency characteristics is insignificant and all of them have values close to the desired ones.

In the range ω = 0 ... 0,75 s^{-1}, with an increase of V_o to 2,0 m.s^{-1}, the amplitude-frequency characteristics increase, and the input overshoot coefficient can be as high as 6,8. However, a further increase in speed causes the amplitude-frequency response of the machine to decrease, gradually approaching the desired values. So, to perform the amplitude of control action by an articulated frame tractor-based plowing unit satisfactorily, first of all, it is necessary that the speed of the working movement of this plowing machine-tractor unit should be greater than 2 m·s^{-1}. If the velocity mode is close to this value, the frequency of oscillation of the control action must be more 0,75 s^{-1}.

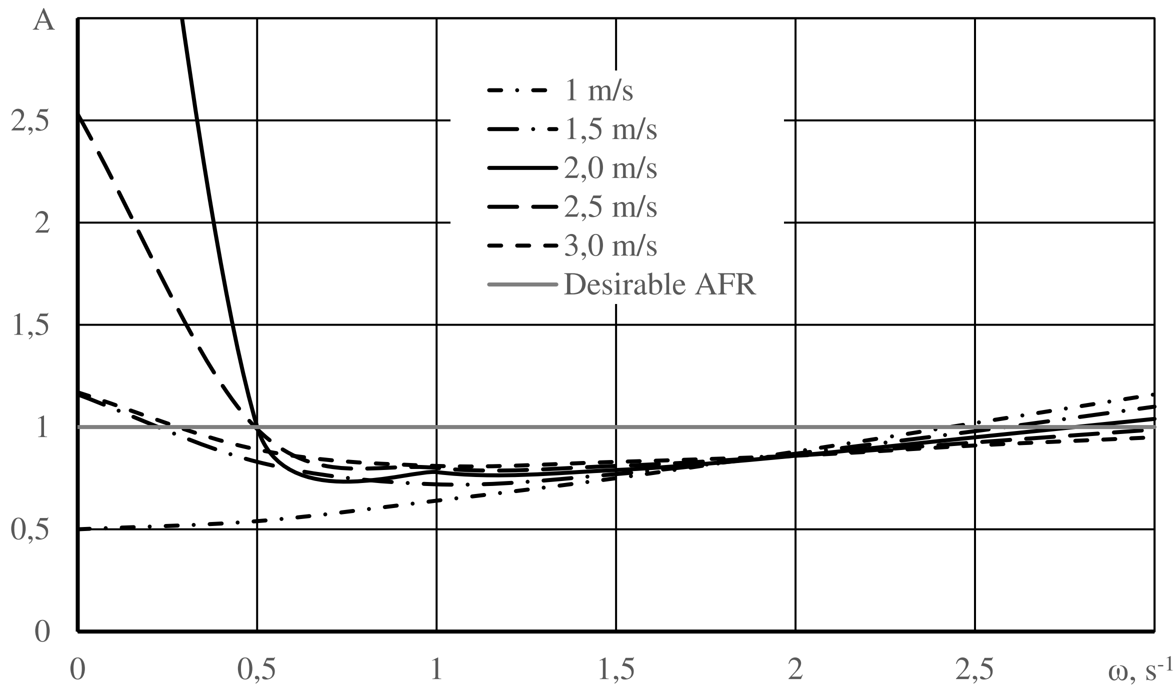


Fig. 1.20 – Actual and desirable amplitude-frequency response (AFR) of a plowing machine and tractor unit to the control action α at different operating speeds

Now let us consider the question, which concerns the rate of response of the dynamic system considered to the control action. The phase frequency characteristics differ in the same frequency range as the amplitude frequency characteristics (Fig. 1.21).

However, the smaller lag in the response of this unit to the input signal occurs in the frequency range 0 ... 0,4 s^{-1} at an operational speed not more than 2,5 $m \cdot s^{-1}$. With a larger value of this parameter, the phase shift in the entire operating frequency range is never less than 180°.

The resistance coefficient of the front tires of pneumatic tractor wheels has little effect on the process of changing the phase-frequency response (as well as the amplitude-frequency response).

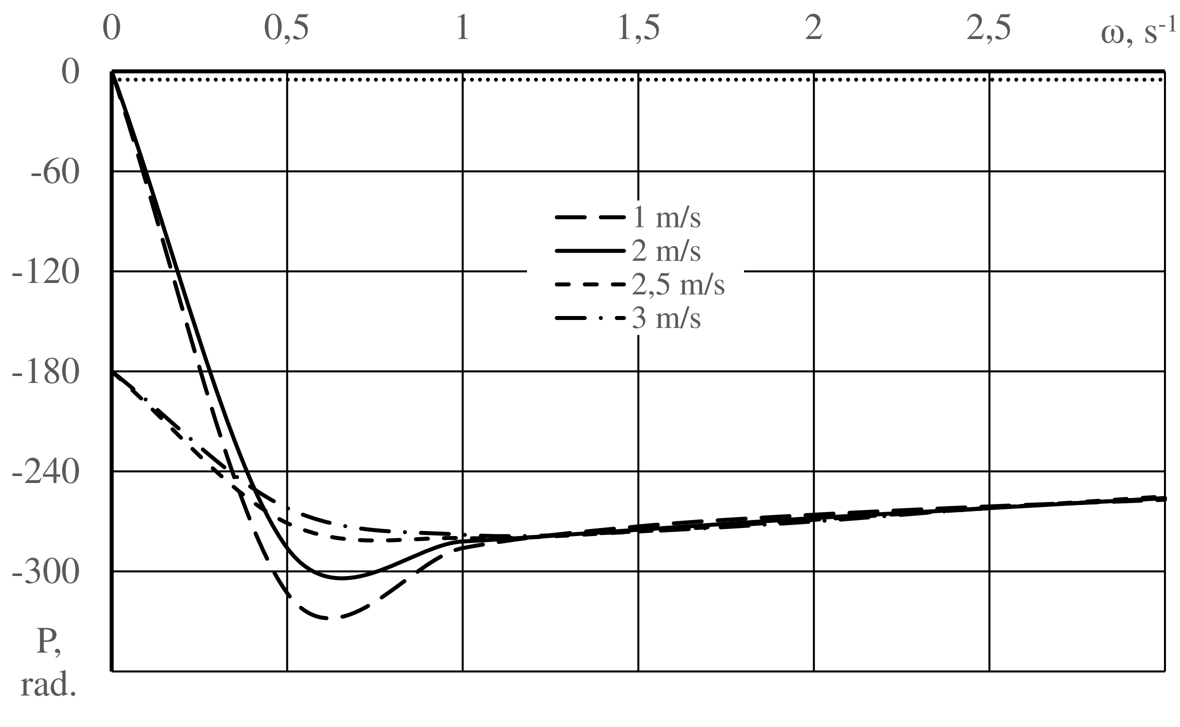


Fig. 1.21 – Actual and desirable (········) phase-frequency characteristics of the plowing machine-tractor control response α at different operation speeds

As for the coefficient k_b, its value must be greater than 100 $kN \cdot rad^{-1}$. Otherwise, the phase shift will always be greater than 2π radian (Fig. 1.22). But, at high values of the resistance coefficient of the rear tires of the pneumatic wheels of the tractor in the range of frequencies of oscillation of the steering action 0 … 0,4 s^{-1} the phase lag can be much smaller than 180°, which is the desired one. This fact can be confirmed by the following example.

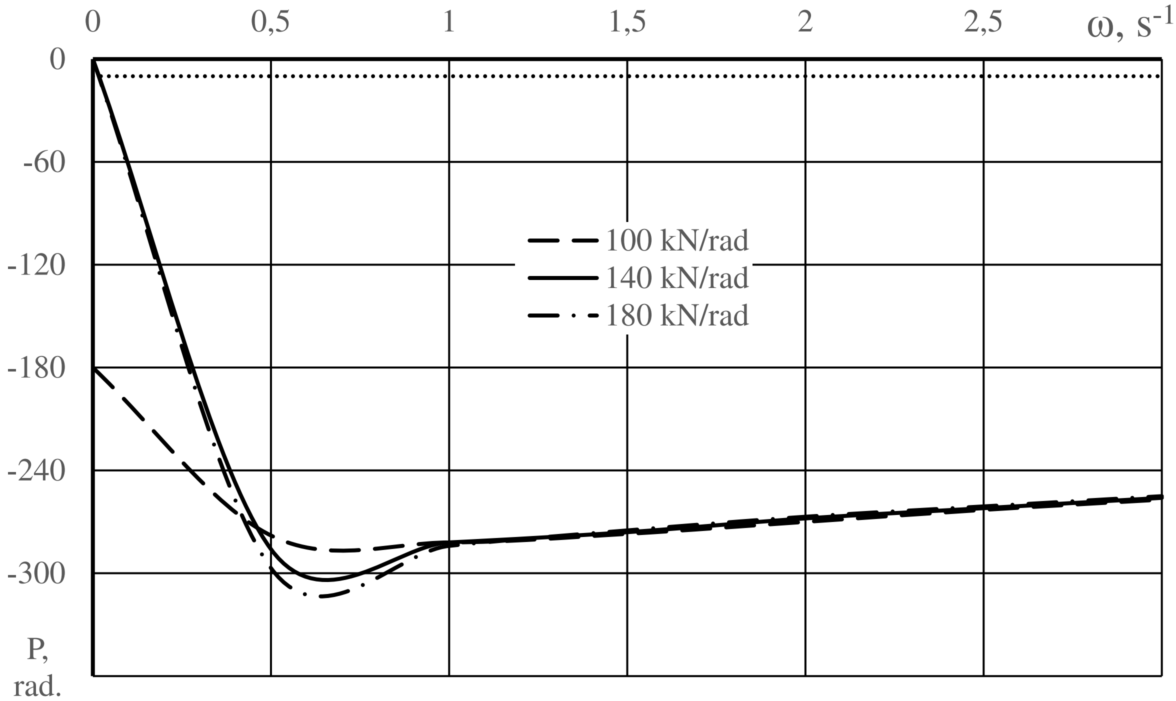


Fig. 1.22 – Actual and desirable (·······) phase-frequency characteristics of the plowing machine-tractor control response α for different values of the coefficient k_b of the tractor's rear tire slip resistance

Thus, with ω = 0,15 s^{-1} and k_b = 100 kN.rad^{-1} the phase shift is 200° or 3,5 rad. In fact, this means that the lagged response of a given plowing machine to the control action is equal to 3,5·(0,15)$^{-1}$ = 23,3 s. When k_b is increased to 180 kN.rad^{-1}, then the phase shift at the same value of ω is 1,05 rad, and the lagged response of the unit – 7 s, i.e. 3,3 times less.

Since it is the phase shift that determines the phase of the control action by the tracking dynamic system, the following three conditions must be met to ensure satisfactory controllability of the motion of the plowing machine-tractor unit in question:

1) $\omega = 0 \ldots 0{,}4\ \mathrm{s}^{-1}$;

2) $V_o < 2.5\ \mathrm{m{\cdot}s}^{-1}$;

3) $k_b > 100\ \mathrm{kN{\cdot}rad}^{-1}$.

There is no difficulty in fulfilling the first two conditions. As for the third condition, it can be realized only by setting the appropriate pressure ρ_g in the tires of the tractor rear pneumatic wheels. For this purpose, we can use the well-known formulas of R. Smiley and W. Horn, which relate the coefficient of k of the pneumatic tire slippage to the pressure ρ_g of the air in the tires, namely:

$$-\frac{h}{D} < 0{,}0885$$

$$k = C.\left[1{,}75.\frac{h}{D} - 12{,}7.\left(\frac{h}{D}\right)^2\right].\rho_g.b^2;$$

$$-\frac{h}{D} > 0{,}0885$$

$$k = C.\left[0{,}095 - 0{,}49.\frac{h}{D}\right].\rho_g.b^2$$

$$\frac{h}{D} = A.\frac{Q}{\rho_g.D^2}.\sqrt{\frac{D}{b}},$$

where h – depth of tracks, m;

D – tire outer diameter, m;

C – proportionality factor (for tires of conventional design $C = 60$);

ρ_g – air pressure in the tire, Pa;

b – tire width, m;

A – proportionality factor (for tires of conventional design A = 0,42);

Q – vertical load on the tire, N.

Using the last obtained analytical equations, it is possible to solve successfully the third condition (the necessary numerical value k_b depending on the pressure ρ_g in the tires of the pneumatic rear drive wheels of the tractor), which will provide satisfactory controllability of the movement of this plowing machine and tractor unit.

Thus, the qualitative character of plowing by the plowing unit based on the integral tractor with articulated frame is the same as for the machine-tractor unit based on the tractor with the classical configuration. As in the classic version, the three-point attachment design of the rear linkage of the power vehicle, compared to the two-point attachment system, provides better stability of the plowing machine movement in the horizontal plane.

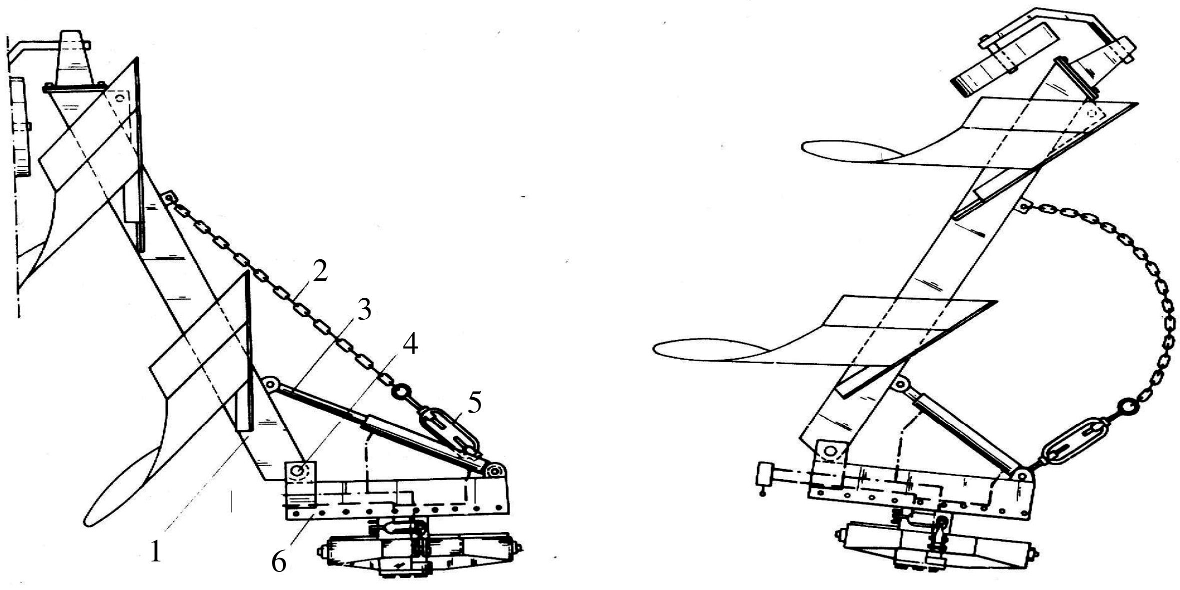
CHAPTER 2

AGGREGATING FRONT PLOWS

2.1. Trends in front plow designs

In the beginning, it is necessary to formulate the basic definitions and concepts associated with front plows. The terms “front plow”, “forward moving plow”, “front-mounted section”, “front-mounted plow”, etc. must be understood to mean plowing tools that are attached to the front-mounting vehicle (aggregate tractor) and work in its forward pushing mode. The technological process of plowing with such a plow does not differ at all from the same process performed by conventional trailed mouldboard plows. In any case, the frontal tools we are considering are not fundamentally different from those with plow bodies (moldboards) traditionally located [6].

For example, let us consider the design of a front plow according to the patent № 2481871 (France), whose diagram is presented in Fig. 2.1.



a) b)

Fig. 2.1 Front plow in working (a) and transport position (b)

This tool differs from an ordinary plowing tool by a chain 2 with a length regulator 5, as well as by a joint 4 and a hydraulic cylinder 3, through which the frame 1 is connected to the bar 6. Chain length 2 is selected so that the plow's support wheel in the floating position of the cylinder 3 is in the furrow.

How effective is the method of aggregating plows according to the “push-pull” scheme? First of all, it should be emphasized that the first patent for the front plow was obtained only in 1979, while now more than 13 companies around the world have established production of these plowing tools.

Table 2.1 shows some technical characteristics of the listed front mounted plows on the aggregating tractor [7, 8].

Analysis of the data in Table 1 shows that most European companies offer 2- or 3-body front plows. Four-body front plows are in most cases only made to order. Technical characteristics of similar plowing tools differ only slightly.

The French company Naud, which first offered front plows in 1980, found (Table 2.2), on the basis of a peculiar study, that the use of “push-pull” plows provides an increase in productivity by an average 45 ... 50 %, and a reduction in fuel consumption by 30 ... 35 % [9].

The normal operation of such units requires a tractor with a reversing control post. In our opinion, by implementing the needle method of turning, unproductive time and turn lane width can be reduced, while shift time utilization factor and shift productivity can be increased.

In foreign countries, and recently in Ukraine as well, rotary plows which are equipped with both left-hand (left-hand turning) and right-hand (right-hand turning) rotary bodies are widely used. Both the former and the latter, as a rule, are set by the toes of plow shares to one side. However, there are also plows of plowing machine-tractor units [10], in which left-handed (left-turning) bodies of both sections are directed opposite to right-handed bodies (Fig. 2.2).

Table 2.1 – Main technical characteristics of front plows

Characteristic Brand, company	N, nmb	B_n, m	L, m	H, m	M, kg
A 270 “Naud”	2	0,71	0,90	0,70	720
RPN 22 AH “Naud”	2	0,72	0,90	0,66	810
RPV 22 AH “Naud”	3	1,08	0,90	0,66	720
RPV 33 + 1AH “Naud”	4	1,14	0,90	0,72	1050
RP 23B2 “Souchu – Pinet”	2	0,70	0,96	0,68	730
RP 23T2 “Souchu – Pinet”	3	1,05	0,96	0,68	1000
BN 3327/0 “Thieme”	2	0,72	0,95	0,68	980
TN 3327/3 “Thieme”	3	1,05	0,95	0,68	1230
GP – II – 70 – 40 “Rabewerk”	2	0,70	0,93	0,75	750
GP – III – 70 – 40 “Rabewerk”	3	1,05	0,93	0,75	960
BA 40 – 140L “Charlier”	2	0,68	1,00	0,71	800
TA 40 – 140L “Charlier”	3	1,02	1,00	0,66	1010
T 65 R “Demblon”	2	0,70	1,05	0,65	810
T 65 S “Demblon”	3	1,20	1,05	0,65	1040
SB 8500 “Duro”	2	0,76	1,00	0,67	736
NS 8500 “Duro”	3	1,14	0,93	0,67	1012
FZ 73 R 32 “Gassner’	2	0,70	0,85	0,73	550
FD 73 N 32 “Gassner”	3	1,05	0,85	0,73	780
FV 73 N 32 “Gassner”	4	1,40	0,85	0,73	1050
Плуги фирмы “Losange”	2	0,71	0,83	0,75	990
Плуги фирмы “Bonnel”	2	0,70	1,00	0,65	670

N – number of main bodies; B_n – plow working width; L – distance between bodies; H – underframe height; M – mass.

Table 2.2 – Efficiency of combining tractors with front and rear attachments with plows (based on data [9])

Tractor brand	Number of front and back section plow bodies	Increasing the productivity of the machine and tractor unit, %	Reduced fuel consumption %
“Renault” 1181 – 4	6 (2 + 4)	50	29
“International Harvester” 1455	7 (3 + 4)	58	30
“Mersedes” 1300	6 (2 + 4)	35	16
“Massey Ferguson” 2680	8 (3+ 5)	60	26
“John Deer” 4040	5 (2 + 3)	33	57
“Renaulr” 1181 – 4	7 (3 + 4)	58	37

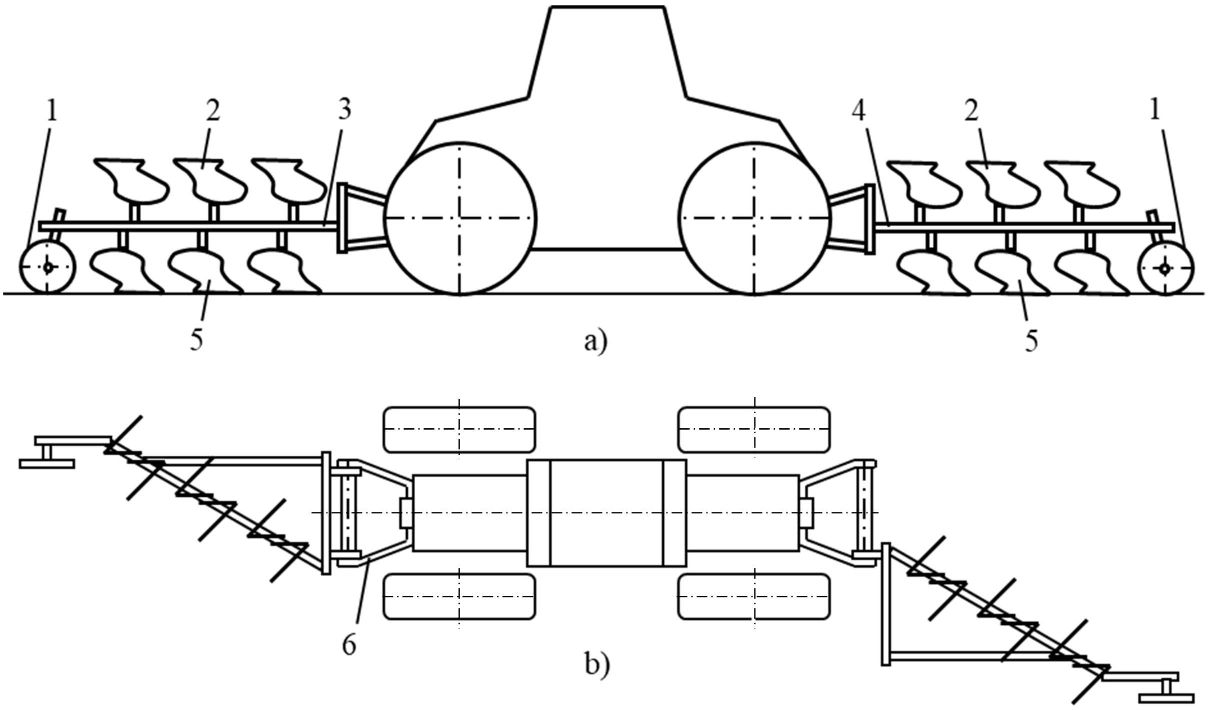


Fig. 2.2 – “Push-pull” plowing unit diagram
a) side view; b) pane view:
1 – support wheel; 2 – left-turning bodies; 3 – rear plow frame;
4 – front plow frame; 5 – right-turning bodies; 6 – tractor hitch

The same number of front and rear plow bodies is not optimal for a “push-pull” plow set. In [11] it is emphasized that the number of bodies of the front plow should be less than the rear plow. Sometimes the total capacity of a (2 + 3) tractor aggregate can exceed that of a (2 + 4) one.

Researchers in other countries have their own point of view on this issue. They also prefer plowing machine-tractor units with a large number of rear section bodies [12, 13]. It is emphasized that the stability of movement in the horizontal plane of the unit operating under the scheme (2 + 4) may be higher than that of the unit operating under the schemes (2 + 3) or (3 + 4). Pulling resistance of the front plow should not exceed 40% of the total pulling resistance of the plowing unit [12]. At the same time they unambiguously emphasize that plowing machine-tractor units working according to the scheme (2 + 3) are 14 ... 19 % more productive and 8 ... 10 % more economical than units with one rear plow (i.e. working according to the scheme 0 + 5).

During work, the position of the front plow frame relative to the tractor's front linkage can be movable (Fig. 2.1) or fixed (Fig. 2.3). The first scheme is designed in such a way that, according to the developers, no more attention can be paid to the front plow than to the rear plow. In such a scheme of frontal plow aggregation, its frame rotates around imaginary or real axis, which can be behind or before the point of application of equivalent resistance forces of the plowing tool (Fig. 2.4).

According to a number of researchers, the movable position of the front- mounted plow frame relative to the tractor's front linkage increases the stability of the plowing machine-tractor unit in the horizontal plane [12, 14]. Undoubtedly, accepting this result, in turn, we emphasize that the limitation of maneuverability of the implement is carried out on its support wheel, which, as mentioned above, moves along the furrow. If at the bottom of the latter there are lumps of soil with a diameter of 0,10 m or more (which is quite likely), the support wheel can plow out the front plow when it hits it. As a result, this will inevitably affect (quite negatively) the uniformity of both the depth and width of the plowing by this plowing machine.

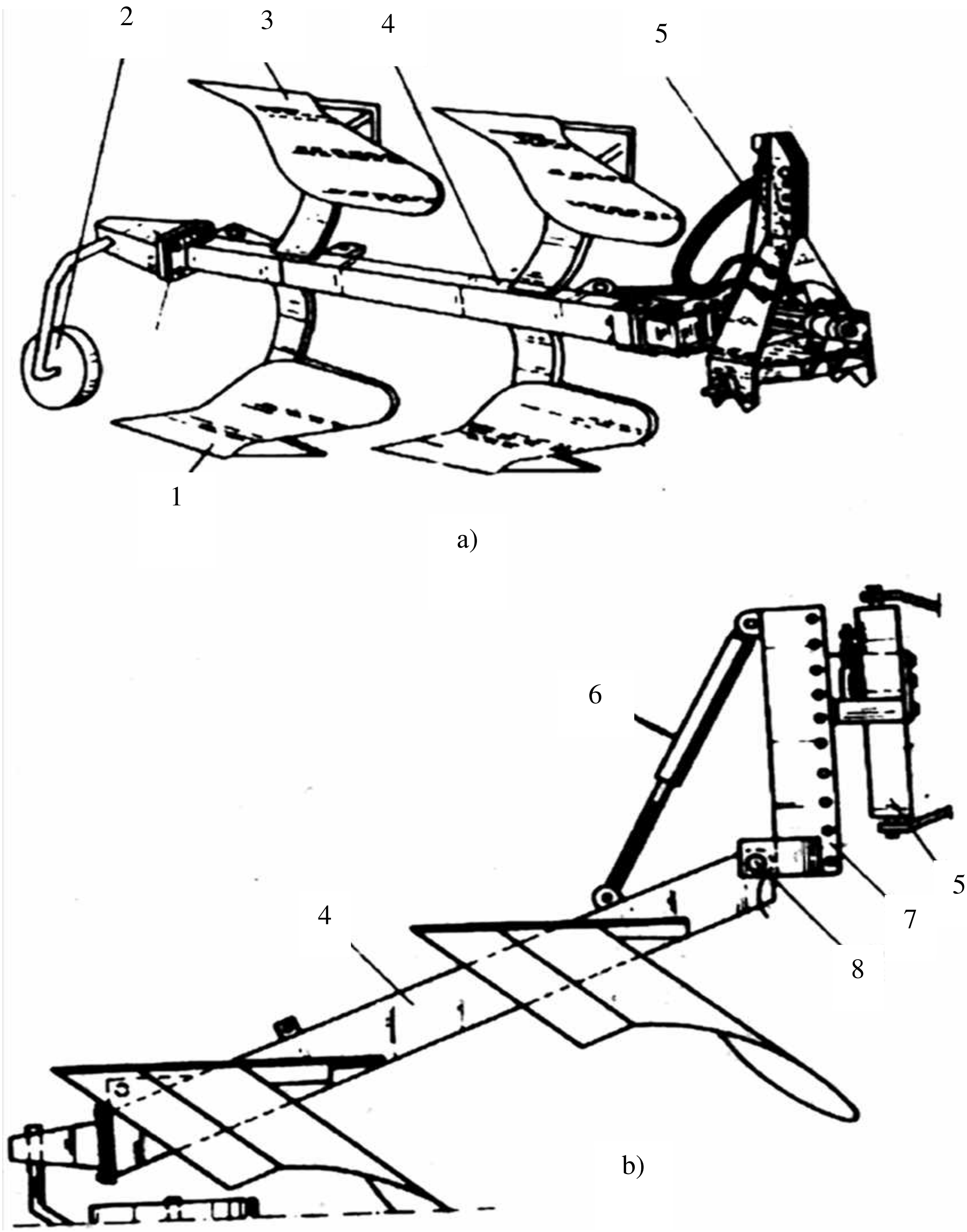


Fig. 2.3 – Front plow with fixed frame position relative to the tractor's front linkage:
1 – left-hand turning body; 2 – support wheel; 3 – right-hand turning body; 4 – plow frame; 5 – plow hitch; 6 – hydraulic cylinder; 7 – cross beam; 8 – joint

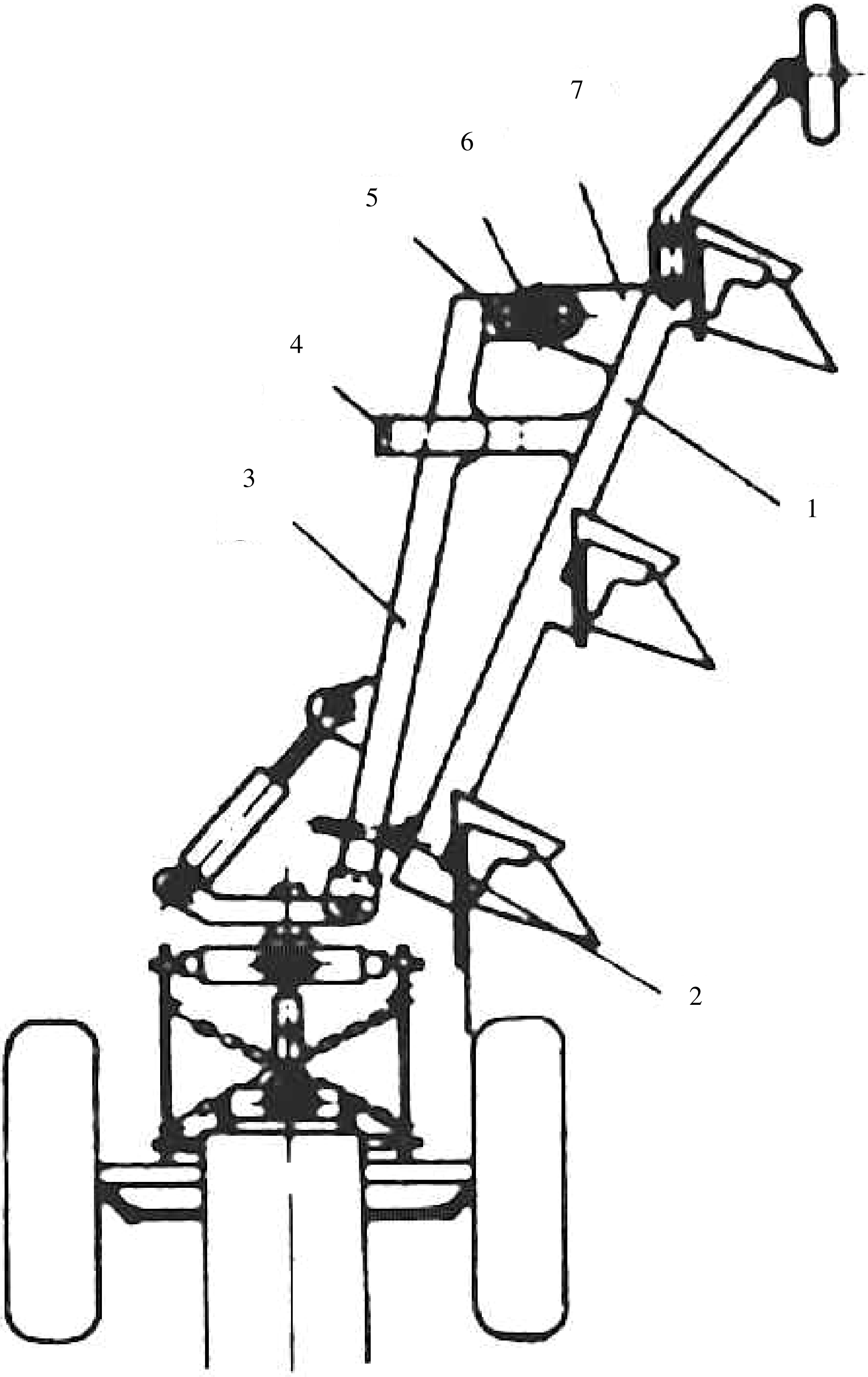


Fig. 2.4 – Front plow with the location of its pivot point in front of the point of application of equivalent drag forces:
1 – movable part of the plow frame; 2 – adjustment support; 3 – fixed part of the frame; 4 – slider; 5 – slot; 6 – roller; 7 – bracket

Based on the above, we can, in our opinion, conclude that the support wheel of the front mounted plow should move outside the furrow. In this case, one should keep in mind the following: If the tractor moves its right wheels in the furrow, the indicators of the plowing trajectory of this aggregating tractor are determined only by the curvature of the furrow of its previous pass. On this basis, there is no need for movable (and, therefore, more complex in design) connection of the front plow with the front attachment of the aggregating tractor. In addition, when the rear-mounted plow is connected symmetrically to the tractor, it is primarily affected by a small torque M_{fr} of the front plow. If, on the other hand, the rear-mounted tool will be attached with a left-hand lateral displacement, the effect of the torque M_{fr} will at least be balanced by the action of the momentum from the rear plow. It is highly likely that in the first, and even more so in the second case, the stability and controllability of the movement of the plowing machine-tractor unit with a rigidly fixed frontal tool, in which the support wheel moves outside the furrow, does not deteriorate.

Most researchers note that the use of a frontal plow allows (at least potentially) to increase the mass of adhesion of a power vehicle without the use of mechanical ballast, thereby increasing the degree of use of the power of its engine [9 – 13].

However, there are works that emphasize the fact of unloading the front axle of the tractor due to the action of the front plow. [9]. To eliminate this drawback, it is necessary to select the design parameters of the front hitch of the tractor and the hitch of the front plow correctly. At the moment, there is practically no clear way to solve this problem. Recommendation for an almost horizontal arrangement of the lower links of the front linkage of the power plant [9] cannot be considered sufficient, let alone exhaustive. After all, theoretical and experimental studies have shown [15, 16], that a significant influence on the vertical additional loading of the front wheels of the tractor is also exerted by the angle of the central link of its front hitch.

At the same time, units are used in which this hitch is open, i.e. the front plow works in semi-mounted mode. [11]. Here, the plowing depth is adjusted using the support wheel and the lower links of the front linkage of the tractor. Taking into account the influence of the angle of inclination of the latter on the vertical reloading of the front propellers of the power vehicle [15, 16], such a constructive solution, in our opinion, cannot be considered optimal. After all, setting a predetermined plowing depth can lead to the fact that the front ends of the lower rods of the front hitch will be below the hinges of their connection with the frame of the power unit, and this, as emphasized in [9, 15, 16], will undoubtedly lead to unloading of the front wheels of the latter with all the ensuing negative consequences.

It's another matter when the length (and, consequently, the angle) of the central link of the front linkage of the tractor is dynamically adjusted (Fig. 2.5). During operation, with an increase in the vertical load on wheel 4, a signal is automatically sent to the hydraulic valve, which supplies oil to the hydraulic cylinder 1 [17]. In this case, the rod of the latter is retracted and, lifting the plow frame together with the wheel, accordingly loads the front wheels of the power vehicle. This process, in essence, resembles the power method of adjusting the adhesion mass of tractors of the MTZ family.

Along with the advantages, the design of the frontal plow under consideration has a number of disadvantages. Firstly, this power method of adjusting the additional loading of the tractor propellers does not differ in satisfactory stability of the tool stroke in depth, since the change in this parameter and tractive resistance does not always unambiguously correlate. Secondly, it is characterized by a complex design, since it requires the installation of an additional automatic control system for the technological process, as well as chains (not shown in Fig. 2.5) that limit the lowering of the lower links of the front hitch of the tractor.

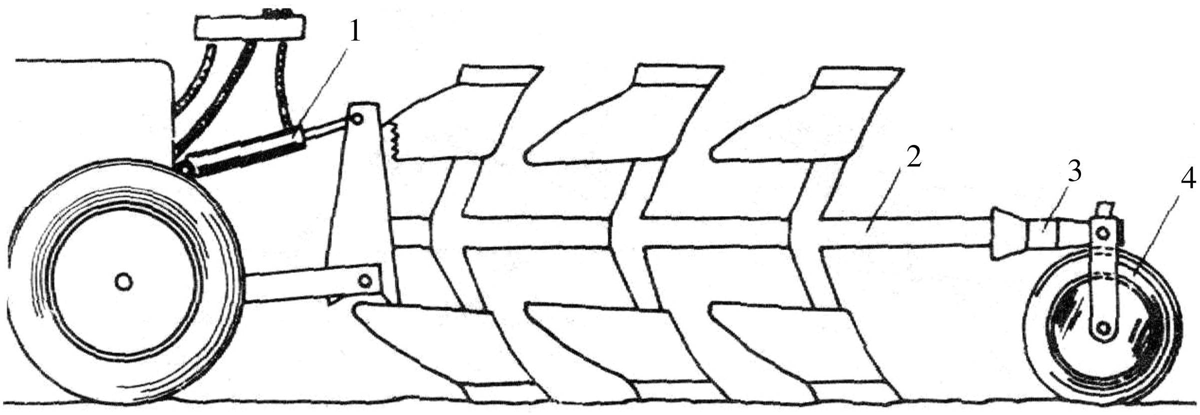


Fig. 2.5 – Front plow with power adjustment of plowing depth:
1 – hydraulic cylinder; 2 – plow frame; 3 – bracket; 4 – support wheel

An analysis of this and other schemes of front-mounted plows shows that for them, in contrast to rear-mounted plowing tool, the issue of choosing a place for installing the support wheel is practically not handled. With the function of adjusting the plowing depth with the front position of the support wheel on the plow frame, it does not cope effectively enough, in our opinion, especially when the support wheel moves in the furrow. Perhaps that is why recently studies have been carried out to determine the possibility and efficiency of the front plow without a support wheel at all [18].

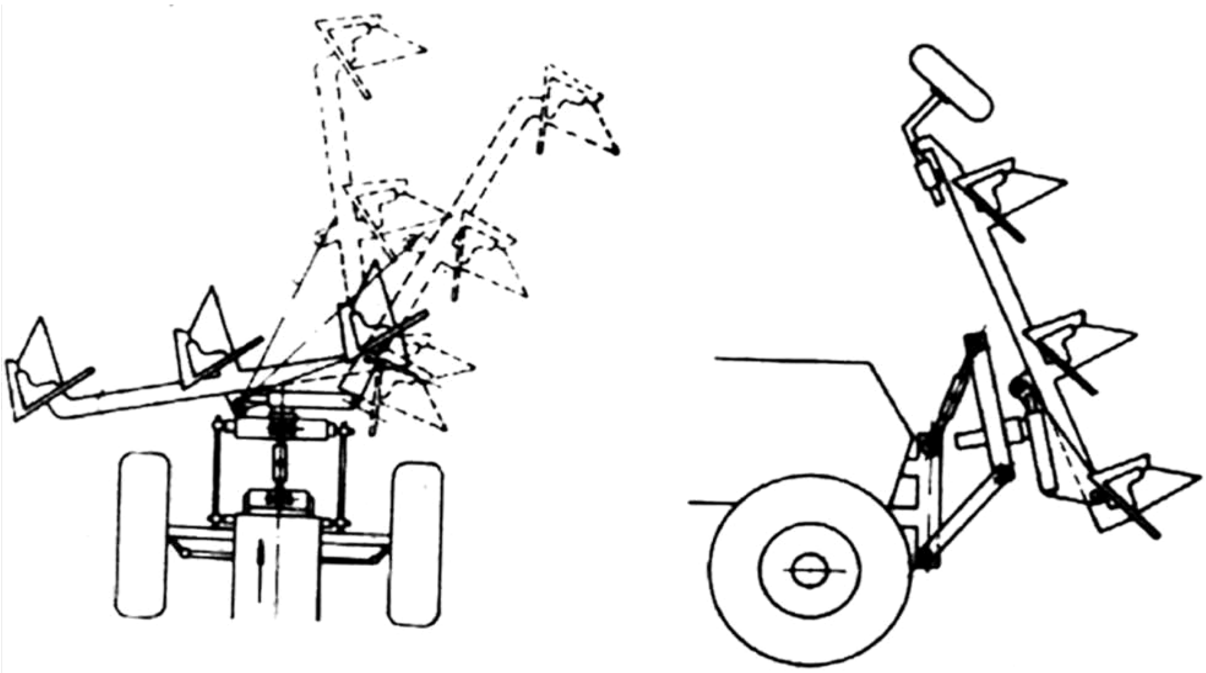
It should be emphasized that in some published works attention is given to increasing the stability of the plowing machine-tractor unit, operating according to the “push-pull” scheme, when driving in transport mode [16, 19]. There is no doubt in this statement, since during idle crossings with only one rear-mounted plow, the vertical load on the front wheels of the aggregating tractor is reduced – the greater the mass of the rear tool of the plowing machine-tractor unit, the bigger this reduction is. As a result, this leads to a deterioration in the controllability of the movement of such a plowing machine-tractor unit, especially when the front wheels of the tractor are steerable.

The presence of a front mounted implement significantly reduces the unloading of the front propellers of the power vehicle and, thus, contributes to an increase in the controllability and stability of the movement of the plowing machine-tractor unit during transport routes.

However, there is a need to consider this problem from a different angle. The front plow increases the total (and therefore kinematic) length of the plowing machine-tractor unit. A corresponding increase in the minimum design headland width should not be considered problematic, since some increase in non-production shift time must be compromised to increase productivity and efficiency. Moreover, there may not be an increase in the latter, since, as practice shows [20], an increase in the minimum design headland width does not always lead to an increase in its actual size. In other words, not every increase in the kinematic length of the unit leads to a corresponding increase in the width of the headland and, therefore, it is not accompanied by an increase in non-productive time.

Another thing is the impact of an increase in the total length of the new plowing machine-tractor unit on road safety. Some foreign companies see a solution to this problem in the design of such frontal plows, which would ensure their folding during transport crossings in horizontal or vertical planes (Fig. 2.6). Simultaneously with a decrease in the length of the unit, this, according to the developers [21], allows to relieve the front hitch of the tractor and its front axle from dynamic loads during transport crossing significantly.

Without questioning the above interpretations, it is necessary to emphasize the complexity of the design of such plows. In our opinion, such a technical solution can be resorted to in the case when the number of bodies of the front plow is at least three. The presence of only two-body front mounted plows does not lead to complications either during the working cycle or during transport routes [15, 16].As for the dynamics of vertical loads, it is not so much the transport as the working character of the movement of such a plowing machine-tractor unit that deserves attention. Despite the whole range of work in this direction, the dynamics of the movement of the power vehicle (tractor) in the longitudinal-vertical plane with only one rear-mounted plow was studied only to some extent [22 - 25].



a) b)

Рис. 2.6 – Front plow in transport position when folded in horizontal (a) and vertical (b) planes.

Certain studies of plow aggregates according to the "push-pull" scheme were carried out in this direction only at a static level. [26]. The object of research is a universal row-crop tractor of traction class 2 – MTZ-142. Theoretical calculations have shown that the hitching of 2-body front and 3-body rear plows, in comparison with the rear-hinged 4-body plowing tool, increases the aggregating weight of the tractor by 11 ... 13 %. The productivity of the plowing machine-tractor unit can be increased by 10 ... 15 %, which, by the way, casts doubt on the data published by the company “Naud” (see Table 2.2).

In the context of the analysis of these data, a natural question arises: What will be the result when comparing options for plowing machine-tractor units operating according to the schemes (2 + 3) and (0 + 5)? Indeed, with a plowing depth of 24 cm and a specific soil resistance of 50 kPa, as noted in [9], traction resistance of the five-body plow PLN-5-35 (for example) will be only 21 kN. According to traction tests, the MTZ-142 tractor with such a traction load can operate in its nominal mode [23].

In our opinion, the substantiation of the design parameters of the front hinged mechanism of the tractor when aggregating it with the front plow is not entirely complete [27]. First of all, this concerns the development of a well-founded methodology for choosing the angles of inclination of the central α and the lower β rods of the hinged mechanism of the aggregating tractor in the longitudinal-vertical plane in terms of ensuring the deepening of the plow into the soil both at the initial moment and during the working movement.

Correct selection of angular parameters α and β must at least prevent unloading of the front axle of the aggregated tractor due to the influence of the front mounted plowing tool. In the best case, it (the axle) should be loaded not only by the plow's working weight, but also by the vertical component of the traction resistance of the plow.

With an incorrect approach to the choice of the optimal values of the inclination angles of the central and lower rods of the aggregating tractor’s front hitch, the traction resistance of the front plow will redistribute the working and coupling weight of the plowing tool in favor of its rear axle (Table 2.3), which for known reasons is undesirable. It is quite possible to use a methodical approach to the selection of parameters α and β for modular power vehicles in general. However, the obtained patterns require additional verification, since they may not always be correct. [2, 15, 16].

For arable machine-tractor units, formed on the basis of an integral wheeled tractor HTZ-120/160, they may be fundamentally different, because of the fundamental difference between the design of this modular plowing tool and a power vehicle.

Table 2.3 – Distribution of the operating weight of tractors on axles when aggregated with front and rear plows (based on data from [9])

Indicators	4K2 tractor		4K4 tractor	
	Front axle	Rear axle	Front axle	Rear axle
Tire support surface, %	40	60	50	50
Unladen weight, %	48	52	60	40
Number of plow bodies	3	4	3	4
Weight in operation, %	36	64	47	53
Unloading the front axle, %	12	–	13	–

In addition, the question of the influence of the "push-pull" scheme on the labor intensity of aggregation, operational, technological and quality characteristics of the plowing unit remains unclear. The definition of the latter is especially important when the front plow is rigidly attached to the tractor with the front hitch and the support wheel of the tool located outside the furrow.

In general, the advantage of "push-pull" plows can be represented by the following block diagram:

$$\Delta N \longrightarrow \Delta G_{zh} \longrightarrow \Delta P_{kr} \longrightarrow \Delta B_p \longrightarrow \Delta W$$

Its essence lies in the fact that the mass and traction resistance of the front tool increase the vertical load on the front drive wheels of the power vehicle by a certain amount ΔN. As a result, its adhesion mass $\Delta G_{зч}$ and traction effort[1] ΔP_{kr} increase, accordingly. Through the growth of P_{kr} it is possible to increase the working width ΔB_p of the aggregated plowing tool and hence, the productivity ΔW of the whole plowing machine-tractor aggregate.

[1] difference between the tangential traction force and the rolling resistance of the tractor

With conventional (rear-mounted) plow aggregation, the coupling mass of the aggregated tractor $G^{з}_{zh}$ can be represented by the following equation:

$$G^{з}_{зч} = G + \Delta N^{з}_{зн}.g^{-1} = G_{п} + G_{з}, \tag{2.1}$$

where G – operating weight of the tractor;

$\Delta N^{з}_{зн}$ – additional loading of the rear axle of the tractor from the action of the vertical component of the traction effort of the tractor;

g – free fall acceleration;

$G_{п}$, $G_{з}$ – the adhesion weight of the tractor, which falls on its front and rear axles, respectively.

Since when the plowing tool is located at the rear, the front axle of the aggregating tractor is unloaded, the coupling weight from the plowing tool, which is transferred to the front axle of the tractor, will be equal to:

$$G_{п} = G_{по} - \Delta N^{з}_{п}.\, g^{-1}, \tag{2.2}$$

where $G_{по}$ – weight of the tractor, which falls on its front axle;

$\Delta N^{з}_{п}$ – force of vertical unloading of the front axle of the tractor.

For the rear axle of the aggregating tractor:

$$G_{з} = G_{зо} + \left(\Delta N^{з}_{п} + \Delta N^{з}_{зн}\right).g^{-1}. \tag{2.3}$$

where $G_{зо}$ – operating weight of the tractor on its rear axle.

From expression (2.3) it can be seen that the additional loading on the rear axle of the power unit (tractor) is due to both the direct effect of the rear-mounted plow $\Delta N^{з}_{зн}$, and the redistribution of mass over the axles by an amount $\Delta N^{з}_{п}.g^{-1}$. To ensure controllability of the movement of such a plowing machine-tractor unit, the value of the latter should not exceed the value $\lambda.G_{по}$ (where λ – coefficient of permissible unloading of the tractor steering wheels).

Taking into account expressions (2.2) and (2.3), equation (2.1) will have the following form:

$$G_{зч}^{з} = G_{по} + G_{зо} + \Delta N_{зн}^{з}.g^{-1}. \tag{2.4}$$

With the correct aggregation of the plows according to the "push-pull" scheme, additional loading of the front axle of the tractor takes place. As a result, we have:

$$G_{п} = G_{по} + \Delta N_{зн}^{з}.g^{-1}.$$

Taking this into account, we have:

$$G_{зч}^{з} = G_{по} + G_{зо} + \left(\Delta N_{зн}^{з} + \Delta N_{зн}^{п}\right).g^{-1}, \tag{2.5}$$

where $G_{зч}^{з}$ – traction weight of the tractor when using the „push-pull“ scheme.

Taking this into account, we can conclude: from formula (2.5) it follows that with a new aggregation of the plows, the coupling mass of the tractor increases - in the general case, by the amount $\Delta N_{зн}^{з}.g^{-1}$.

Since the number of rear plow bodies (and therefore the width of the plow’s coverage) in a traditional plowing machine-tractor unit is greater than that in the rear section of a unit operating in a “push-pull” mode, it is usually:

$$\Delta N_{зн}^{з} > \Delta N_{зн}^{н},$$

where $\Delta N_{зн}^{н}$ – additional loading of the rear axle of the tractor from the impact of the rear section of the plowing machine-tractor unit, operating according to the “push-pull” scheme.

If in real operating conditions, it will be so that:

$$\Delta N_{зн}^{п} + \Delta N_{зн}^{н} \le \Delta N_{зн}^{з},$$

then the advantages of the new plow aggregation scheme will be nullified. The parameters of the front hitch and front plow must ensure the fulfillment of the condition:

$$\Delta N_{зн}^{п} + \Delta N_{зн}^{н} > \Delta N_{зн}^{з}.$$

If this is achieved, then the adhesion weight of the tractor with the new scheme of its aggregation as part of the plowing machine-tractor unit will be increased due to:

$$\Delta G_{зч} = \left(\Delta N_{зн}^{п} + \Delta N_{зн}^{н} - \Delta N_{зн}^{з} \right) . g^{-1}. \qquad (2.6)$$

Since the traction force of the aggregating tractor will be equal to $P_{kr} = G_{зч}.g.(\varphi - f)$, then the new plowing machine-tractor unit will have a larger one. Its growth will be such a value:

$$\Delta P_{kr} = (\varphi - f)\left(\Delta N_{зн}^{п} + \Delta N_{зн}^{н} - \Delta N_{зн}^{з} \right), \qquad (2.7)$$

where *φ, f* – respectively, the coefficients of adhesion and rolling resistance of the running wheels of the aggregated tractor.

Additional loads included in expression (2.7) can be represented as follows:

$$\Delta N_{зн}^{п} = N_{зн}^{п} - G_{зо}.g,$$

$$\Delta N_{зн}^{н} = N_{зн}^{н} - G_{зо}.g, \qquad (2.8)$$

$$\Delta N_{зн}^{з} = N_{зн}^{з} - G_{зо}.g,$$

where $N_{зн}^{п}$, $N_{зн}^{н}$ – actual values of vertical load on the rear and front axles of the tractor when aggregating plows according to the “push-pull” scheme;

$N_{зн}^{з}$ – actual value of the vertical load on the rear axle of the tractor with its traditional aggregation with a rear-mounted plow.

Substituting the value of expressions (2.8) into expressions (2.6) and (2.7), after the performed transformations, we obtain:

$$\Delta G_{зч} = \left(\Delta N_{зн}^{п} + \Delta N_{зн}^{н} - \Delta N_{зн}^{з}\right).g^{-1},$$

$$\Delta P_{kp} = (\varphi - f).\left(\Delta N_{зн}^{п} + \Delta N_{зн}^{н} - \Delta N_{зн}^{з} - G_{no}.g\right).$$

Values $N_{зн}^{п}$, $N_{зн}^{н}$ and $N_{зн}^{з}$ may be obtained either by direct measurement in the course of field experimental studies, or theoretically.

When driving uniformly on the surface of the field with a slight slope, the pulling power of the tractor $P_{кр}$ and working width B_p are related to each other by the following dependence:

$$B_p = P_{кр}.(k.h)^{-1},$$

where k – specific traction resistance of the plow, kN·m^{-2}; h – plowing depth, m.

Thus:

$$\Delta B_p = \Delta P_{кр}.(k.h)^{-1},$$

or

$$\Delta B_p = (\varphi - f).\left(N_{зн}^{п} + N_{зн}^{н} - N_{зн}^{з} - G_{no}.g\right).(k.h)^{-1}. \quad (2.9)$$

In general, increasing the working width of a plowing tool leads to an increase in its traction resistance. If the coupling weight of the aggregated tractor remains unchanged, the slip of its undercarriage increases. As a result, the speed of such a plowing machine-tractor unit is reduced by a certain amount ΔV.

As mentioned above, for a “push-pull” plowing machine-tractor aggregate, simultaneously with the increase in working width, the coupling weight also increases. This gives us the right to assume apriori that the speed of such a plowing machine-tractor unit will change by such a small value that it can be neglected.

The productivity of the new arable machine-tractor unit will increase. Taking into account expression (2.9) its increase will be equal to:

$$\Delta W = 0{,}1.V.(\varphi - f).\left(N_{зн}^{п} + N_{зн}^{н} - N_{зн}^{з} - G_{по}.g\right)(k.h)^{-1}. \quad (2.10)$$

For illustration, we can make this particular calculation. According to our field experimental research [28] for wheeled tractor KhTZ-120, which in one standard variant aggregated plow PLN-5-35, and in the second experimental variant it was adjusted for coupling simultaneously with three – bodied rear plow PLN-3-35 and two - bodied front plow PLN-2-35 (the brands of these three-bodied and two-bodied plows are of course conventional, determined by the number of their bodies in comparison with standard five-bodied plow PLN-5-35). For the second experimental variant of aggregation the result obtained was: $N_{зн}^{п} + N_{зн}^{н}$ = 93 кN; $N_{зн}^{з}$ = 6 кN; $G_{по}.g$ = 50 кN.

For further calculations for both variants we took the same values of the parameters: h = 0,30 m; f = 0,1; φ = 0,66 и k = 70 kN·m^{-2}.

The calculations showed the following. As follows from expression (2.10), at velocity 8 km/h, the increase in productivity of the plowing machine and tractor unit operating under the "push-pull" scheme will be 0,15 ha·h^{-1}. The increase in the working width of the machine will be almost 0,19 m. It follows that the tractor in question can be coupled with a two-bodied front plow and a three-bodied rear plow, with the working width of each of the plow bodies not 0,35 m (as, for example, in a standard plow PLN-5-35), but 0,4 m. The design working width of such a plowing machine is thus, – 2,0 m.

2.2. Conditions for plowing the soil with a front-mounted plow aggregated on the tractor

When attaching a front plow, the center link of the front linkage of the aggregating tractor can occupy two positions, each of which is determined by the angle α (Fig. 2.7). In addition, the modulus of this angle can vary from 0° to maximum ($\alpha_{Max.\partial}$) and from 0° to minimum ($\alpha_{min.\partial}$) constructively permissible values.

As for the lower links of the front linkage, they can only have a positive slope in order to avoid unloading the front wheels of the aggregating tractor. In addition, the value of the angle β should be greater than 0°, but smaller than the maximum constructively permissible value ($\beta_{Max.\partial}$).

The constructively permissible values of the tilt angles of the central ($\alpha_{Max.\partial}$, $\alpha_{min.\partial}$) and lower ($\beta_{Max.\partial}$) links of the aggregating tractor's front are set taking into account the values of the vertical coordinates of points *A* and *B* (Fig. 2.7) on its front linkage and the same coordinates of points *S* and *F* on the front plow hitch.

On this basis, the front linkage of the aggregating tractor can have three different settings:

I) $\alpha_{max.\partial} \geq \alpha \geq 0,\ \beta_{max.\partial} \geq \beta \geq 0,\ \alpha > \beta$;

II) $\alpha_{max.\partial} \geq \alpha \geq 0,\ \beta_{max.\partial} \geq \beta \geq 0,\ \alpha < \beta$;

III) $\alpha_{min.\partial} \leq \alpha \leq 0,\ \beta_{max.\partial} \geq \beta > 0,$

We will consider each of these options separately.

In this case, first of all, for each of these three options of setting, let us build diagrams of forces with the indication of the structural linear and angular dimensions and the selected coordinate systems in the longitudinal-vertical plane.

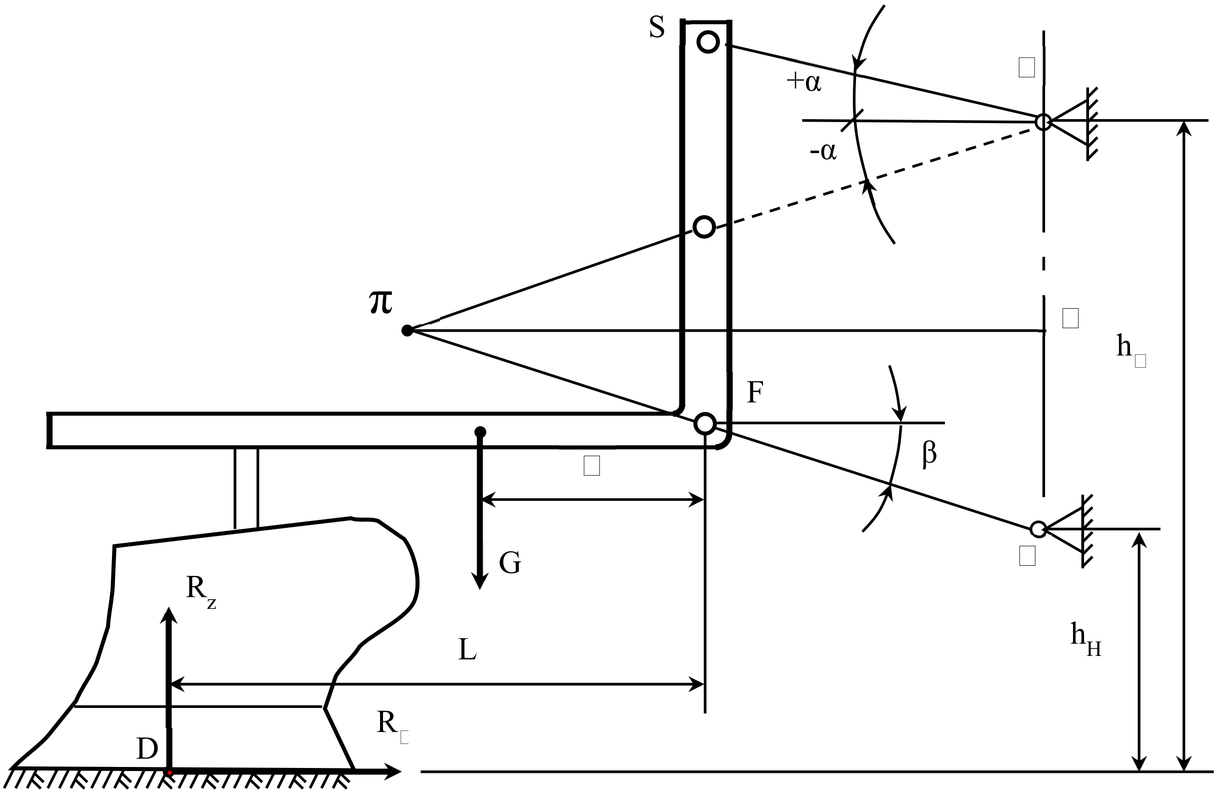


Fig. 2.7 – Diagram of forces acting on the front plow
when ploing into the soil

In case of using variant I, such a scheme of forces is shown in Fig. 2.8. At that, at the moment of putting the frontal plow into the soil, in addition to gravity force *G*, act the total (reduced to one "equivalent" body) vertical R_z and horizontal R_x field surface forces.

The torque $M_{(Rz)}$, arising from the exertion of force R_z on point π (instantaneous center of rotation of the tractor front linkage) prevents the plow body from sinking into the soil. Its value can be found from the following expression:

$$M_{(Rz)} = R_z\left(L + r_н + \cos\beta + C\pi\right), \qquad (2.11)$$

where $r_н$ – lower links length of the tractor's front linkage (i.e. their projection on the longitudinal-vertical plane); *L*, $C\pi$ – the design parameters in Fig. 2.8.

From the analysis of expression (2.11) it follows that to completely exclude the influence of the torque $M_{(Rz)}$ is impossible. You can decrease its value only by selecting the angles α and β to be such that shoulder $C\pi$ is minimal.

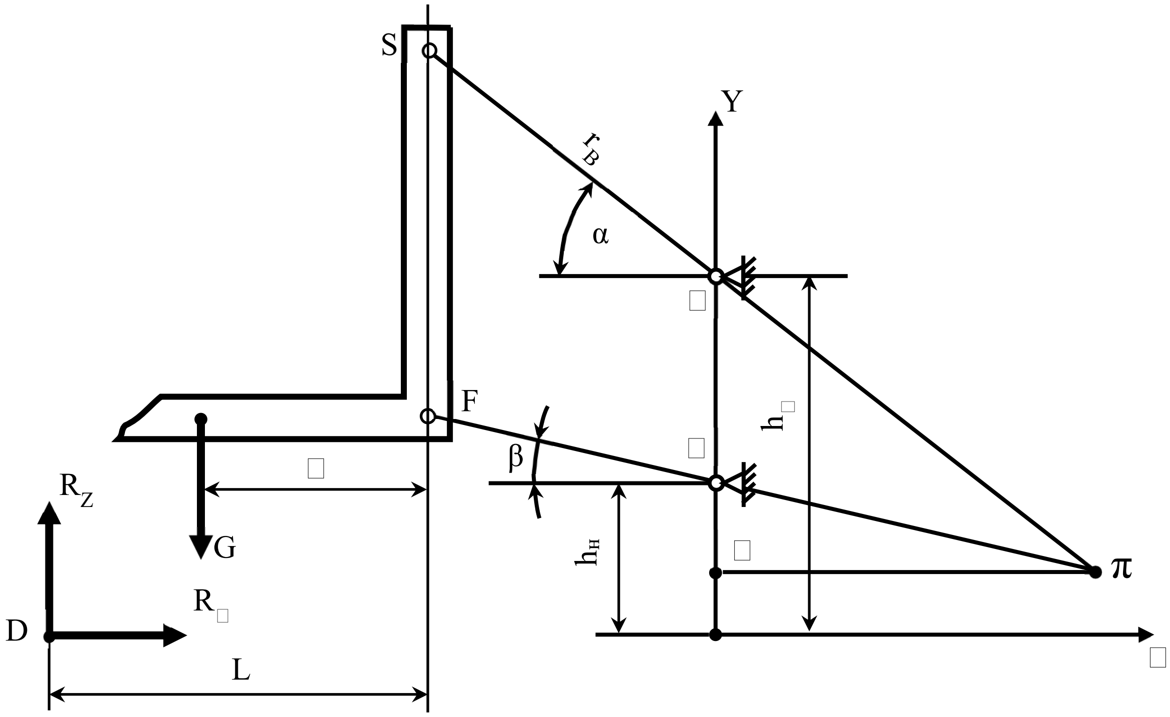


Fig. 2.8 – Diagram of forces that act on the frontal plow at the moment of its penetration into the soil, in the case of $\alpha > \beta$

To find the dependence of the length of $C\pi$ on the angles α and β, we will, first of all, make the equations for the lengths of the straight line segments *SA* and *FB*. In general, we will have this dependence [29]:

$$\frac{Y - Y_1}{Y_2 - Y_1} = \frac{X - X_1}{X_2 - X_1}, \tag{2.12}$$

For segment *SA*:

$Y_1 = h_в + r_в.\sin\alpha;$

$Y_2 = h_в;$

$$X_1 = -r_в.\cos\alpha;$$

$$X_2 = 0, \tag{2.13}$$

where $r_в$ – the length of the center link of the tractor front linkage.

After substituting expression (2.13) into expression (2.12) and appropriate transformations we obtain:

$$Y = -X.\tan\alpha + h_в. \tag{2.14}$$

For segment *FB* the coordinates of its endpoints will be:

$$Y_1 = h_н + r_н.\sin\beta;$$

$$Y_2 = h_н;$$

$$X_1 = -r_н.\cos\beta;$$

$$X_2 = 0.$$

The equation for straight line *FB* has the following expression:

$$Y = -X.\tan\beta + h_н. \tag{2.15}$$

As point π is the crossing point of lines *SA* and *FB*, by equating the right-hand sides of expressions (2.14) and (2.15) we find that:

$$X = C\pi = (h_в - h_н).(\tan\alpha - \tan\beta)^{-1}. \tag{2.16}$$

Given expression (2.16), equation (2.11) can now be written as follows:

$$M_{(Rz)} = R_Z.\left[L + r_н.\cos\beta + \left(h_в - h_н\right).\left(\tan\alpha - \tan\beta\right)^{-1}\right]. \tag{2.17}$$

Analysis of the resulting expression (2.17) shows that, in order to reduce the moment $M_{(Rz)}$ of exerting force R_z to minimum, only one condition must be fulfilled, namely the condition when the difference ($\tan\alpha - \tan\beta$) will be maximal. And this is only possible when:

$$\alpha \to \alpha_{max.\partial} \text{ and } \beta \to \beta_{max\partial} \quad (2.18)$$

It should be noted that this trend in the value of the angle β leads to an increase of the multiplication sum $r_н.\cos\beta$. But since there is a reason to believe that this increase is completely outweighed by the magnitude of the decrease in the expression $(h_в - h_н).(\tan\alpha - \tan\beta)^{-1}$, then the end result is that condition (2.18) leads to the desired reduction in the magnitude of the torque $M_{(Rz)}$.

Unlike the torque $M_{(Rz)}$, the torque relative to point π from the exertion of force R_x helps the plow to get into the soil. As follows from the diagram in Fig. 2.8, the value of this sinking moment will be:

$$M_{(Rx)} = R_x(h_н - BC). \quad (2.19)$$

From the analysis of the expression obtained (2.19) it can be seen that torque $M_{(Rx)}$ will have a maximum value when the distance $BC = 0$. Given that $BC = C\pi.\tan\beta$, with the dependence (2.16), we obtain:

$$BC = (h_в - h_н).\tan\beta.(\tan\alpha - \tan\beta)^{-1},$$

and the final expression for determining the torque $M_{(Rx)}$ will look as follows:

$$M_{(Rx)} = R_x.[h_н - (h_в - h_н).\tan\beta.(\tan\alpha - \tan\beta)^{-1}. \quad (2.20)$$

It is easy to see that the value of the segment BC will be equal to zero in the case when $\beta = 0$. However, the value of this angle at the moment of plowing into the soil should be greater than zero, since after setting the plow to a given depth of tillage, the lower links of the tractor front linkage will have a slope at which unwanted unloading of the front wheels of the aggregating tractor is possible [16].

However, it also follows that the segment *BC* will be the smallest, and the sinking moment, defined by expression (2.20), will respectively be the greatest, provided that the difference ($\tan\alpha - \tan\beta$) is maximal, i.e. in the case when the value of $\tan\beta$ is the smallest. As in the previous case, this will be possible when:

$\alpha \to \alpha_{\max.\partial}$ and $\beta \to \beta_{\min.\partial}$.

The plowing torque $M_{(G)}$ from the exerted force *G* of the plow's gravity will be determined by the following expression:

$$\begin{aligned} M_{(G)} &= G(a + r_{н}.cos\beta + C\pi) = \\ &= G[a + r_{н}.cos\beta + (h_{в} - h_{н}).(\tan\alpha - \tan\beta)^{-1}, \end{aligned} \tag{2.21}$$

where *a* – longitudinal coordinate of the plow's center of mass (Fig. 2.8).

Taking into account expressions (2.20) and (2.21), the total torque $M_{з}$, which contributes to the plow's penetration into the soil, can be determined according to the following expression:

$$M_{з} = R_{x}.\, h_{н} + G(a + r_{н}.\cos\beta) + (h_{в} - h_{н}).(G - R_{x}.\tan\beta).(\tan\alpha - \tan\beta)^{-1}.$$

A certain ratio of angle values α and β can cause the length of the segment *BC* to become equal to $h_{н}$. In this case, as follows from expression (2.19), the moment from the force R_x will be equal to zero. When the value of segment *BC* is greater than the value of $h_{н}$, this moment will not change its direction at all. Since force R_z is almost equal to force *G*, but acts on a larger arm (Fig. 2.8), the plow will not sink into the ground in this case.

It follows that the value of the segment *BC* must be less than the design parameter $h_{н}$. And this will be possible if:

$(h_в - h_н).\tan\beta.(\tan\alpha - \tan\beta)^{-1} < h_н$,

or

$h_в.h_н^{-1} < \tan\alpha.\tan_\beta^{-1}$.

Thus, for variant I of setting the aggregating tractor's front linkage, the conditions of ploughing of the front plough into the ground are:

$\alpha \to \alpha_{max.\partial}$ and $\beta \to \beta_{min.\partial}$,

$$h_в.h_н^{-1} < \tan\alpha.\tan_\beta^{-1}. \quad (2.22)$$

When considering variant II of setting the front linkage of the aggregated tractor, keep in mind that in this case, the momentary center of rotation of the front linkage of this tractor takes the position shown in Fig. 2.9.

Regardless of the size of segment C, – be it greater or smaller than parameter L, the torque from the action of force R_z will prevent the plow from sinking into the soil. When determining the value of $M_{(Rz)}$ it is sufficient to operate with the absolute value of the difference ($L - \pi C$):

$M_{(Rz)} = R_z.|L - \pi C|$.

As in the previous version, to find the value of the segment $C\pi$ it is necessary to make the equations of the straight lines SA and FB and solve them with respect to the abscissa X. Let us follow these steps.

For segment SA we have:

$Y_1 = h_в + r_в.\sin\alpha$;

$Y_2 = h_в$;

$X_1 = r_в.\cos\alpha$;

$X_2 = 0$;

$Y = X.\tan\alpha + h_в$.

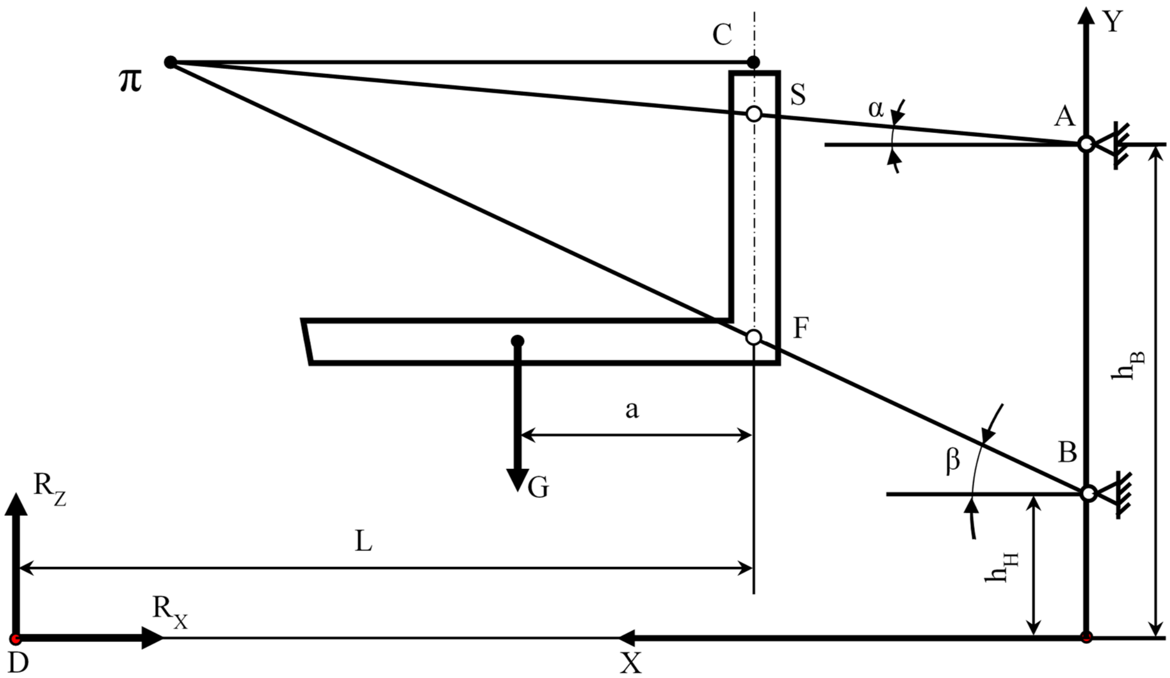


Fig. 2.9 – Diagram of forces acting on the front plough
at the moment of its penetration into the soil in the case of $\alpha < \beta$

For segment *FB*:

$Y_1 = h_н + r_н.\sin\beta$;

$Y_2 = h_н$;

$X_1 = r_н.\cos\beta$;

$X_2 = 0$;

$Y = X.\tan\beta + h_н$.

As a result, we obtain:

$$X = (h_в – h_н).(\tan\beta - \tan\alpha)^{-1}. \quad (2.23)$$

As it can be seen from Fig. 2.9, we can present the abscissa X with the following sum:

$X = \pi C + r_н.\cos\beta$,

thus:

$$\pi C = X - r_н.\cos\beta = (h_в - h_н).(\tan\beta - \tan\alpha)^{-1} - r_н.\cos\beta. \qquad (2.24)$$

In the resulting dependence, it is better to operate with the length of $r_н$ the lower links of the tractor linkage, because their size, in contrast to the length of the central link $r_в$, remains constant during the adjustment of the tractor's front linkage.

Taking into account expression (2.24), to determine the moment from the force R_z we will make the following expression:

$$M_{(Rz)} = P_z \left| L - (h_в - h_н).(\tan\beta - \tan\alpha)^{-1} + r_н.\cos\beta \right|.$$

We can avoid torque $M_{(Rz)}$ only when $L = \pi C$. And in order to do this, the following condition must be met:

$$L = (h_в - h_н).(\tan\beta - \tan\alpha)^{-1} - r_н.\cos\beta. \qquad (2.25)$$

Force R_x on arm $y = X.\tan\beta + h_н$ creates a sinking torque. Taking into account expression (2.23) we have:

$$M_{(Rx)} = R_x.\left[(h_в - h_н).\tan\beta.(\tan\beta - \tan\alpha)^{-1} + h_н \right]. \qquad (2.26)$$

The maximum value of torque $M_{(Rx)}$ will be obtained when both the difference ($\tan\beta - \tan\alpha$), and the value of $\tan\beta$ are minimal. As with variant I, adjustment of the front linkage on the aggregated tractor will only be possible when:

$\alpha \rightarrow \alpha_{max.\partial}$ and $\beta \rightarrow \beta_{min.\partial}$.

Forces R_x and G together create a moment of plow penetration into the soil, which is significantly higher than the torque of plow removal $M_{(Rz)}$. Moreover, the influence of the latter can be completely eliminated if the requirements according to expression (2.25) are satisfied.

Considering the above, we can conclude that in variant II of setting the front linkage of the aggregating tractor, the conditions for plowing into the soil will be as follows:

$\alpha \rightarrow \alpha_{max.\partial}$ and $\beta \rightarrow \beta_{min.\partial}$

$$L = (h_в - h_н).\left(\tan\alpha - \tan\beta\right)^{-1} - r_н.\cos\beta. \qquad (2.27)$$

If variant III is used to set up the front linkage of the aggregated tractor, note that the ordinate of its instantaneous center of rotation (point π, Fig. 2.7) will be greater than the distance $h_н$, but smaller than the distance $h_в$. The momentum from force R_z may be obtained from the following expression:

$$M_{(Rz)} = R_z.\left(\left|L + r_н.\cos\beta - \pi C\right|\right).$$

As it can be seen from Fig. 2.10, segment BC will be equal to:

$$BC = \pi C.\tan\beta. \qquad (2.28)$$

However, on the other hand:

$$h_в - h_н - BC = \pi C.\tan\left|\alpha\right|. \qquad (2.29)$$

Solving together expressions (2.28) and (2.29) with respect to BC, we obtain:

$$\pi C = (h_в - h_н).\left(\tan\left|\alpha\right| + \tan\beta\right)^{-1}. \qquad (2.30)$$

And then the value $M_{(Rz)}$ of the action of force R_z will be equal to:

$$M_{(Rz)} = R_z.\left[\left|L + r_н.\cos\beta - (h_в - h_н).\left[\tan|\alpha| + \tan\beta\right]^{-1}\right|\right].$$

It follows that in order to reduce the magnitude of the torque $M_{(Rz)}$ to the maximum possible, and in fact to zero, it is necessary to fulfill this condition:

$$L = (h_в - h_н).\left(\tan|\alpha| + \tan\beta\right)^{-1} - r_н.\cos\beta.$$

Concerning the torque $M_{(Rx)}$ of the action of force R_x, it can be found from this expression:

$$M_{(Rx)} = R_x.(BC + h_н).$$

Given the expressions (2.28) and (2.30), we finally have the value of the torque $M_{(Rx)}$:

$$M_{(Rx)} = R_x.\left[(h_в - h_н).\tan\beta.\left(\tan|\alpha| + \tan\beta\right)^{-1} + h_н\right]. \tag{2.31}$$

From the analysis of expression (2.31) we see that the maximum value of the torque $M_{(Rx)}$ will be at the minimum allowable value of the angle α and the maximum possible value of the angle β.

Finally, the plow penetration in the soil in the variant III of setting the front attachment of the aggregating tractor will occur under the following conditions:

$\alpha \to \alpha_{min.\partial}$ and $\beta \to \beta_{max.\partial}$

$$L = (h_в - h_н).\left(\tan|\alpha| + \tan\beta\right)^{-1} - r_н.\cos\beta. \tag{2.32}$$

It should be borne in mind that during the working movement of the plowing machine and tractor unit, the vertical coordinates of the “center of resistance” of the plow change (point D, Fig. 2.10), and the direction of the force R_z will also change. However, the options for setting up the front linkage of the aggregated tractor remain identical.

Since the horizontal coordinate of the point D does not change, then for each of the three setting options, the torque $M_{(Rz)}$ is described by the same equation obtained when analyzing the initial phase of plow penetration into the soil. Concerning torque $M_{(Rx)}$, then in comparison with expressions (2.20), (2.26) and (2.31) it is less than the value $R_x.h.2^{-1}$ (where h – plowing depth).

Taking into account that $R_z \cdot 0{,}2.R_x$ [1], the total moment of plow penetration into the soil during its operation, for each of the three options of setting up the front linkage of the connecting tractor, can be found from the following expressions, respectively:

$$1.M_з = G\left(K_1 + r_н.\cos\beta + a\right) + \\ +R_x\left[0{,}2\left(L + r_н.\cos\beta + K_1\right) + h_н - h.2^{-1} - K_1.\tan\beta\right];$$

$$2.M_з = G\left(K_2 + r_н.\cos\beta - a\right) + \\ +R_x\left[0{,}2\left(L + r_н.\cos\beta - K_2\right) + h_н - h.2^{-1} + K_2.\tan\beta\right];$$

$$3.M_з = G\left(K_3 - r_н.\cos\beta - a\right) + \\ +R_x\left[0{,}2\left(L + r_н.\cos\beta - K_3\right) + h_н - h.2^{-1} + K_3.\tan\beta\right]. \tag{2.33}$$

where

$K_1 = (h_в - h_н).(\tan\alpha - \tan\beta)^{-1}$;

$K_2 = (h_в - h_н).(\tan\beta - \tan\alpha)^{-1}$;

$K_3 = (h_в - h_н).(\tan\alpha + \tan\beta)^{-1}$.

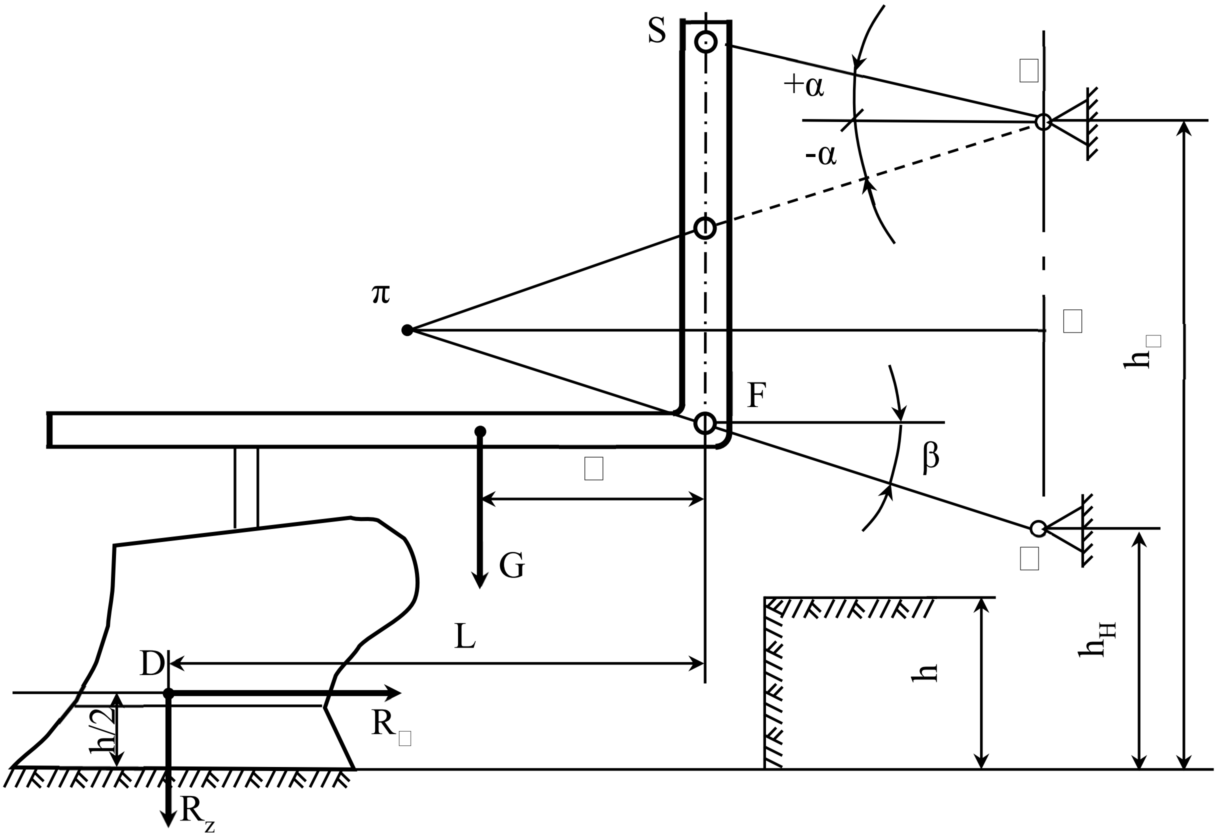


Fig. 2.10 – Diagram of forces, which determine plow penetration in the soil in the third variant of setting the tractor's front linkage

In the system of equations (2.33) force R_x can be calculated using this formula:

$R_x = k_o . B_n . h,$

where k_o – specific traction resistance of the plow;

B_n – working width of the plow.

So that even at the maximum plowing depth h_{max} = 0,30 m angle β is greater than zero degrees, it is necessary that in all considered variants of setting up the front attachment of an aggregating tractor for a front plow, the following condition is fulfilled:

$l_o > h_н + h_{max} .2^{-1},$

where l_o – distance from the plow's bearing surface to the attachment point of connection of the lower links of the aggregating tractor's front linkage.

Since for the wheeled aggregate tractor brand HTZ-120, the value of $h_н$ = 0,62 m, then $l_о$ > 0,77 m.

For the prototype of our two-bodied front plow, the distance from the bearing surface of the plow to the point of attachment to the lower links of the aggregating tractor front linkage equals $l_о$ = 0,84 m.

At the moment of putting into the soil of the plow, adjusted according to variant I, the minimum permissible $\beta_{min.\partial}$ value of angle β can be determined by this expression:

$$\beta_{min.\partial} = \arcsin\left[(l_o - h_н).r_н^{-1}\right].$$

In this case we have $\beta_{min.\partial}$ = 15°.

From the condition defined by expression (2.31) it follows that:

$$\alpha_{max.\partial} > \arctan(h_в.\tan\beta_{min.\partial}.h_н^{-1}).$$

Since for the wheeled aggregate tractor HTZ-120, the value of $h_в$ = 1,27 m, then $\alpha_{max.\partial}$ should be greater than 28°. For further calculations we assume $\alpha_{max.\partial}$ = 30°.

At known values of angles $\beta_{min.\partial}$ and $\alpha_{max.\partial}$ the height of the riser of the connection triangle of the plow ($S_п = SF$, see Fig. 2.8) can be obtained through this expression:

$$S_п = h_в - h_н - r_н.\sin\beta_{min.\partial} + r_н.\cos\beta_{min.\partial}.\tan\alpha_{max.\partial}.$$

In this case, after substitution of known values and calculations it was found that the height of the connection riser $S_п$ will be equal to $S_п$ = 0,89 m.

Note that when the equilibrium of the plow and the wheeled aggregating tractor is considered only in the longitudinal-vertical plane, the tilt of the power vehicle in the transverse-vertical plane can be ignored [89].

However, its wheels must be shown on the drawing so that they are below the surface of the field at a distance that is equal to half the plowing depth h, that is $h.2^{-1}$ (Fig. 2.11). In this case, the actual values of the angles of inclination of the lower β and center α link of the tractor front linkage for version I of the front plow setup can be found from the following expressions:

$$\beta = \arcsin\left[(l_o - h_н - h.2^{-1}).r_н^{-1}\right],$$

$$\alpha = \arctan\left[(S_n - h_в + h_н + r_н.\sin\beta).(r_н.\cos\beta)^{-1}\right]. \qquad (2.34)$$

It should be noted that for all analyzed variants the value of the angle β will be the same, and the value of the angle α, on the contrary, different.

We will determine the value of angles α and β, using the expression (2.34). Thus, for the plowing depth $h = 0{,}3$ m, the values of these angles will be equal: $\beta = 5^o$, $\alpha = 21^o$. Taking into account the initial design data for the aggregation of a two-bodied front plow (G= 2,95 kN; $h_в$ = 1,27 m; $h_н$ = 0,62 m; $r_н$ = 0,82 m; a = 0,5 m; k_o = 70 kN·m^{-2}; B_n = 0,7 m; h = 0,3 m; L = 0,6 m), from the first expression of the system (2.33) we find that the sinking torque $M_з$ in variant I the setting of the front plowing tool is 25,0 kN·m.

For variant II for the setting of the front linkage at L = 0,6 m, the second condition of the system (2.27) is not satisfied. Based on the first condition, we take the value of the angle α closest to the value of β, but not equal to it – 3°. In this case, from the second equation of the system (2.33) we find that $M_з$ = 31,2 kN·m. At = 4° the value of the sinking torque increases to 55 kN·m. Plow riser height ($S_n = SF$, see Fig. 2.9) s determined by the following expression:

$$S_n = h_в - h_н - r_н.\sin\beta + r_н.\cos\beta.\tan\alpha.$$

In this case, if the values of the angles are: $\beta = 5^{o}$ and $\alpha = 3^{o}$, plow riser height S_n = 0,62 m.

As for variant III of the plow setup, the conditions of the system (2.32) are fulfilled at $\beta = 5^{o}$ and $\alpha = -11^{o}$. In this case, as follows from the third equation of the system (2.33), the sinking torque $M_з$ = 10,0 kN·m, and the height of the plowing tool riser ($S_n = SF$, see Fig. 2.10) will be equal to:

$$S_n = h_в - h_н - r_н.\sin\beta + r_н.\cos\beta.\tan|\alpha| = 0,42 \text{ m}.$$

As a result of comparing the values of torques $M_з$ for all three variants of setting we can conclude that to ensure the sinking of the front plow into the soil, at the initial moment and during its subsequent work, the front linkage of the aggregating tractor must be set so that the following conditions are met (variant II):

$$\alpha_{max.д} \geq \alpha \geq 0; \beta_{max.д} \geq \beta > 0; \alpha < \beta;$$

$$\alpha \to \alpha_{\max.д};\ \beta \to \beta_{\min.д}; \tag{2.35}$$

$$L = (h_в - h_н).(\tan\beta - \tan\alpha)^{-1} - r_н.\cos\beta.$$

2.3. Conditions for rear and front plow equilibrium in the longitudinal-vertical plane

To investigate this issue it is necessary first of all to make schemes of forces that act in longitudinal-vertical plane on the rear-mounted and front-mounted plow. Let us first consider such a scheme for the rear-mounted plow, in the working (sinking) state (Fig. 1.11). In the system of forces shown in this diagram, the unknown values are the reactions $P_{вз}$ and $P_{нз}$ n the links of the rear linkage of the aggregate tractor and the vertical reaction $N_{кз}$ on the support wheel of the mounted rear plow.

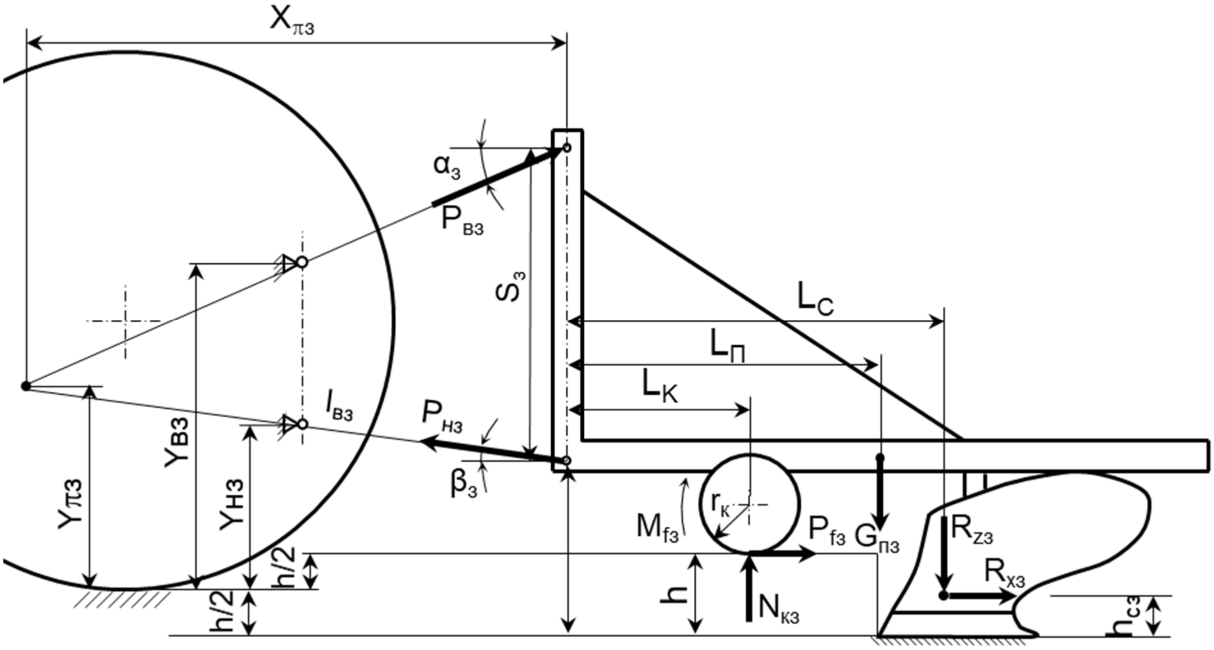


Fig. 2.11 – Diagram of forces acting on the rear plow

To determine these unknown values, it is sufficient to make the following three equations of statics, considering the equilibrium of the given unit in the longitudinal-vertical plane:

$$P_{вз}.\cos\alpha_з - P_{нз}.\cos\beta_з + P_{f\,з} + R_{x\,з} = 0;$$

$$P_{вз}\cdot\sin\alpha_з + P_{нз}\cdot\sin\beta_з + N_{кз} - R_{zз} - G_{пз} = 0; \qquad (2.36)$$

$$G_{пз}\cdot(L_п + X_{\pi з}) - N_{кз}\cdot(L_к + X_{\pi з}) - P_{fз}\cdot\left(Y_{\pi з} - h\cdot 2^{-1}\right) -$$
$$-R_{xз}\left(Y_{\pi з} + h\cdot 2^{-1} - h_{сз}\right) + R_{zз}\cdot(L_с + X_{\pi з}) + M_{fз} = 0.$$

In the system of equations (2.36) the following designations are taken: $P_{fз}$, $M_{fз}$ – force and rolling resistance moment of the plow's support wheel; $R_{xз}$, $R_{zз}$ – horizontal and vertical components of the traction resistance of the plow, related to one body; $G_{пз}$ – gravity of the rear mounted plow; h – plowing depth L_n, $L_к$, $X_{πз}$, $Y_{πз}$, $h_{cз}$, h, L_c – the design parameters adopted in the power diagram of Fig. 2.11.

Considering that, $P_{fз} = f.N_{кз}$; $M_{fз} = f.N_{кз}.r_к$; $R_{zз} = 0,2R_{xз}$; $h_{cз} = h.2^{-1}$, after appropriate transformations, the original system of equations (2.36) can finally be presented in the following form:

$$P_{вз}.\cos\alpha_з - P_{нз}.\cos\beta_з + f.N_{кз} + R_{xз} = 0;$$

$$P_{вз}.\sin\alpha_з + P_{нз}.\sin\beta_з + N_{кз} - 0,2.R_{xз} - G_{пз} = 0; \qquad (2.37)$$

$$\begin{aligned} &G_{пз}\left(L_n + X_{πз}\right) - N_{кз}\left[L_к + X_{πз} - f.\left(Y_{πз} - h.2^{-1} - r_к\right)\right] - \\ &\qquad -R_{xз}\left[Y_{πз} - 0,2\left(L_c + X_{πз}\right)\right] = 0. \end{aligned}$$

To these equations we can add three more equations defining such quantities:

$$X_{πз} = [h_{вз} - h_{нз} + l_{нз}(\tan\alpha_з.\cos\beta_з + \sin\beta_з)].(\tan\alpha_з + \tan\beta_з)^{-1};$$

$$Y_{πз} = X_{πз}.\tan\beta_з + h_{нз} - l_{нз}.\sin\beta_з;$$

$$R_{xз} = k_o.B_{пз}.h,$$

where $r_к$ – radius of the plow's support wheel;

$l_{нз}$ – projection on the longitudinal-vertical plane of the length of the lower links of the rear hitch of the tractor;

$h_{вз}$, $h_{нз}$ – vertical coordinates of attachment points of central and lower links of the rear hitch to the tractor;

$X_{πз}$, $Y_{πз}$ – horizontal and vertical coordinates of the instantaneous center of rotation of the tractor rear linkage;

k_o – coefficient of specific resistance of the plow;

$B_{пз}$ – working width of the plowing tool.

The angles of inclination of the lower $\beta_з$ and central $\alpha_з$ links of the rear linkage of the tractor can be found from the following equations, similar to those obtained in the analysis of options for aggregating the front plow, namely:

$$\beta_з = \arcsin[(h_{нз} - l_o + h.2^{-1}).l_{нз}^{\ -1}],$$

$$\alpha_3 = \arctan[(l_o - h_{вз} - h.2^{-1} + S_з).(l_{нз}.\cos\beta_з)^{-1}],$$

where l_o – distance from the lower (supporting) surface on which the plow moves to the point of connection of the lower links of the aggregating tractor's rear linkage;

$S_з$ – height of rear plow connecting triangle riser.

Calculations of the system of equations (2.37) were performed at the following values of design parameters of the unit in question: $f = 0{,}1$; $r_к = 0{,}23$ m; $h_{вз} = 1{,}27$ m; $h_{нз} = 0{,}61$ m; $l_{нз} = 0{,}82$ m; $k_o = 70$ kN·m^{-2}; $l_o = 0{,}62$ m; $S_з = 0{,}87$ m; $h = 0{,}22$ ... 0,30 m; $B_{пз} = 1{,}75$ (1.05)[2] m; $L_п = 2{,}0$ (1,0) m; $L_к = 1{,}25$ (0,5) m; $L_c = 2{,}0$ (1,0) m; $G_{пз} = 7{,}85$ (4,36) kN.

The analysis of these results, given in Table 2.4, shows that when increasing the depth of plowing with a five-bodied plow from 0,22 to 0,30 m, angle $\beta_з$ tends to decrease while angle $\alpha_з$ on the contrary, to increase. The force acting on the center link of the tractor's rear linkage is reduced by 21 %, and the total force on the lower linkage is increased by about the same amount.

The deepening of the three-bodied plow into the soil increases both force $P_{вз}$ (by 18,5 %), and the total force $P_{нз}$ (by 32,1 %).

Let us perform the same actions when determining the equilibrium conditions in the longitudinal-vertical plane for the front-mounted plow as well. For this

[2] in parentheses are data for three-bodied plows (PLN-3-35), and behind parentheses - for five-bodied plows (PLN-5-35)

purpose, we also make a scheme of forces that act on the front plow in this plane during its working movement (Fig. 2.12). Just as in the previous case, let us make equations of static equilibrium of the given plowing machine-tractor unit. As a result, we have the following three equations of equilibrium:

$$P_{вп}.\cos\alpha_n - P_{нп}.\cos\beta_n + f.N_{кп} + R_{хп} = 0;$$

$$P_{вп}.sin\alpha_n + P_{нп}.sin\beta_n + N_{кп} - 0{,}2R_{хп} - G_{пп} = 0; \qquad (2.38)$$

$$G_{пп}\left(X_{\pi п} - D_n\right) + N_{кп}\left[D_к - X_{\pi п} - f.(Y_{\pi п} - h.2^{-1})\right] - \\ -R_{хп}\left[Y_{\pi п} - 0{,}2\left(X_{\pi п} - D_c\right)\right] = 0.$$

Table 2.4 – Results from the calculations of the system of equations (2.37)

h, m	$\alpha_з$, deg.	$\beta_з$, deg.	PLN-5-35		PLN-3-35	
			$P_{вз}$, kN	$P_{нз}$, kN	$P_{вз}$, kN	$P_{нз}$, kN
0,22	8,1	7,7	3,8	31,8	5,4	22,1
0,25	7,1	8,8	3,0	34,8	6,0	25,0
0,27	6,4	9,5	2,3	36,6	6,2	26,7
0,30	5,3	10,5	0,8	39,0	6,4	29,2

The following three equations can be added to these equations:

$$X_{\pi п} = [h_{вп} - h_{нп} - l_{нп}\left(\tan\alpha_n.\cos\beta_n + \sin\beta_n\right)].\left(\tan\alpha_n + \tan\beta_n\right)^{-1};$$

$$Y_{\pi п} = X_{\pi п}.\tan\beta_n + h_{нп} + l_{нп}.\sin\beta_n;$$

$$R_{хп} = k_o.B_{пп}.h.$$

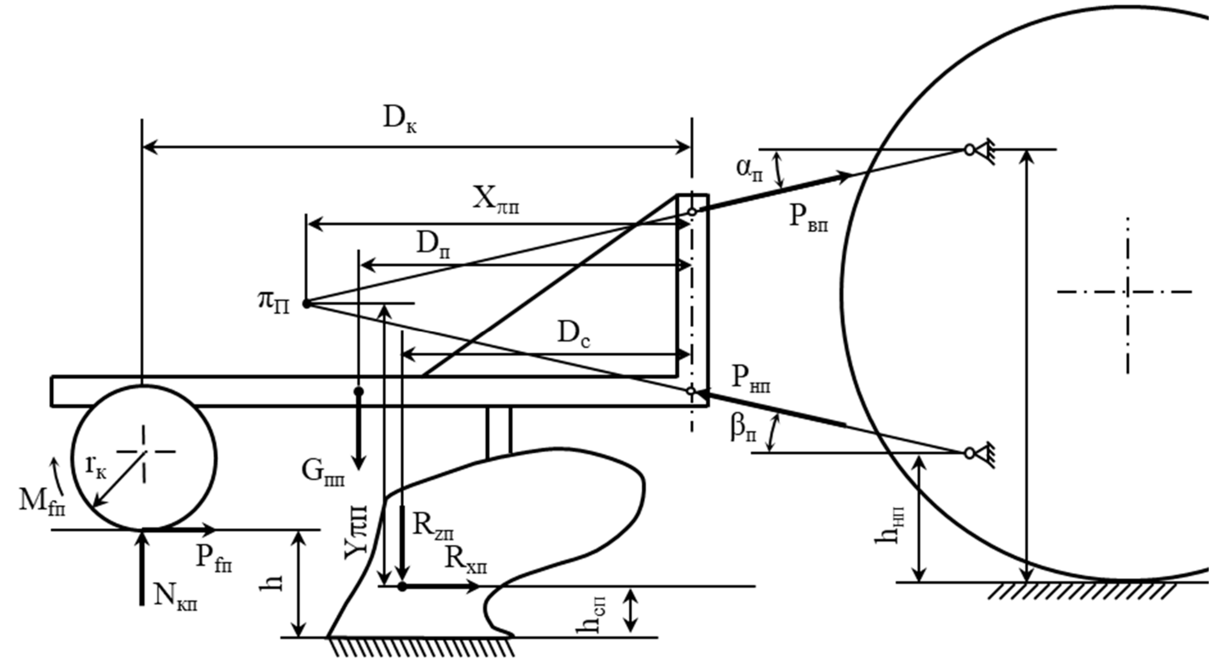


Fig. 2.12 – Diagram of forces acting on the front plow

As we can see, the equations obtained are actually identical to those obtained earlier for the rear-mounted plow. Of the equations obtained, of greatest interest is the value of the angle β_n, which depends on the adopted values of design parameters l_o and $h_{нп}$, as well as on the depth of plowing h, and height S_n of the coupling triangle riser (see Fig. 2.12). Using similar calculations for the rear-mounted plow, the value of the angle β_n can be determined using the following dependence:

$$\beta_n = \arcsin\left[\left(h_{нп} - l_o + h.2^{-1}\right).l_{нп}^{-1}\right].$$

Depending on the defined optimal tilt angle value α_n of the central link of the front linkage of the tractor, the height of the front plow attachment triangle riser S_n can be found from this expression:

$$S_n = h_в - h_н - r_н.\sin\beta_n \pm r_н.\cos\beta_n.\tan|\alpha_n|. \quad (2.39)$$

The last term in expression (2.39) is taken with the sign "–" when adjusting the front linkage of the aggregating tractor according to variant III.

2.4. Influence of the scheme and parameters of a plowing machine on the changes in vertical reactions of the steered wheels of HTZ-120

To solve the problem, consider the equilibrium conditions in the longitudinal-vertical plane of the wheeled aggregate tractor. In this case, it is necessary to replace its connections to the front and rear mounted plows with their reactions. We consider that from the side of the front plow, the force directed along the central link of the front linkage $P_{вп}$ and the resultant forces directed along the lower links $P_{нп}$ act on the wheeled aggregate tractor. From the rear-mounted plow similar forces $P_{вз}$ and $P_{нз}$ act on the latter as well (Fig. 2.13).

To find the unknown vertical reactions on the front N_A and rear $N_в$ wheels of the power vehicle, it is enough to devise a system of equations, in which the sum of the projections of all forces in the vertical plane is equal to zero and so is the sum of moments with respect to point *A*, for example (2.13).

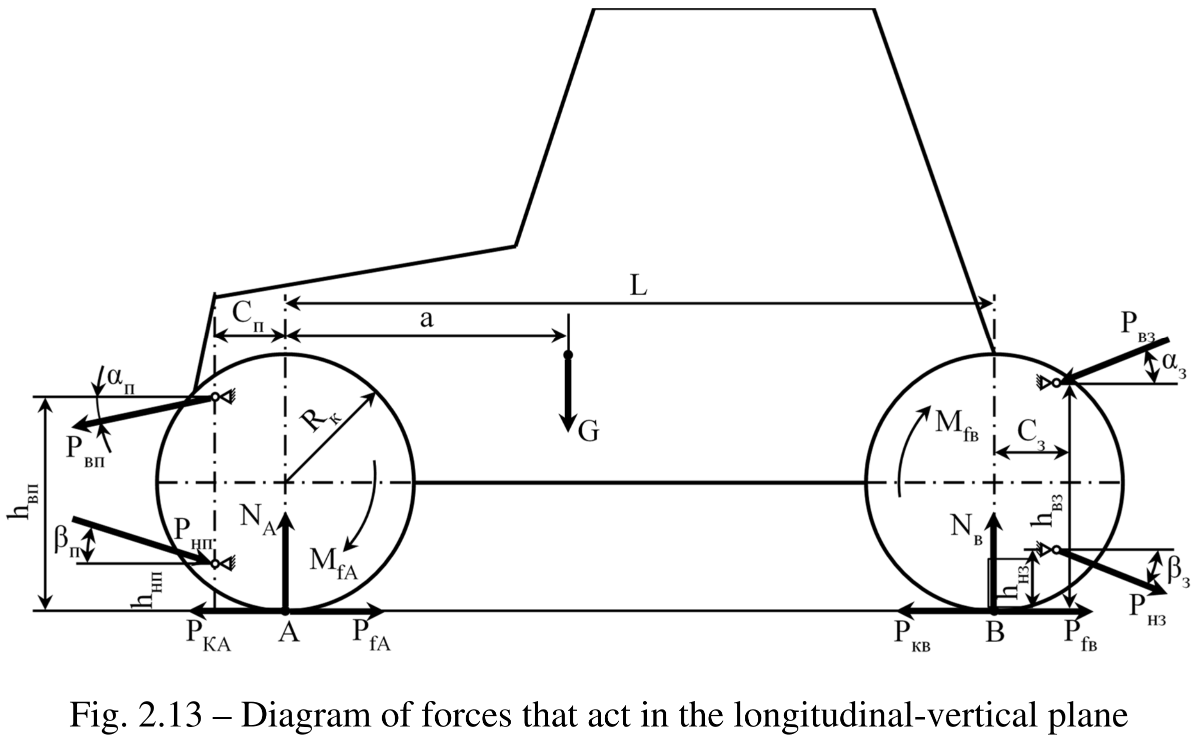


Fig. 2.13 – Diagram of forces that act in the longitudinal-vertical plane on a wheeled aggregate tractor

As a result, we have a system of three equilibrium equations:

$$N_A - G + N_в - P_{вп}.\sin\alpha_п - P_{нп}.\sin\beta_п - P_{вз}.\sin\beta_з - P_{нз}.\sin\beta_з = 0,$$

$$G.a - P_{вп}.cos\alpha_п.h_{вп} - P_{вп}.sin\alpha_п.c_п + P_{нп}.cos\beta_п.h_{нп} - P_{нп}.sin\beta_п.c_п -$$
$$-N_в.L - P_{вз}.cos\alpha_з.h_{вз} + P_{вз}.sin\alpha_з.(c_з + L) + P_{нз}.cos\beta_з.h_{нз} +$$
$$+P_{нз}.sin\beta_з.(c_з + L) + M_{fA} + M_{fB} = 0,$$

where G, a, L – gravity, longitudinal coordinate of center of mass and tractor base, respectively;

M_{fA}, M_{fB} – rolling resistance moments of the front and rear tractor axles, respectively;

$\alpha_п$, $\beta_п$, $\alpha_з$, $\beta_з$, $h_{вп}$, $h_{нп}$, $h_{вз}$, $h_{нз}$, $c_п$, $c_з$ – design parameters, shown on Fig. 2.13.

The rolling resistance moments of the axles of the wheeled tractor are found from the following expressions [115]:

$$M_{fA} = f.N_A.R_к,$$

$$M_{fB} = f.N_B.R_к, \tag{2.41}$$

where $R_к$ – rolling radius of the tractor wheels.

Given the expressions (2.41), the system of equations (2.40) will have the following form after the appropriate transformations:

$$\left.\begin{aligned} &N_A - G + N_B - P_{вп}.\sin\alpha_п - P_{нп}.\sin\alpha_п - P_{вз}.\sin\alpha_з - P_{нз}.\sin\beta_з = 0, \\ &G.a + f.R_к\left(N_A + N_B\right) - N_B.L - P_{вп}.K_1 + P_{нп}.K_2 - P_{вз}.K_3 + P_{нз}.K_4 = 0, \end{aligned}\right\}$$

where:

$K_1 = \cos\alpha_п.h_{вп} + \sin\alpha_п.c_п$;

$K_2 = \cos\beta_п.h_{нп} - \sin\beta_п.c_п$; (2.42)

$K_3 = \cos\alpha_з.h_{вз} - \sin\alpha_з.(c_п + L)$;

$K_4 = \cos\beta_з . h_{нз} + \sin\beta_з .(c_з + L).$

In stationary position, the gravity of the tractor HTZ-120 (81 kN) is distributed so that its front axle is 51 kN (63 %), and the rear axle is 30 kN (37 %).

The combined solution of systems (2.36) and (2.42) shows that aggregating the tractor with a rear-mounted plow PLN-5-35 unloads its front axle by 8,6 ... 9,6 % and adds to the load on his rear axle by 25,6 ... 36,3 %. The increase of the mounting force of tractor weight in this case is 3,3 ... 6,0 kN. The intensity of reduction of the vertical reaction on the front tractor wheels when changing the plowing depth from 0,22 to 0,30 m is significantly less than its (reaction) increase on the rear propulsors (Table 2.5).

Table 2.5 – Vertical reactions on the front N_A and rear N_B axles of the aggregated tractor

h, m	N_A, kN	N_B, kN	Additional load, kN
0,22	46,5	37,7	3,2
0,25	46,4	38,8	4,2
0,27	46,3	39,7	5,0
0,30	46,2	40,8	6,0

When combining this power vehicle with a two-bodied front plow and a three-bodied rear plow (scheme 2 + 3), the installation of the lower links of the front linkage at a negative angle leads, as previously established [16], to unloading of the tractor's front wheels (Fig. 2.14).

At the same time, with a constant value of the angle $\beta_n = 2^o$ the additional loading of the front axle of the power vehicle takes place only for variants II and III when setting up its front linkage. For variant I, when α_n is greater than 10^o, in addition, the support wheel of the front plow is completely unloaded. When $\alpha_n = 5^o$ and $\beta_n = 5^o$, or $\alpha_n = 10^o$ and $\beta_n = 2^o$, the coupling force of the tractor weight for a plowing machine-tractor unit operating under the scheme (2 + 3) is almost doubled (11,6 kN) in comparison with that of a similar unit, but operating under the scheme (0 + 5).

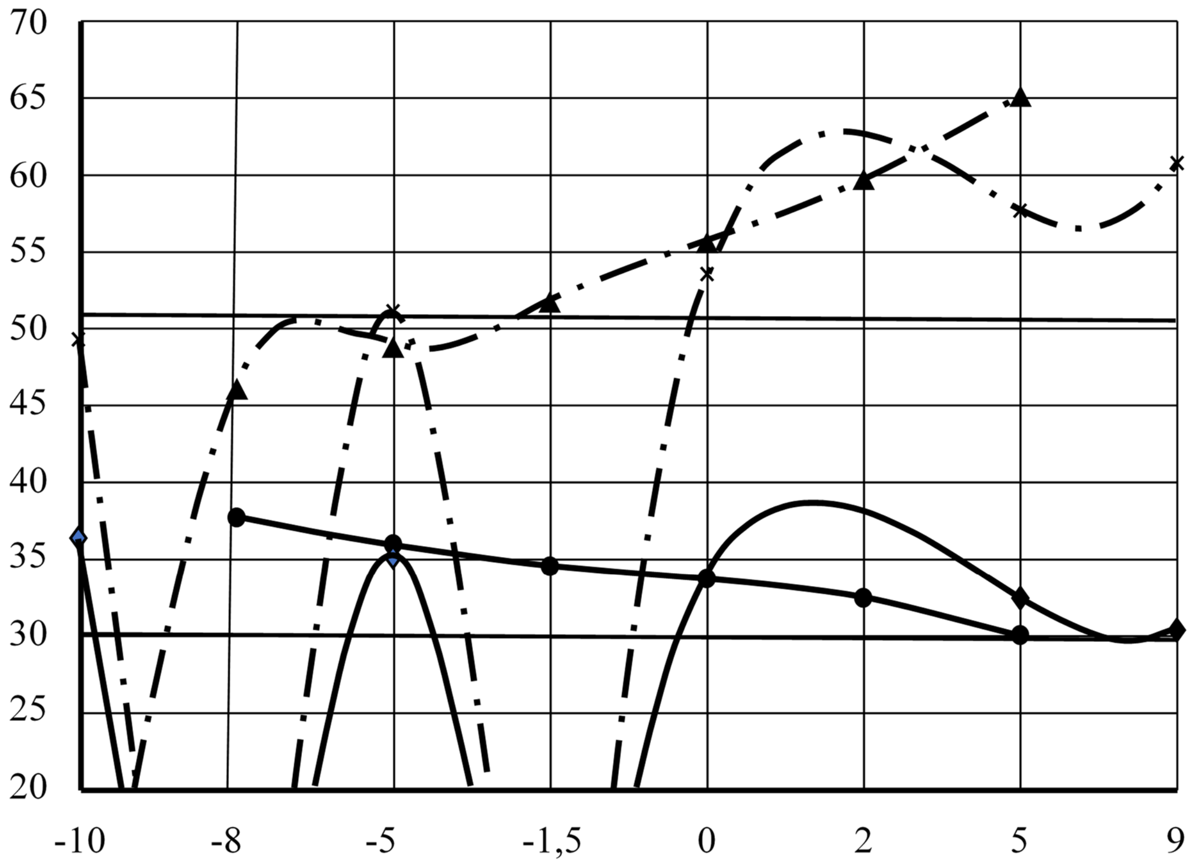


Fig. 2.14 – Dependence of the vertical reactions on the front (– • –) and rear (——) axles of a wheeled tractor HTZ-120:
x, ◊ - $\beta_п$ = const =2°; Δ, o - $\alpha_п$ = const =5°

Although increasing the distance between the front support wheel of the plow and the connecting triangle reduces the vertical load on the front wheels of the aggregating tractor, the change N_B (as well as the increase N_A) are insignificant. On this basis, when choosing the location of these wheels, design constraints should be taken into account in this case.

It should be noted that the adjustment of the front linkage of the aggregating tractor in accordance with variant III, as we have seen, contributes to the additional loading of the tractor, but significantly worsens the kinematics of the lifting of the front tool in transport position [16]. In order to eliminate this drawback, the front linkage of the power tool must be equipped with a device that automatically sets the angle of the central link: during operation – in accordance with variant III, and when raising the tool to the transport position – in accordance with variants I or II.

2.5. General provisions and assumptions made when modelling vertical vibrations of a plow and tractor unit

It is convenient to consider the model of operation of a plowing machine-tractor unit on the basis of an aggregated tractor HTZ-120 as a dynamic system, based on its response to the input perturbation. In this case, disturbances are shocks and impacts caused mainly by irregularities in the supporting surface of the field and uneven traction resistance of front-mounted and rear-mounted plows. In further studies, we will take into account both factors equally, although some researchers believe that the unevenness of the arable terrain has a more significant impact on the dynamics of vertical oscillations of the dynamic system than the unevenness of the traction resistance of the technological part of the plowing machine-tractor unit.

The quality of the dynamic system processing of input variables depends on its characteristics, namely the scheme and structural, kinematic and other parameters. Therefore, the correct choice of the latter in terms of the desired stability of motion provides the plowing machine with an optimal transformation of the incoming disturbances on it.

It is known that the transformational properties of a dynamical system can be expressed with the help of some operator. For example, GOST 21878-76 presents the operator as an algorithm for converting input influences into output variables, and on this basis, defines its most general and fundamental characteristics when analyzing or synthesizing the scheme and parameters of a plowing machine and tractor unit.

The basic operators for dynamic systems are transfer functions and frequency characteristics. It is argued that such operators provide the most complete and physical representation of the unit's response to various perturbations, as well as the transient and steady-state processes of its operation [4].

Theoretical determination of transfer functions and frequency response requires a system of appropriate differential equations that relate output variables to input perturbations, that is, a mathematical model of the process under study. At this stage, it is reasonable to consider it as a system of linear equations. Such an idealization is in many cases quite effective for designing and investigating complex agricultural units, the dynamics of motion of which is not yet sufficiently studied. In this case, it provides an opportunity to physically comprehend the result obtained and to gain experience in the design of new machine-tractor units [4].

In the process of solving the optimization problems of linear stationary dynamic system parameters, we will use amplitude and phase frequency characteristics as operators. To do this, as emphasized above, it is necessary to compose a system of appropriate differential equations, develop transfer functions based on them, and then calculate and analyze the amplitude-frequency and phase-frequency characteristics for both perturbations we have adopted.

It should be emphasized that even in the linear interpretation, the mathematical model of a plowing machine-tractor unit based on an aggregating tractor of HTZ-120 type, working according to the “push-pull” scheme, is a system of rather complex differential equations. To simplify their composition it is necessary to take the following provisions and assumptions.

1. In the general case, when driving with a plow, the machine tractor HTZ-120 performs translational vertical (bouncing) and angular longitudinal (galloping) interconnected movements. Since the main position of the front and rear mounted mechanisms of the aggregating tractor during work is "floating", the angular oscillations of plows, due to their relatively short length, can be neglected. Any noticeable turn of the plowing tool in the longitudinal-vertical plane will occur only if the plow unit overcomes significantly high irregularities of the field surface.

2. It is advisable to make separate equations describing the vertical vibrations of the plowing machine-tractor unit on the basis of an aggregate tractor HTZ-120 for each link of the considered dynamic system. The influence of front-mounted and rear-mounted plows on the aggregating tractor is expressed by the main moments and components of the main vectors of forces. The latter are brought to the centre of the axles of the front and rear wheels of the tractor HTZ-120 (Fig. 2.15).

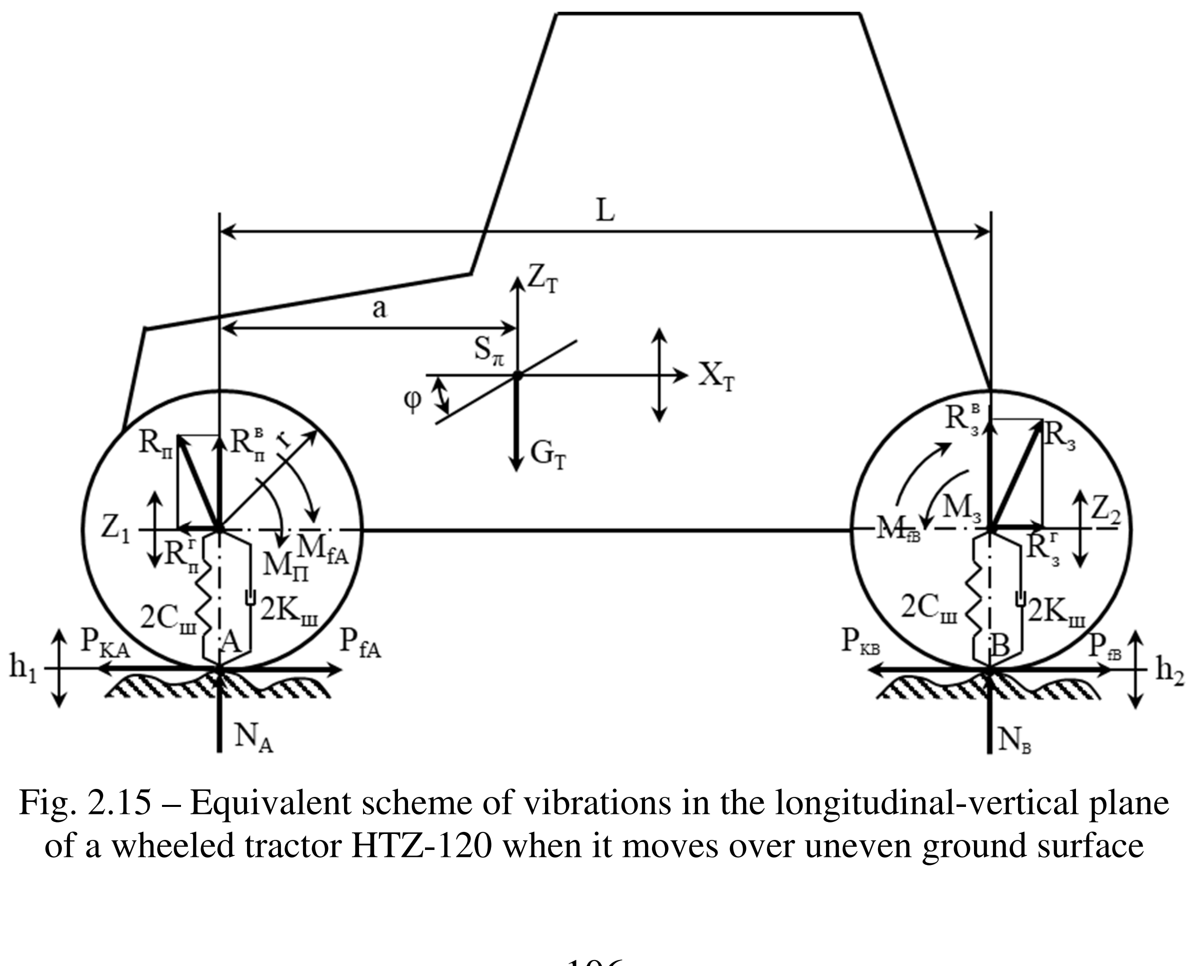


Fig. 2.15 – Equivalent scheme of vibrations in the longitudinal-vertical plane of a wheeled tractor HTZ-120 when it moves over uneven ground surface

3. When formulating the equations of motion of the dynamic system under consideration, we assume that the fluctuations of the traction resistance of plows have no significant effect on the forward speed of the machine-tractor unit, so it is assumed that it is constant. This assumption is explained as follows. The coefficient of variation (V) is often used as an indicator of the variability of a particular parameter. Variability is considered to be insignificant if the value V does not exceed 10 %, average if it is greater than 10 % but less than 20 %, and significant if V is 20 % ore more [30].

Many experimental studies have shown that practically for the average variability of traction resistance of plows, the coefficient of variation of forward speed of arable machine-tractor units has not exceeded 5 %. The relative error in determining the average speed has been less than 1 %. It follows that the assumption about the uniformity of the working motion of a plowing machine-tractor unit is quite correct.

4. With an accuracy acceptable for practice, we can assume that the wheels of the tractor HTZ-120 maintain constant contact with the surface of the agricultural background while driving on the field, which is a random stationary ergodic function of the path [4].

5. Tilts of the aggregating tractor in the longitudinal-transverse plane are taken into account, based on the fact that when compiling expressions for the main moments and main vectors of forces acting on the power vehicle from the aggregated front and rear plows, the first is located in relation to the second without tilting on the surface, which is lower than the field surface by half the plowing depth. This assumption is widely used in studies of the dynamics of movement of plowing units with single rear-mounted plows [89].

6. If the amplitude of the vertical oscillations is insignificant, the resistance forces in the tires of the running wheels of the tractor may be taken as proportional to the oscillation speed, and the characteristics of the elastic elements are considered linear [23].

2.6. Mathematical model of the tractor's forced vertical vibrations

Let us represent a real power vehicle, a wheeled integral tractor HTZ-120, in the form of a dynamic model, by devising its equivalent diagram (Fig. 2.15). In this case, in the equivalent diagram, the wheeled integral tractor is represented by the front and rear axles, to which the two left and right wheels of each axle (the so-called "equivalent" front and rear tractor wheels) are mounted. Since each of the designated "equivalent" pneumatic wheels is the same, they (thanks to the pneumatic tires) are represented as elastic-damping models with the same stiffness coefficients $2C_ш$ and the same damping coefficients $2K_ш$.

Each wheel marked on the diagram in the lower parts is in contact at the points A and B (respectively front and rear wheels) with irregularities of the ground surface, marked as heights of irregularities – h_1 and h_2, respectively. At the designated contact points A and B driving forces are created in each of the wheels – P_{kA} and P_{kB}, directed towards the tractor and positioned horizontally. At the same points, also horizontally, but directed in the opposite direction to the direction of travel of the wheeled tractor, drag forces P_{fA} and P_{fB} are also created.

At the center of the tractor mass, indicated by the point S_m, the force of its gravity is applied G_m. The longitudinal base of the tractor is marked – L, and the distance from its center of mass to the axle of the front wheels – a.

The effect of a front-mounted plow on a given aggregating tractor is expressed by the main torque M_n and the main vector R_n, whose vertical $R_n^в$ and horizontal $R_n^г$ components are concentrated on the axle of the tractor's front wheels. To the rear wheel axis of the tractor there is a vertical $R_з^в$ and horizontal $R_з^г$ components of the main vector of force $R_з$ and main torque $M_з$, which act on the aggregating tractor from the rear-mounted plow.

The computational dynamic model of this wheeled tractor has two degrees of freedom: the vertical displacement of its center of mass Z_m and angular vibrations

φ of the frame. These coordinates can be considered generalized coordinates in the study of this dynamic system.

The generalized coordinates Z_m and φ are connected to the vertical movements of points A and B by the following dependences (Fig. 2.15):

$$Z_m = \left[Z_1.(L - a) + Z_2.a \right].L^{-1},$$

$$\tan\varphi = \left(Z_2 - Z_2 \right).L^{-1}, \tag{2.43}$$

where Z_1, Z_2 – vertical displacements of the front and rear axles of the wheeled tractor, respectively;

L, a – base and longitudinal coordinate of the center of mass of the power vehicle.

Since at small angular displacements $\tan\varphi \approx \varphi$, then:

$$\varphi = \left(Z_2 - Z_2 \right).L^{-1}. \tag{2.44}$$

Thus, the transition from generalized coordinates Z_m and φ to the generalized coordinates Z_1 and Z_2 is carried out.

Let us further determine the kinetic energy of the dynamical system under consideration.

The expression to calculate the kinetic energy T_m of the vertical vibrations of the tractor is as follows:

$$T_m = \left(M_m.\dot{Z}_m^{\ 2} + J_m^{\ 2}.\varphi^2 \right).2^{-1}, \tag{2.45}$$

where M_m, J_m – mass and moment of inertia of tractor HTZ-120.

By differentiating expressions (2.43), (2.44) by time t and substituting the values of the corresponding derivatives into expression (2.45), after transformations we finally obtain:

$$T_m = \left(D_1.\dot{Z}_1^{\,2} + 2.D_2.\dot{Z}_1.\dot{Z}_2 + D_3.\dot{Z}_2^{\,2} \right).2^{-1},$$

where:

$$D_1 = \left[M_m.(L-a)^2 + J_m \right].L^{-2},$$

$$D_2 = \left[M_m.a.(L-a) - J_m \right].L^{-2},$$

$$D_3 = \left[M_m.a^2 + J_m \right].L^{-2}.$$

The potential energy E_m of the tractor is equal to the work of the elastic forces of its front and rear axles. These forces are functions of the corresponding deflections of the tractor wheel tires. For such elastic elements as pneumatic tires of a power vehicle, the deflections calculated from the static equilibrium position of the dynamic system can be expressed as:

$$Z_1 - h_1,$$

$$Z_2 - h_2,$$

where h_1, h_2 – height of unevenness of the ground surface under the front and rear tractor wheels (see Fig. 2.15).

Time-variable values h_1 and h_2 – are exactly the perturbations that act on the agro-background side, and which were discussed above.

Considering the above, the expression for finding the potential energy E_m of the tractor is as follows:

$$E_m = C_ш \left(Z_1^2 - 2.Z_1.h_1 + h_1^2 + h_2^2 + Z_2^2 - 2.Z_2.h_2 + h_2^2 \right),$$

where $C_ш$ – tractor pneumatic tire stiffness of the tractor wheels.

Dissipative function F_m of energy dissipation is defined through the drag forces, which are proportional to the velocities of movement [121]:

$$F_m = K_ш\left(\dot{Z}_1^2 - 2.\dot{Z}_1.\dot{h}_1 + \dot{h}_1^2 + \dot{Z}_2^2 - 2.\dot{Z}_2.\dot{h}_2 + \dot{h}_2^2\right),$$

where $K_ш$ – damping coefficient of pneumatic tires of tractor wheels.

The differential equations of tractor vibrations are written in the Lagrange form of the 2nd kind [121]:

$$\frac{d}{dt}\left(\frac{\partial T_m}{\partial \dot{q}_i}\right) - \frac{\partial T_m}{\partial q_i} + \frac{\partial E_m}{\partial q_i} + \frac{\partial F_m}{\partial \dot{q}_i} = Q_i \tag{2.46}$$

where q_i, Q_i – generalized coordinates and corresponding generalized forces; $i = \overline{1,2}$.

Since the kinetic energy T_m depends only on the velocity and does not depend on the generalized coordinate q_i, we will have:

$$\frac{\partial T_m}{\partial q_i} = 0,\ i = \overline{1,2}$$

Partial derivatives of kinetic energy T_m on the generalized velocities will be, respectively, equal to:

$$\frac{\partial T_m}{\partial \dot{Z}_1} = D_1.\dot{Z}_1 + D_2.\dot{Z}_2,$$

$$\frac{\partial T_m}{\partial \dot{Z}_2} = D_2.\dot{Z}_1 + D_3.\dot{Z}_2$$

Partial time derivatives t from the last two expressions will be respectively equal to:

$$\frac{d}{dt}\left(\frac{\partial T_m}{\partial \dot{Z}_1}\right) = D_1.\ddot{Z}_1 + D_2\ddot{Z}_2,$$

$$\frac{d}{dt}\left(\frac{\partial T_m}{\partial \dot{Z}_2}\right) = D_2.\ddot{Z}_1 + D_3\ddot{Z}_2.$$

Partial derivatives of the potential energy *Em* on the displacements will be equal to:

$$\frac{\partial E_m}{\partial Z_1} = 2.C_{ш}.(Z_1 - h_1),$$

$$\frac{\partial E_m}{\partial Z_2} = 2.C_{ш}.(Z_2 - h_2).$$

Finally, the partial derivatives for the dissipative function F_m of energy dissipation by generalized velocities will be equal to:

$$\frac{\partial F_m}{\partial \dot{Z}_1} = 2.K_{ш}.(\dot{Z}_1 - \dot{h}_1),$$

$$\frac{\partial F_m}{\partial \dot{Z}_2} = 2.K_{ш}.(\dot{Z}_2 - \dot{h}_2).$$

To determine the generalized forces Q_i let us give this dynamic system a possible displacement δZ_1. Rear tractor axle displacement in this case will be $\delta Z_1 = 0$.

The active forces and moments that perform work on the specified possible movement of the system are (see Fig. 2.15): G_m, $R_n^в$, M_n and $M_з$. Forces $P_{кA}$, P_{fA} and $R_n^г$ do not perform at this displacement, since they are directed perpendicularly to the possible displacement δZ_1. Force N_A does not perform either, because its point of application (point *A*, Fig. 2.15) when carrying out the said displacement δZ_1 remains static. For the same reason, the torque M_{fA} does not perform either because it is the product of the force P_{fA} of the dynamic radius *r* of the wheels.

Let uscalculate the amount of work δA all active forces and torques on the possible movement of the point A perform (see Fig. 2.15):

$$\delta A = M_з.\delta\varphi - M_n.\delta\varphi - R_n^в.\delta Z_1 + G_m.\delta Z_m.$$

Since $\delta\varphi = \delta Z_1.L^{-1}$, for $\delta Z_2 = 0$, according to expression (2.43), δZ_m is equal to:

$$\delta Z_m = \delta Z_1.(L - a).L^{-1}$$

then

$$\delta A = [M_з.L^{-1} - M_n.L^{-1} - R_n^в + G_m(L - a).L^{-1}]\delta Z_1 .$$

From this we obtain the value of the generalized force on the generalized coordinate Z_1:

$$Q_{Z1} = [M_з - M_n - R_n^в.L + G_m.(L - a)].L^{-1}.$$

Similarly, we determine the second generalized force, by the generalized coordinate Z_2:

$$Q_{Z2} = (M_n - M_з - R_з^в.L + G_m.a).L^{-1}.$$

Finally, by substituting the values of the corresponding partial derivatives and generalized forces obtained above into the system of equations (2.46), we obtain a system of differential equations, which represents a mathematical model of forced vibrations of the wheeled aggregating tractor HTZ-120 in the longitudinal-vertical plane:

$$\left.\begin{aligned} A_{11}.\ddot{Z}_1 + A_{12}.\dot{Z}_1 + A_{13}.Z_1 + A_{14}.\ddot{Z}_2 = B_{11}.\dot{h}_1 + B_{12}.h_1 + B_{13}, \\ A_{21}.\ddot{Z}_2 + A_{22}.\dot{Z}_2 + A_{23}.Z_2 + A_{24}.\ddot{Z}_1 = B_{21}.\dot{h}_2 + B_{22}.h_2 + B_{23}, \end{aligned}\right\} \quad (2.47)$$

where:

$$A_{11} = [M_m(L - a)^2 + J_m].L^{-2} ;$$

$A_{21} = (M_m.a^2 + J_m).L^{-2}$;

$A_{12} = 2K_ш$;

$A_{22} = A_{12}$;

$A_{13} = 2C_ш$;

$A_{23} = A_{13}$;

$A_{14} = 2[M_m.a.(L - a) - J_m].L^{-2}$;

$A_{24} = A_{14}$;

$B_{11} = B_{21} = A_{12}$;

$B_{12} = B_{22} = A_{13}$;

$B_{13} = [M_з - M_n - R_n^в.L + G_m.(L - a)].L^{-1}$

$B_{23} = (M_n - M_з - R_з^в.L + G_m.a).L^{-1}$

Next, let us determine the vertical components of the main vectors and main moments of forces acting on the wheeled aggregating tractor from the front and rear plows.

As for the front plowing tool, in the system of equations (2.47) the corresponding force and torque are denoted as $R_n^в$ and M_n. The value of force $R_n^в$ can be found from the sum of projections of all forces, acting on the front plow in the longitudinal-vertical plane. From Fig 2.12 it follows that:

$$R_n^в = G_{nn} - N_{кn} + R_n^з - P_n^в.\sin\alpha_n - P_n^н.\sin\beta_n .$$

Concerning the main torque M_n, it can be found from the sum of the moments of all forces with respect to the point lying on the axis of the tractor front wheels:

$$\begin{aligned} M_n = M_{fn} - G_{nn}.(D_n + l_{нn}.\cos\beta_n + c_n) + N_{kn}(D_k + l_{нn}.\cos\beta_n + c_n) - \\ -R_{зn}.(D_c + l_{нn}.\cos\beta_n + c_n) - P_{fn}(R_к - h.2^{-1}) - R_{xn}.R_к + P_n^в.\sin\alpha_n.c_n + \\ +P_n^в.\cos\alpha_n.(h_{вn} - R_к) + P_n^н.\cos\beta_n(R_к - h_{нn}) + P_n^н.\sin\beta_n.c_n \end{aligned}$$

Using the diagram shown in Fig. 2.11, we similarly find the force $R_з^в$ and torque $M_з$:

$$R_з^в = G_{пз} - N_{кз} + R_{zз} - P_з^в \cdot \sin\alpha_з - P_з^н \cdot \sin\beta_з,$$

$$\begin{aligned} M_з = & -M_{fз} - G_{пз}.\left(L_n + l_{нз}.\cos\beta_з + c_з\right) + N_{kз}\left(L_k + l_{нз}.\cos\beta_з + c_з\right) - \\ & -R_{zз}\left(L_c + l_{нз}.\cos\beta_з + c_з\right) - P_{fз}\left(R_к - h.2^{-1}\right) - R_{xз}.R_к + P_з^в.\sin\alpha_з.c_з + \\ & +P_з^в.\cos\alpha_з.\left(h_{вз} - R_к\right) + P_з^н.\cos\beta_з\left(R_к - h_{нз}\right) + P_з^н.\sin\beta_з.c_з. \end{aligned}$$

After substituting the expressions to determine the main torques M_n and $M_з$ as well as forces $R_n^в$ and $R_з^в$ into the system of equations (2.47), the mathematical model of "push-pull" motion of a plowing unit in the longitudinal-vertical plane will be as follows:

$$\left.\begin{aligned} A_{11}.\ddot{Z}_1 + A_{12}.\dot{Z}_1 + A_{13}.Z_1 + A_{14}.\ddot{Z}_2 = B_{11}.\dot{h}_1 + B_{12}.h_1 + B_{13}.R_x + B_{14} \\ A_{21}.\ddot{Z}_2 + A_{22}.\dot{Z}_2 + A_{23}.Z_2 + A_{24}.\ddot{Z}_1 = B_{21}.\dot{h}_2 + B_{22}.h_2 + B_{23}.R_x + B_{24} \end{aligned}\right\} \quad (2.48)$$

where:

$$A_{11} = \left[M_m\left(L-a\right)^2 + J_m\right] \cdot L^{-2};$$

$$A_{21} = \left(M_m a^2 + J_m\right) \cdot L^{-2};$$

$$A_{12} = 2K_ш;$$

$$A_{13} = 2C_ш;$$

$$A_{23} = A_{13};$$

$$A_{14} = 2 \cdot \left[M_m \cdot a \cdot \left(L-a\right) - J_m\right] \cdot L^{-2};$$

$$A_{24} = A_{14};$$

$$B_{11} = B_{21} = A_{12};$$
$$B_{12} = B_{22} = A_{13};$$

$$B_{13} = R_{к} \cdot L^{-1};$$

$$B_{23} = -B_{13};$$

$$B_{14} = \left[K_{зп} - K_{пп} - R_{п}^{в} \cdot L + G_{т} \cdot (L - a) \right] \cdot L^{-1};$$

$$B_{23} = \left(K_{пп} - K_{зп} - R_{з}^{в} \cdot L + G_{т} \cdot a \right) \cdot L^{-1};$$

$$K_{зп} = -P_{f3} \cdot h \cdot 2^{-1} - G_{пз} \cdot \left(L_{п} + l_{нз} \cdot \cos\beta_{з} + c_{3} \right) + N_{кз} \cdot \left(L_{к} + l_{нр} \cos\beta_{з} + c_{3} \right) -$$
$$-R_{z3} \cdot \left(L_{c} + l_{нпз} \cdot \cos\beta_{з} + c_{з} \right) + P_{з}^{в} \cdot \left[\sin\alpha_{з} \cdot c_{з} - \cos\alpha_{з} \cdot \left(h_{вз} - R_{к} \right) \right] -$$
$$-P_{з}^{н} \cdot \left[\cos\beta_{з} \cdot \left(R_{к} - h_{нз} \right) + \sin\beta_{з} \cdot c_{з} \right];$$

$$K_{пп} = P_{fn} \cdot h \cdot 2^{-1} - G_{пп} \cdot \left(D_{п} + l_{нп} \cdot \cos\beta_{п} + c_{п} \right) + N_{кп} \cdot \left(D_{к} + l_{нп} \cdot \cos\beta_{п} + c_{п} \right) -$$
$$-R_{zn} \cdot \left(D_{c} + l_{нп} \cdot \cos\beta_{п} + c_{п} \right) + P_{п}^{в} \cdot \left[\sin\alpha_{п} \cdot c_{п} + \cos\alpha_{п} \cdot \left(h_{вп} - R_{к} \right) \right] +$$
$$+P_{п}^{н} \cdot \left[\cos\beta_{п} \cdot \left(R_{к} - h_{нп} \right) + \sin\beta_{п} \cdot c_{п} \right];$$

$$R_{п}^{в} = G_{пп} - N_{кп} + R_{п}^{з} - P_{п}^{в} \cdot \sin\alpha_{п} - P_{п}^{н} \cdot \sin\beta_{п};$$

$$R_{вз} = G_{пз} - N_{кз} + R_{z3} - P_{з}^{в} \cdot \sin\alpha_{з} - P_{з}^{н} \cdot \sin\beta_{з};$$

$$R_{x} = R_{xп} + R_{xз}.$$

If in the resulting system of differential equations (2.48) we perform the Laplace transformation [29], we obtain a mathematical model of the dynamics of motion of a plowing machine-tractor unit in the operator record form:

$$\left. \begin{aligned} K_{11}.Z_{1}(s) + K_{12}.Z_{2}(s) = F_{11}.h_{1}(s) + F_{12}.R_{x}(s) + F_{13}; \\ K_{21}.Z_{1}(s) + K_{22}.Z_{2}(s) = F_{21}.h_{1}(s) + F_{22}.R_{x}(s) + F_{23}; \end{aligned} \right\} \qquad (2.49)$$

where:

$$K_{11} = A_{11} \cdot s^{2} + A_{12} \cdot s + A_{13};$$

$F_{11} = B_{11} \cdot s + B_{12};$

$K_{12} = A_{14} \cdot s^2;$

$F_{12} = B_{13};$

$K_{21} = A_{24} \cdot s^2;$

$F_{21} = B_{21} \cdot s + B_{22};$

$K_{22} = A_{21} \cdot s^2 + A_{22} \cdot s + A_{23};$

$F_{22} = B_{23};$

$F_{13} = B_{14};$

$F_{23} = B_{24}.$

$s = \frac{d}{dt}$ – differentiation operator.

The input variables in the system of equations (2.49) are the variations of the heights of the unevenness of the bearing surfaces under the front h_1 and rear h_2 tractor wheels, and total traction resistance R_x of the front and rear plows.

The output parameters of the functioning of the dynamic system under consideration are the amplitudes of vertical oscillations of the front Z_1 and rear Z_2 axles of the wheeled tractor HTZ-120.

The given transfer function is found from the expression [5]:

$$W(s) = \frac{F_m.s^m + F_{m-1}.s^{m-1} + ... + F_1.s + F_o}{C_n.s^n + C_{n-1}.s^{n-1} + ... + C_1.s + C_o}, \tag{2.50}$$

where F_m, C_n – numerator and denominator transfer function coefficients determined by the design and kinematic parameters of the plowing machine;

m, n – powers of equations.

By substituting in (2.50) value $i \cdot \omega$ (where $i = \sqrt{-1}$, a ω – disturbance frequency) instead of s, after transformations, we obtained expressions of the given transfer functions, on the basis of which we calculated the amplitude-frequency characteristics and phase-frequency characteristics of this dynamic system.

In the process of theoretical analysis we compiled:

– the transfer function of the path profile under the front wheels of the tractor relative to the oscillation of its front axle: $W_1(s) = D_1.D^{-1}$;

– the transfer function of the path profile under the rear wheels of the tractor relative to the oscillations of its front axle: $W_1(s) = D_2.D^{-1}$;

– transfer function for the plow drag with respect to the tractor's front axle oscillation: $W_1(s) = D_3.D^{-1}$;

– the transfer function of the traction resistance of the plows relative to the oscillation of the tractor rear axle: $W_1(s) = D_4.D^{-1}$.

Each of the above determinants D ... D_4 we find using methods, presented in [5, 23]:

$$D = \begin{vmatrix} K_{11} & K_{12} \\ K_{21} & K_{22} \end{vmatrix};$$

$$D_1 = \begin{vmatrix} F_{11} & K_{12} \\ 0 & K_{22} \end{vmatrix};$$

$$D_2 = \begin{vmatrix} 0 & K_{12} \\ F_{21} & K_{22} \end{vmatrix};$$

$$D_3 = \begin{vmatrix} F_{12} & K_{12} \\ F_{22} & K_{22} \end{vmatrix};$$

$$D_4 = \begin{vmatrix} K_{11} & F_{12} \\ K_{21} & F_{22} \end{vmatrix}.$$

The solution of these determinants allowed us to obtain expressions for the corresponding transfer functions.

The first transfer function is:

$$W_1(s) = \frac{F_3 \cdot s^3 + F_2 \cdot s^2 + F_1 \cdot s + F_o}{C_4 \cdot s^4 + C_3 \cdot s^3 + C_2 \cdot s^2 + C_1 \cdot s + C_o},$$

where:

$$C_4 = A_{11} \cdot A_{21} - A_{14} \cdot A_{24};$$

$$C_3 = A_{11} \cdot A_{22} + A_{12} \cdot A_{21};$$

$$C_2 = A_{11} \cdot A_{23} + A_{12} \cdot A_{22} + A_{13} \cdot A_{21};$$

$$C_1 = A_{12} \cdot A_{23} + A_{13} \cdot A_{22};$$

$$C_o = A_{13} \cdot A_{23};$$

$$F_3 = A_{21} \cdot B_{11};$$

$$F_2 = A_{22} \cdot B_{11} + A_{21} \cdot B_{12};$$

$$F_1 = A_{23} \cdot B_{11} + A_{22} \cdot B_{12};$$

$$F_o = A_{23} \cdot B_{12}.$$

The second transfer function is:

$$W_2(s) = \frac{F_3 \cdot s^3 + F_2 \cdot s^2}{C_4 \cdot s^4 + C_3 \cdot s^3 + C_2 \cdot s^2 + C_1 \cdot s + C_o},$$

where:

$$F_3 = -A_{14}B_{11};$$

$$F_2 = -A_{14} \cdot B_{22}.$$

The third transfer function is:

$$W_1(s) = \frac{F_2 \cdot s^2 + F_1 \cdot s + F_o}{C_4 \cdot s^4 + C_3 \cdot s^3 + C_2 \cdot s^2 + C_1 \cdot s + C_o},$$

where:

$$F_2 = B_{13} \cdot (A_{21} + A_{14});$$

$$F_1 = B_{13} \cdot A_{22};$$

$$F_o = B_{13} \cdot A_{23}.$$

And, finally, the fourth transfer function is:

$$W_4(s) = \frac{F_2 \cdot s^2 + F_1 \cdot s + F_o}{C_4 \cdot s^4 + C_3 \cdot s^3 + C_2 \cdot s^2 + C_1 \cdot s + C_o},$$

where:

$$F_2 = -B_{13}(A_{11} + A_{24});$$

$$F_1 = -B_{13}A_{11};$$

$$F_o = -B_{13}A_{13}.$$

Thus, the set of intended mathematical actions is completely fulfilled. The next step is to check the correctness of the developed mathematical model of forced vertical vibrations of the wheeled tractor.

2.7. Checking the mathematical model for adequacy

The problem was solved by comparing the theoretical and experimental amplitude frequency characteristics of a plowing unit, operating by the “push-pull” scheme consisting of: a reversible-adjusted aggregating tractor type HTZ-120; a frontal two-bodied plow of the conventional brand PLN-2-35; a rear-mounted four-bodied plow PLN-4-35.

The experimental amplitude-frequency response of this plowing machine - tractor unit $A(\omega)$ was found with the expression [4]:

$$A(\omega) = \frac{\sigma_y}{\sigma_x} \cdot \sqrt{\frac{S_y}{S_x}},$$

where ω – frequency of fluctuations of input and output parameters;

σ_y, S_y – standard deviation and normalized spectral density of the output parameter;

σ_x, S_x – standard deviation and normalized spectral density of the input quantity.

The oscillations of the longitudinal profile of the track were taken as an input value. The initial parameter was the vertical vibrations of the front axle of the HTZ-120 tractor with the characteristics $\sigma_{пм}$ and $S_{пм}$.

When recording fluctuations in the longitudinal profile of the track, the following was taken into account. The movement of the aggregating tractor HTZ-120 as a part of the plowing machine-tractor unit described above is provided by the right wheels in the furrow, and the left wheels on the uncultivated agricultural background. It was assumed a priori that the nature of the vertical oscillations of the left and right side of the power unit may not coincide. With this in mind, both the longitudinal profile of the pre-pass groove of this plowing machine-tractor unit and the profile of the untreated field were taken.

Further, using a personal computer, statistical characteristics were calculated (σ_{np} and S_{np}) for both implementations. The corresponding averaged statistical characteristics of the profile were taken for the analysis. Thus, the possible non-uniform nature of vertical oscillations of the left and right sides of the aggregating tractor was taken into account.

The next stage was to conduct a field experiment with recording the vertical oscillations of the front axle of the aggregating tractor on an oscilloscope tape, to process the data obtained and determe the appropriate statistical characteristics $\sigma_{пм}$ and $S_{пм}$ on a PC and, finally, to calculate of the experimental amplitude-frequency response using this formula:

$$A(\omega) = \frac{\sigma_{пм}}{\sigma_{np}} \cdot \sqrt{\frac{S_{пм}}{S_{np}}} \,. \tag{2.51}$$

Using the above first transfer function of the path profile under the front wheels of the aggregating tractor relative to the oscillations of its front axle, we calculated the theoretical amplitude-frequency characteristic $A_m(\omega)$ of the plowing machine-tractor unit of the same scheme.

The adequacy of the developed mathematical model was judged after comparing the amplitude frequency characteristics $A(\omega)$ and $A_m(\omega)$.

When conducting experimental studies, the average moisture content of the soil in the layer of 0 ... 30 cm was 13,5 %. Its density was in the range of 1,21 ... 1,23 g.cm^{-3}. The clogging of agricultural background was high. Its average value was 406 g.m^{-2}.

Front and rear plows were set to a plowing depth of 27 cm. A plowing machine was moving on the scoring plot with a speed of 1,6 m.s^{-1}.

The oscillations of the averaged longitudinal profile of the treated area of the field, although of relatively high-frequency, were characterized by low energy, since their mean square deviation was only 1,95 cm. From the analysis of the normalized spectral density (Fig. 2.16) we find that the cut-off frequency for this process is about 4 m^{-1}. At the specified speed of this arable machine, it is 6,4 s^{-1} or 1,02 Hz.

Thus, we can say that the bulk of the dispersion of oscillations of the longitudinal profile of the track was concentrated in the frequency range of 0 … 6,4 s^{-1}.

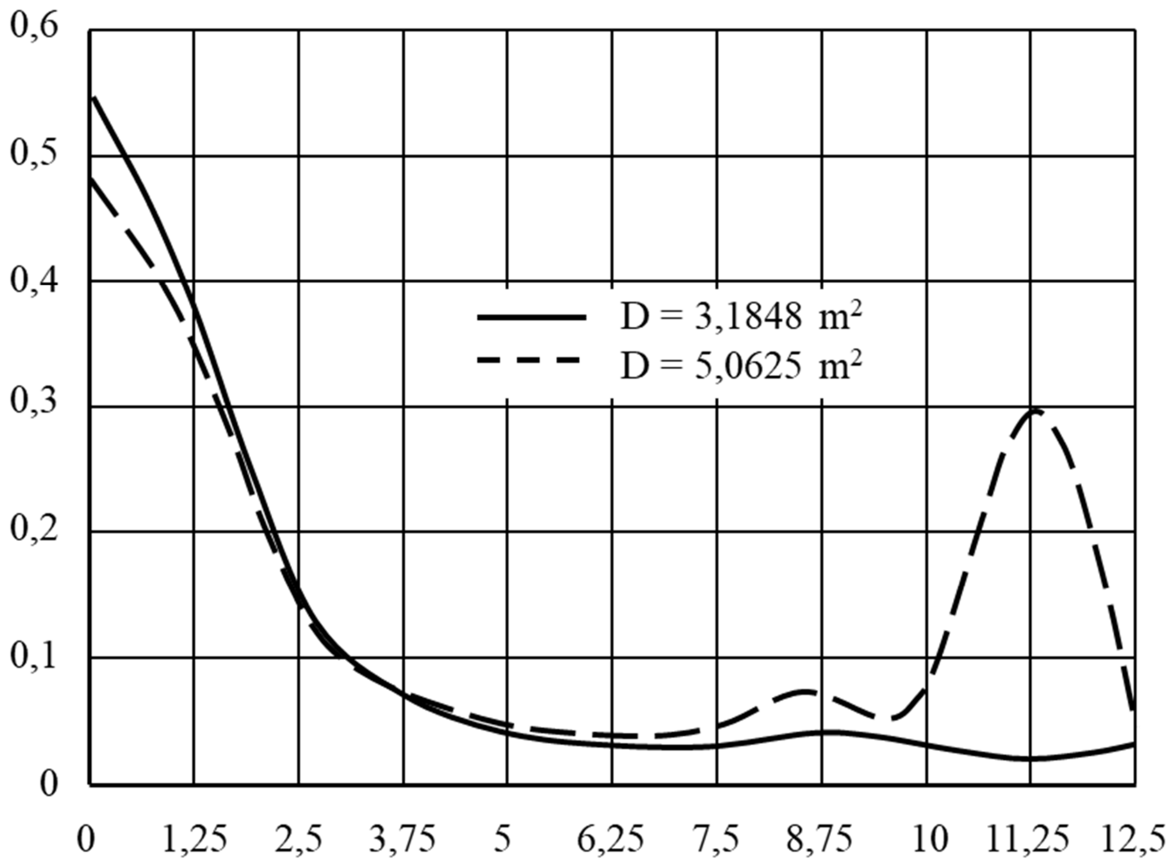


Fig. 2.16 – Normalized spectral densities of oscillations of the longitudinal profile of the track (-------) and the front axle of tractor HTZ-120 (- - - -)

As the analysis of experimental data showed, the main share of dispersion of vertical vibrations of the front axle of aggregating tractor HTZ-120 is concentrated in the same range (see Fig. 2.16). Beyond it, at $\omega = 10,0 \ldots 12,5$ s^{-1} there is a certain increase in the statistical estimate in question.

The mean square deviation of the vertical oscillations of the front axle of the aggregating tractor was 2,25 cm. However, according to the well-known F - Fisher's criterion [30], the zero-hypothesis of equality of the compared variances at the statistical significance level of 0,05 is not rejected.

In other words, with a probability of 95 % it can be argued that the difference between the variance of the longitudinal profile of the track and the front axle of the tractor HTZ-120 is statistically random.

Having determined the required mean square deviations and normalized spectral densities, we calculated the required experimental amplitude-frequency response of the arable machine (Fig. 2.17) by expression (2.51).

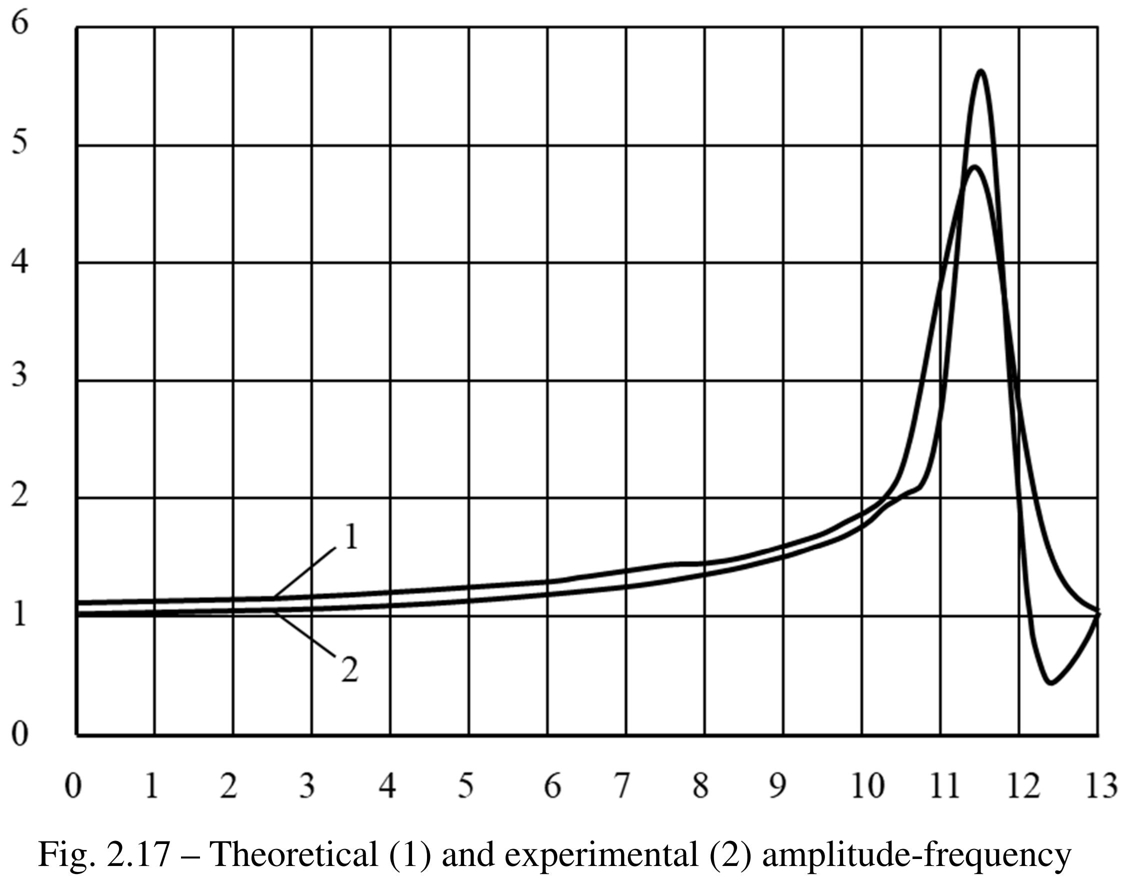


Fig. 2.17 – Theoretical (1) and experimental (2) amplitude-frequency characteristics of a plowing machine-tractor unit

As can be seen from the nature of its course, in the frequency range of 0 ... 6,4 s^{-1} input signal in the form of vertical oscillations of the longitudinal profile of the track, the front axle of the HTZ-120 aggregating tractor reproduced almost without amplification. In general, this process is close to the theoretical one (see Fig. 2.17). Even in a much wider range of frequencies (0 ... 10,0 s^{-1}) the discrepancy between the theoretical and experimental amplitude-frequency characteristics is no more than 12 %.

This coincidence of theoretical and experimental data indicates the high accuracy and suitability of the developed mathematical model of a plowing machine-tractor unit for further theoretical analysis, the results of which should be considered quite reliable.

2.8. Analysis of the smooth running of the HTZ-120 tractor with aggregated front and rear plows

By the nature of the transforming properties, the dynamic system we are considering refers to the tracking system. It is known that for such a system, the desired amplitude-frequency characteristics of the output parameters when it reproduces perturbations in the operating frequency range should be as small as possible (in the ideal case, equal to zero) [4, 5]. The phase shift (or phase-frequency response) in this case should be as large as possible.

The main disturbances that cause vertical movements of the plowing machine in the longitudinal-vertical plane are fluctuations in the longitudinal profile of the track and fluctuations in the traction resistances of the front and rear plows.

As the analysis of the mathematical model shows, perturbations with a frequency of up to 10 s^{-1} are reproduced by the front axle of the wheeled tractor HTZ-120 practically without amplification, both for direct and reverse movements of the aggregating tractor as a part of a plowing machine-tractor unit (Fig. 2.18).

Resonant peaks appear only at frequencies 11 ... 12 s^{-1}. Here $\omega = 11{,}5$ s^{-1} vibration amplitude of the front axle of tractor HTZ-120 during its direct motion is more than 10 times greater than the same vibration amplitude during reversing motion.

As for the phase shift, in the frequency range of 0,5 ... 10,5 s^{-1} it is constant and equal to 360° (Fig. 2.19). This is equivalent to a delay in the response of the front axle of the aggregate tractor to fluctuations in the longitudinal profile of the track from 0,6 to 12,5 s.

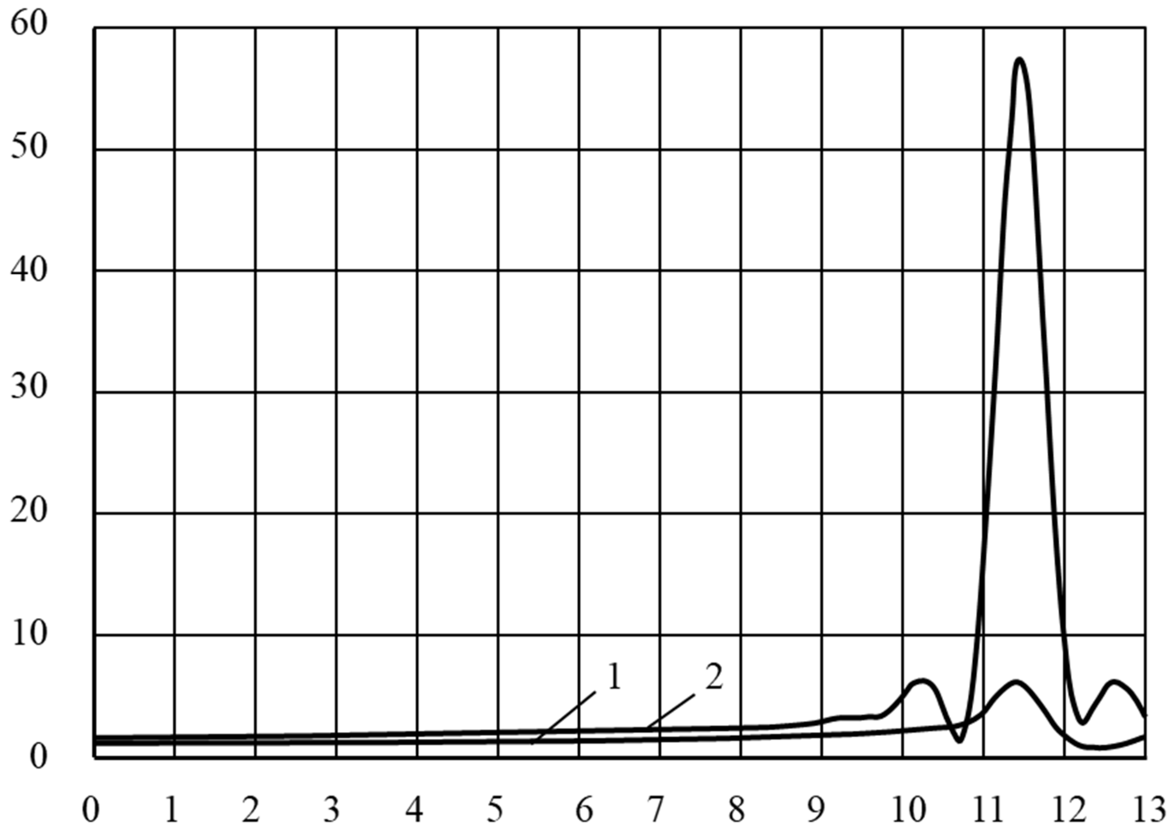


Fig. 2.18 – Vibration amplitude-frequency characteristics
of the front axle of HTZ-120 tractor at its direct (1) and
reversible (2) movements as a part of a plowing machine-tractor unit

At resonant frequencies, the phase shift for both variants of tractor HTZ-120 movement decreases to 200° and even less.

Increasing the stiffness coefficient of tires of the power vehicle, as follows from the analysis of the amplitude-frequency characteristics (Fig. 2.20), leads to the following. At frequencies up to (or approximately) 6,5 s^{-1} the influence of this parameter is practically not felt. At $\omega > 6{,}5$ s^{-1} the increase C_{tir} causes a lowering of the amplitude-frequency characteristics with a simultaneous shift of the resonance peaks towards higher frequencies.

As we see, in terms of desirability of the dynamic system to handle the considered disturbance, increasing the stiffness coefficient of the tires of the tractor HTZ-120 in general is effective.

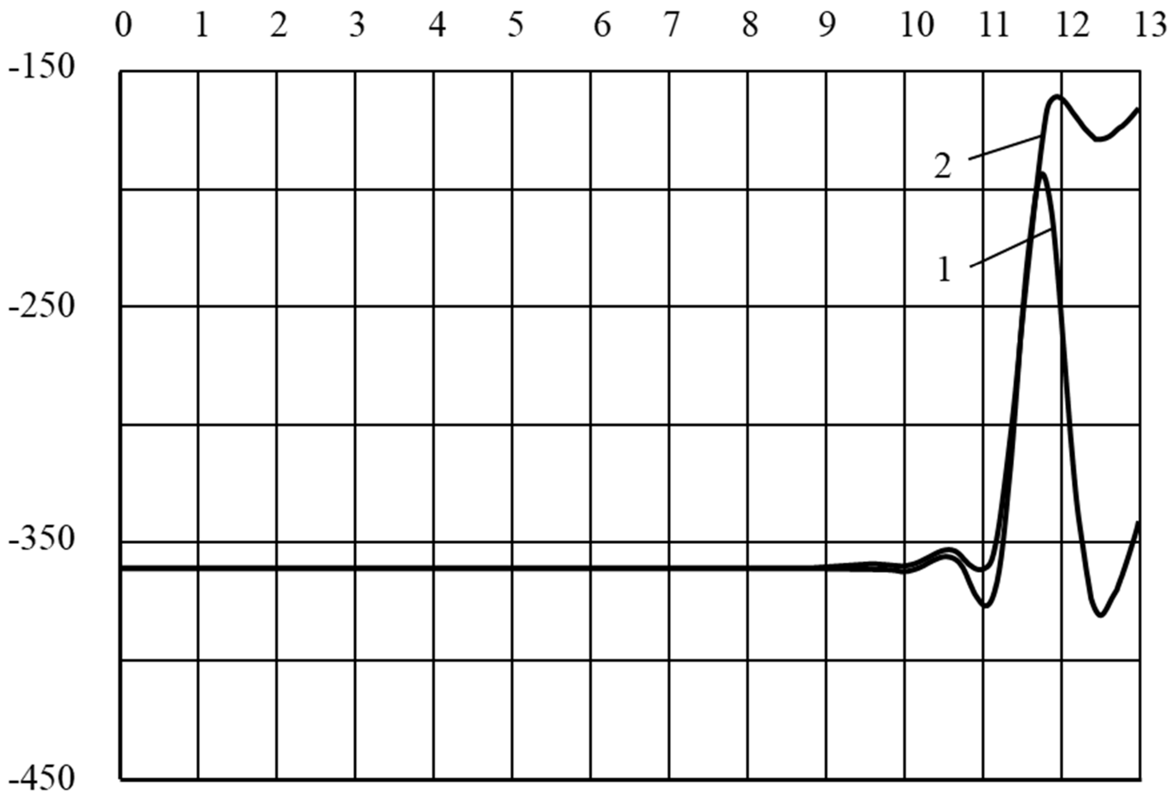


Fig. 2.19 – Phase-frequency characteristics of the front axle of HTZ-120 tractor at its direct (1) and reversible (2) movements as a part of a plowing machine-tractor unit

However, it will be of practical importance only when the longitudinal field profile oscillates with a frequency greater than 6,5 s^{-1}.

In contrast to the parameter C_{tir}, the value of the tire deformation resistance coefficient K_{tir} does not have a significant effect on the nature of the oscillations of the front axle of the HTZ-120 aggregate tractor in the whole range of frequencies considered: from 0,5 to 13,5 s^{-1}.

Oscillations of the longitudinal profile of the track under the tractor's rear wheels perform qualitatively the same, but quantitatively different impact on the vertical movements of the tractor's front axle. In the frequency range 0,5 ... 10,0 s^{-1} oscillations of the tractor bridges can be considered almost mutually independent, because the corresponding amplitude-frequency response is almost at zero level (Fig. 2.21). And again, this is true for both direct and reversible movement of the tractor as part of a plowing machine-tractor unit.

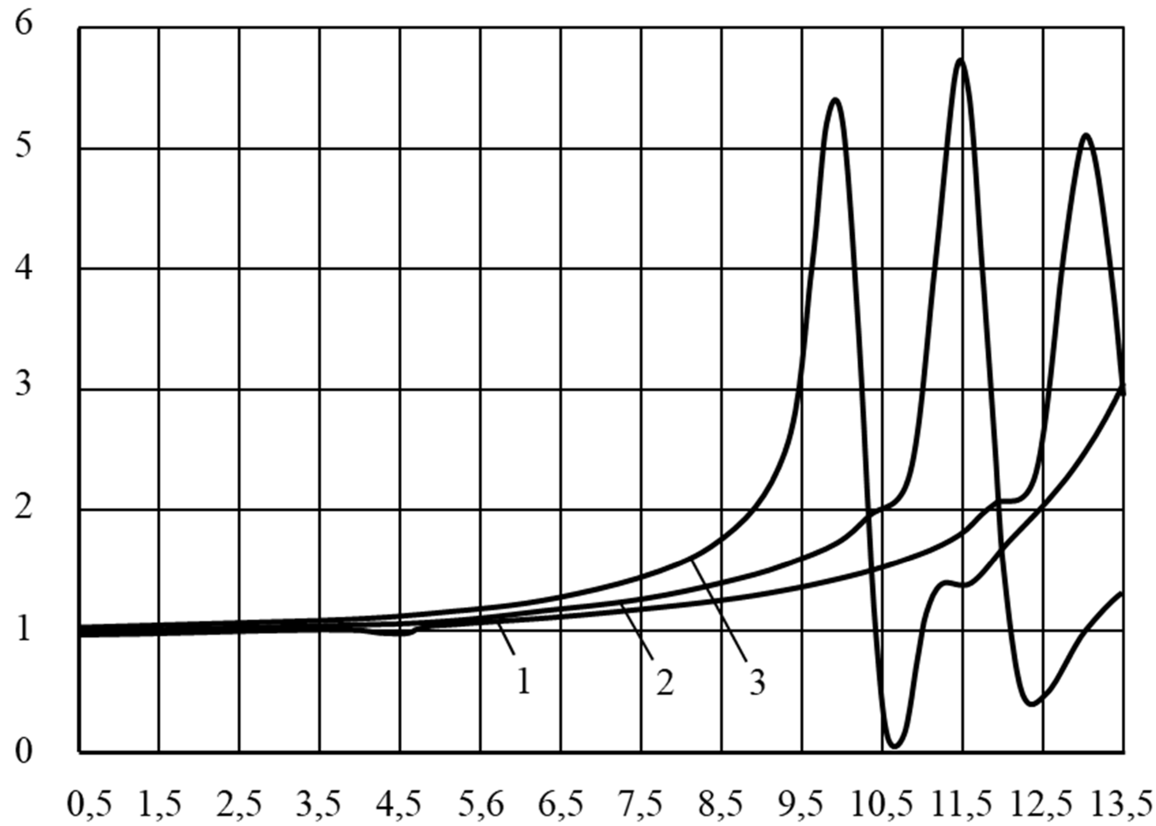


Fig. 2.20 – Amplitude-frequency characteristics of tractor front axle oscillations when reproducing track profile oscillations with different tire stiffness coefficients:
1 – C_{tir} = 250 kN.m^{-1}; 2 – C_{tir} = 350 kN.m^{-1}; 3 – C_{tir} = 450 kN.m^{-1}

Resonance peaks of both amplitude-frequency characteristics occur at the same frequencies as in the variant of perturbation reproduced by the tractor front axle in the form of vertical oscillations of the longitudinal profile of the track under its front wheels.

Another thing is that with the same qualitative character of the ratio of amplitude-frequency characteristics, their maximum values differ not by 10 (Fig. 2.18), but about 1,9 times (Fig. 2.21). This is achieved by increasing the maximum of the amplitude-frequency response when the tractor moves in reverse and decreasing the maximum of the amplitude-frequency characteristics when it moves forward.

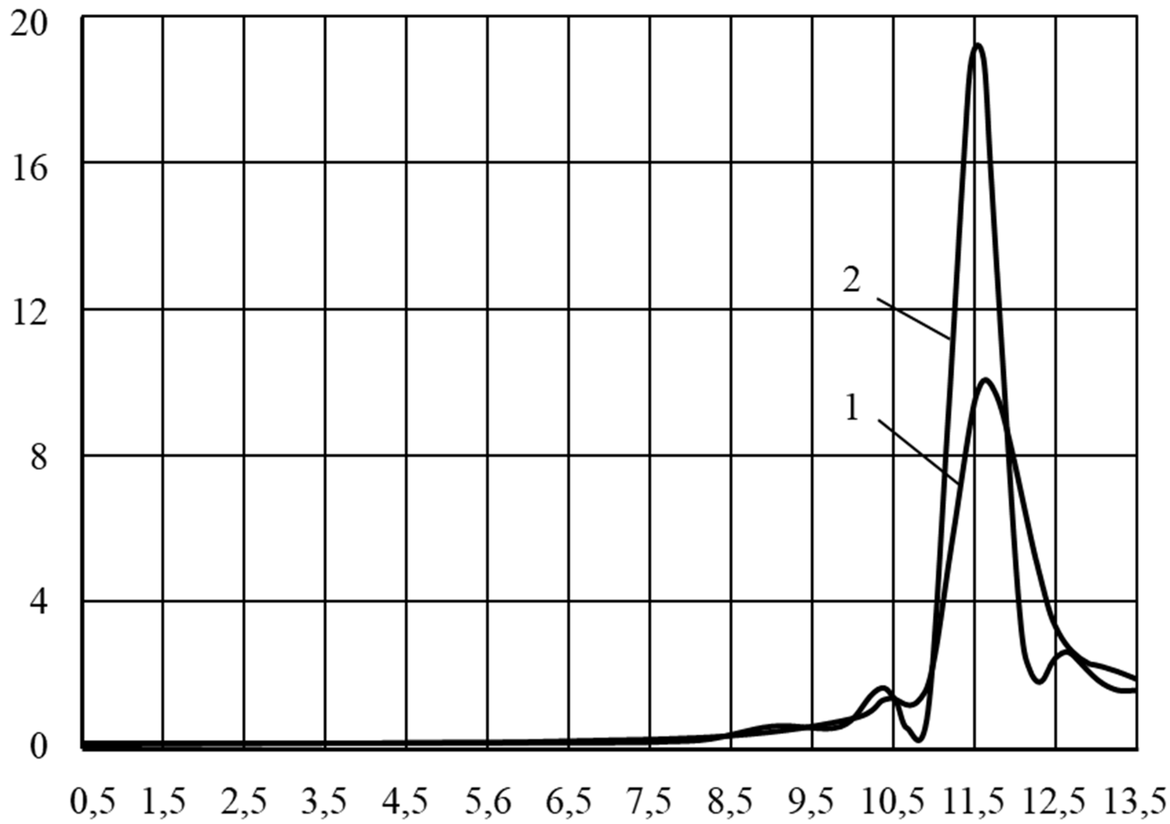


Fig. 2.21 – Amplitude-frequency characteristics of the influence of track profile oscillations under the wheels of the rear tractor axle on the oscillations of its front axle at direct (1) and reversed (2) movements

In other words, the impact of the layout of the power tool as part of the plowing machine-tractor unit on the vertical movements of the front axle caused by fluctuations in the longitudinal profile under its rear thrusters is insignificant. And this result is quite natural, since in the resonant frequency range the oscillations of the tractor bridges, as emphasized above, are mutually independent.

The same fact that in both variants of perturbation resonance peaks of amplitude-frequency characteristics are inherent in the scheme of the plowing machine-tractor unit with direct motion of tractor HTZ-120, is explained by the greater operating mass (and therefore greater inertia) attributable to its front axle (63 %).

Let us consider the influence of the fluctuations of traction resistance of the plows of the investigated plowing machine-tractor unit on the oscillations of the front and rear axles of the HTZ-120 tractor. The main spectrum of dispersion of this parameter in most cases falls on frequencies not greater than 11 s^{-1}.

From the analysis of the amplitude-frequency characteristics obtained, we see that in the frequency range of 0,5 ... 11,0 s^{-1} for each kilonewton of plow traction resistance vibration there is 0,39 to 0,59 mm of vibration amplitude of front axle of HTZ-120 tractor (Fig. 2.22).

Let us take as an example the fact that the mean square deviation of the traction resistance of plows is on average 4 kN [2]. With this in mind, it can be argued that in the frequency range of 0,5 ... 11,0 s^{-1}, oscillations of traction resistance of aggregated plows cause vertical movements of front axle of wheel tractor HTZ-120 with amplitude of no more than 2,4 mm.

A very different picture could take place at resonant frequencies. Thus, at $\omega =$ 12,0 s^{-1}, the oscillation amplitude of the front axle of the HTZ-120 aggregating tractor during its direct motion as a part of the plowing machine-tractor unit can reach almost 60 mm (see Fig. 2.22).

In reverse traffic, it is almost seven times less. It should be emphasized that such a ratio of resonant amplitudes is caused, as in previous cases, by large inertial properties of the tractor front axle during its direct motion as part of the investigated plowing machine-tractor unit.

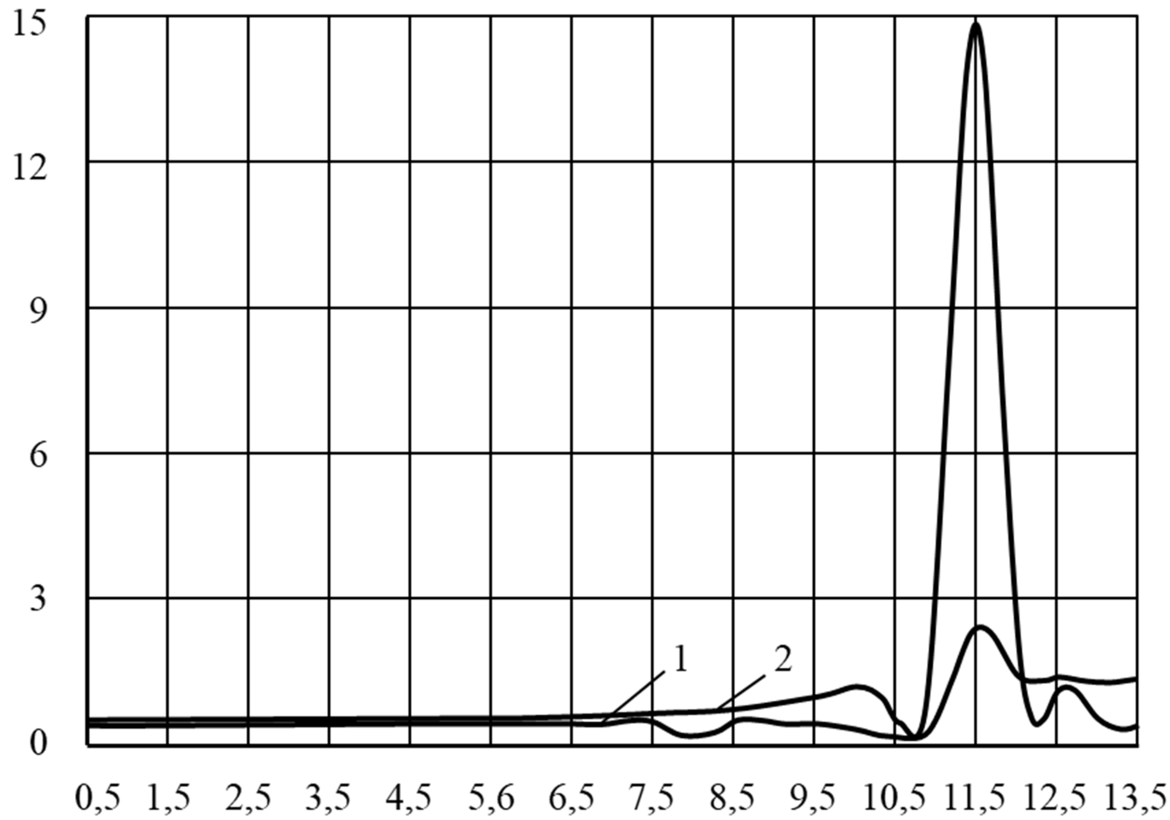


Fig. 2.22 – Amplitude-frequency characteristics of the effect of oscillations of traction resistance of plows it a plowing machine-tractor unit on oscillations of a front axle of an aggregating tractor at its direct (1) and reversed (2) movements

Delayed reaction of the front axle of interconnecting tractor HTZ-120 to oscillations of traction resistance of front and rear plows in the frequency range of 0,5 ... 8,5 s^{-1} is the same for both schemes of plowing machine-tractor units and amounts to 360°. At higher frequencies, the phase shift of the unit with reversible tractor motion decreases (see Fig. 2.23), which is undesirable when these and other disturbing influences are handled by such dynamic systems.

The results obtained by theoretical modeling show that the change in the amplitude of vertical displacements of the rear axle of tractor HTZ-120 under the influence of plow traction resistance fluctuations is qualitatively and practically quantitatively the same as the one shown in Fig. 2.22. At the same time, the coefficient K_{tir} of the tire deformation resistance has no significant effect on the corresponding amplitude-frequency characteristics. When it comes to coefficient C_{tir}, the regularity of its influence on the character of vertical movements of the tractor rear axle during traction resistance fluctuations of plowing machines is qualitatively the same as the one during the longitudinal track profile fluctuations produced by the front axle of HTZ-120 tractor (see Fig. 2.20).

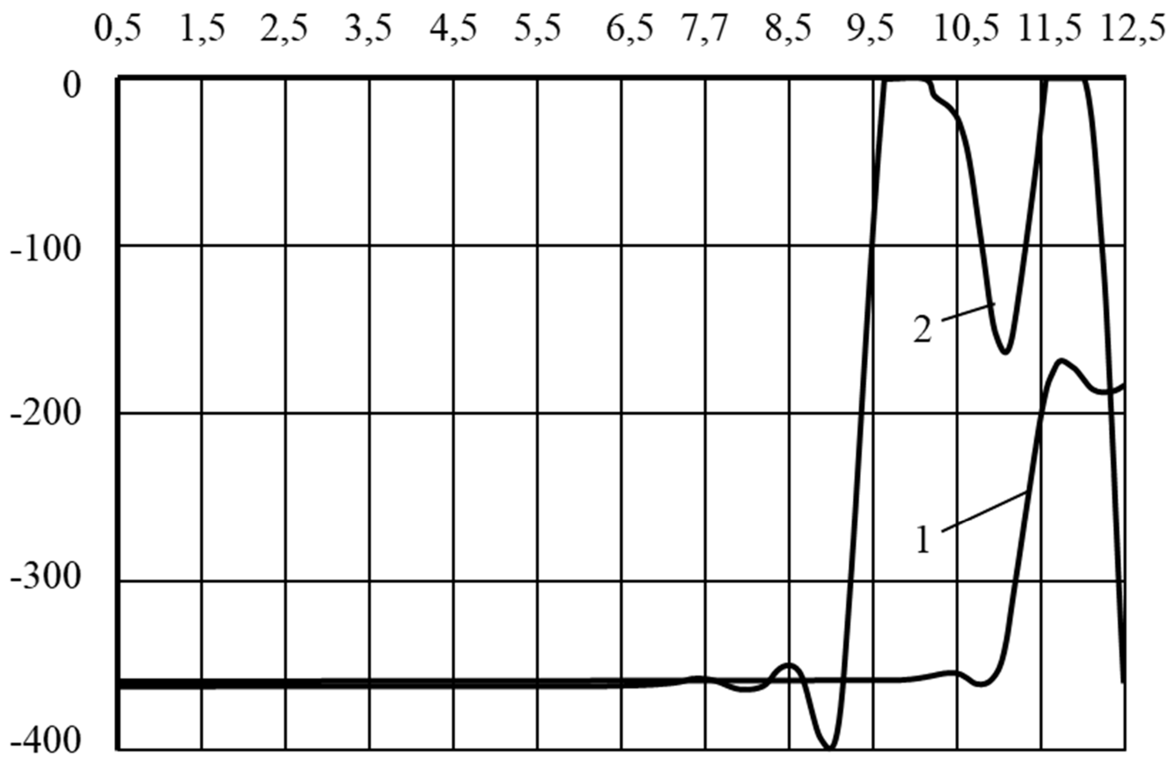


Fig. 2.23 – Phase-frequency characteristics of the influence of oscillations of traction resistance of plows of plowing machine-tractor unit on oscillations of the tractor front axle at its direct (1) and reversed (2) movements

As for the phase frequency characteristics, their analysis shows that the delay of the reaction of the rear axle of tractor HTZ-120 to the fluctuations of traction resistance of aggregated plows in the frequency range of 0,5 ... 9,5 s^{-1} is the same for both schemes of plowing machine-tractor units and is 180° or 0,33 ... 6,28 s^{-1} (Fig. 2.24).

As in the variant with the front axle (see Fig. 2.23), at resonance frequencies the phase shift when a dynamic system handles a disturbance in the form of fluctuations of traction, resistance of front and rear plows of plowing machine-tractor units with reversible motion of aggregating tractor significantly decreases.

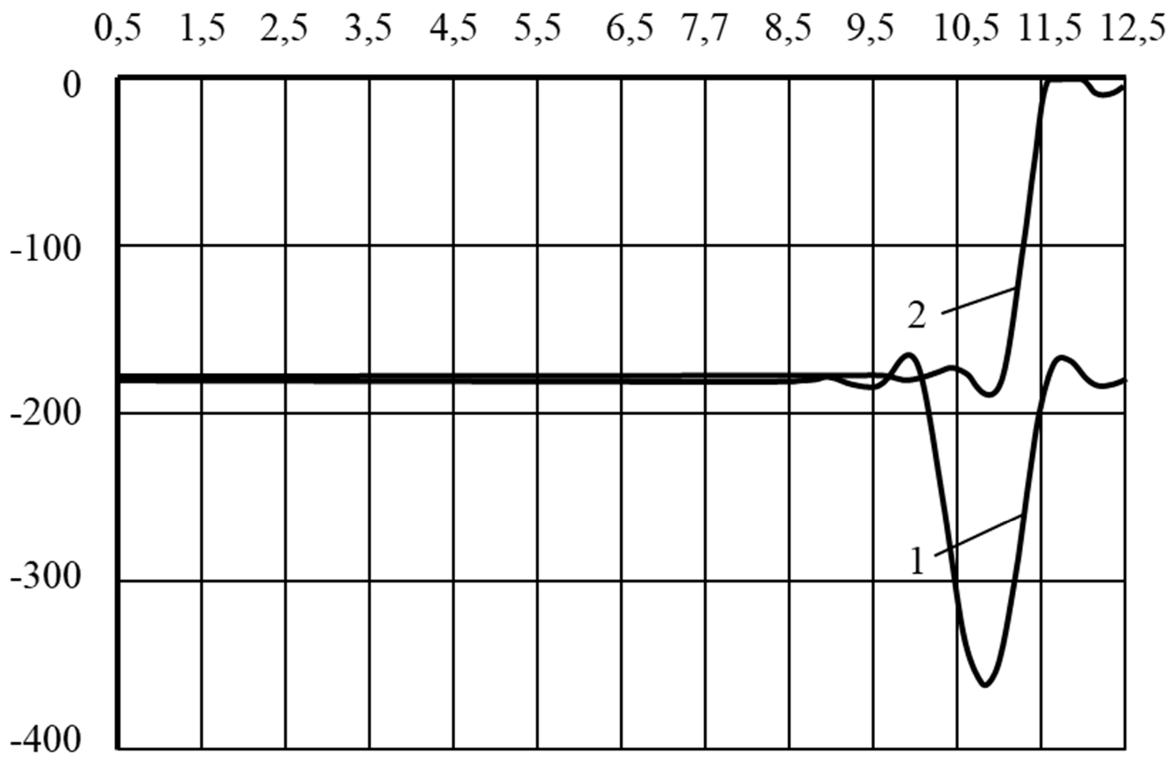


Fig. 2.24 – Phase-frequency characteristics of the influence of oscillations of traction resistance of plows of a plowing machine-tractor unit on oscillations of the tractor's rear axle at its direct (1) and reversed (2) movements

2.9. Influence of the scheme and mode of operation of a plowing machine on its traction and energy performance

During field experimental studies, the wheeled tractor HTZ-120 was configured to run in reverse. Visibility of the front-mounted plow from the tractor driver's cab was quite satisfactory. The operator could see the support wheel and both bodies of the front-mounted plow from the tractor cab without turning his head and torso. As a result, this gave him a good opportunity to trace the furrows of the previous pass of the plowing machine.

Plowing machine-tractor units operating on schemes (0 + 4), (0 + 5) and (2 + 4) moved in 2 and 3 gears of the rear speed range of the aggregate tractor.

All plows were set to the same plowing depth – 27 cm. It turned out that the mean square deviation of traction resistances of plows, judging by the rear-mounted four-bodied plow, in all variants did not exceed 4 kN.

Traction resistance of the front plow was not recorded separately. However, it is logical to assume that the standard of fluctuation of this value is practically the same as that for the rear-mounted plow. In other words, at identical travel speeds (which is quite understandable) and plowing depths with front and rear plows, there is no reason to reject the null-hypothesis about equality of mean square deviations of their traction resistances.

For the rear-mounted four-hulled plough, the traction resistance varied within the range of 20,8 ... 22,8 kN. Coefficient of variation V is 17,5 ... 19,2 %, which indicates the average variability of this process [30].

If traction resistance of two-bodied front plow was (quite possibly) half as much as that of the four-bodied rear plow (i.e. 10,4 ... 11,4 kN), then total resistance of both plows was 31,2 ... 34,2 kN. Based on this, we can say that at a mean square deviation of 4 kN, the variability of traction resistance of the entire plowing machine-tractor unit was close to negligible ($V \leq 10$), as the coefficient of variation of this process was within 11,6 ... 12,8 %.

As it follows from normalized autocorrelation functions of plow traction resistance, correlation time was within 1,15 ... 1,35 s (Fig. 2.25), which is equal to 1,61 ... 1,69 m·s^{-1} for the speed of plow-tractor unit is 1,85 ... 2,29 m.

The main spectrum of dispersion of plow drag oscillations is concentrated in a relatively narrow range of frequencies: 0 ... 10,9 s^{-1} or 0 ... 1,7 Hz (Fig. 2.26).

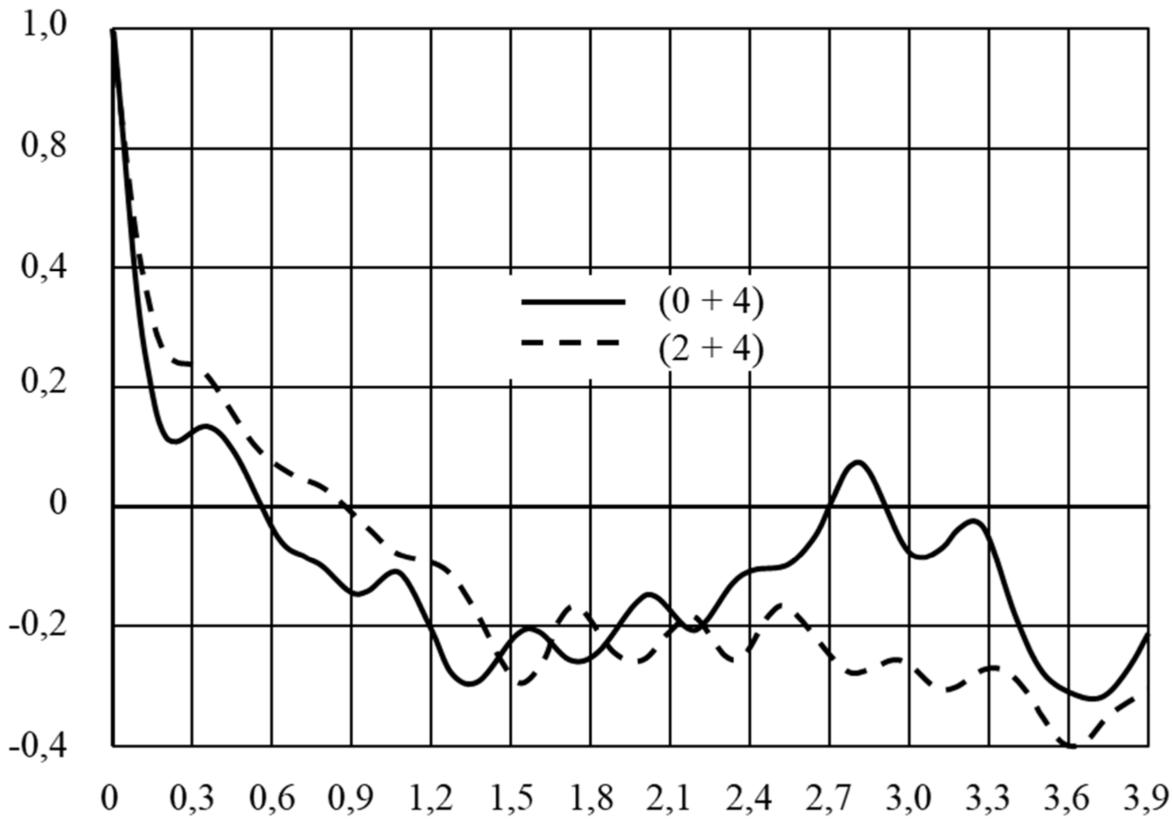


Fig. 2.25 – Normalized autocorrelation functions of oscillations of traction resistance of rear-mounted plows of plowing machine-tractor units operating according to schemes (0 + 4) and (2+4)

This is explained by the conditions of field experimental studies, namely the relatively low density of soil agricultural background, equal to 1,21 ... 1,23 g·cm^{-3}. With a larger value of this parameter and soil moisture at 13,5 %, the basic spectrum of traction resistance dispersion of plows would be in a much wider range of frequencies.

Finally, based on the experimental data, we can conclude that the nature of the vibration of plows of different plowing machine-tractor units on the basis of an aggregate tractor HTZ-120 in these specific experimental conditions was low-frequency. And this, as we will see later, accordingly affected the stability of movement of the plowing tool on the depth of tillage.

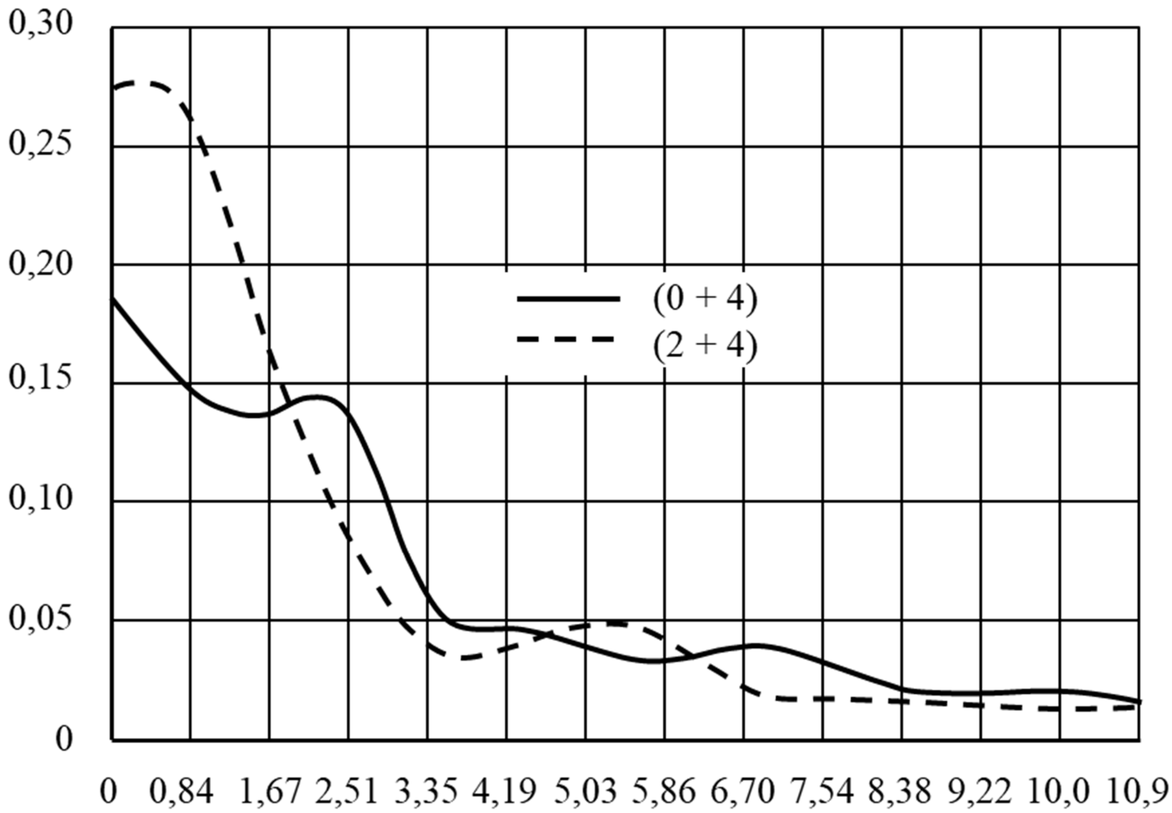


Fig. 2.26 – Normalized spectral densities of traction resistance oscillations of rear-mounted plows of plowing machine-tractor units, operating according to the scheme (0 + 4) and (2 + 4)

According to data of laboratory and field experimental studies it was established that the working width of a plowing machine-tractor aggregate, working according to scheme (2 + 4, Fig. 2.27) was by 21,3 % more than the working width of an aggregate consisting of aggregating tractor HTZ-120 and rear mounted plow PLN-5-35, that is, plowing aggregate, working according to scheme (0 + 5).

As for the operating speed, for the plowing machine-tractor unit operating according to the scheme (0 + 5), it was higher only by 1,4 % – in the second gear, and by 0,06 % - in the third gear (Table 2.6). As a result, the productivity per hour of the main time of the machine working according to the "push-pull" scheme, i.e. (2 + 4) was 20,2 % higher in both traffic modes. If hourly fuel consumption of a plowing machine-tractor unit operating under the scheme (0 + 5) and a plowing machine-tractor unit operating under the scheme (2 + 4) were almost the same (see Table 2.6), then the specific fuel consumption was different. The plowing machine-tractor unit working according to the scheme (2 + 4) was more economical: in the first mode – 18,7 %, in the second mode – 16,5 %. Such efficiency is very significant.

Fig. 2.27 – Wheeled tractor HTZ-120 in aggregate with rear-mounted four-bodied and frontal two-boded plows during field experiments

Compared with a plowing unit, working under the scheme (0 + 4), the use of machine-tractor unit, working under the scheme (2 + 4), can improve productivity for 1 hour of prime time by 33 ... 36 %. As for specific fuel consumption, it is 6,2 % higher in second gear of the aggregate tractor and 12% lower in third gear than in the comparable unit.

Due to practically optimal values of soil density and relatively low moisture content, slipping of the wheeled tractor HTZ-120 in the unit with six plowing bodies did not exceed the agrotechnically permissible level (18 ... 20 %). At the same time the engine load (judging by the hourly fuel consumption) was equal to 99 %. This fact indicates the great prospects of using the tractor HTZ-160 for plowing, which has a more powerful engine than the HTZ-120.

Table 2.6 – Results of the experimental studies of arable machine-tractor units based on the tractor HTZ-120

Composition of the plowing machine - tractor units	V_p 1), $m \cdot s^{-1}$	B_p 2), m	W 3), $ha \cdot h^{-1}$	h 4), cm	δ 5), %	P_{kr} 6), kN	G 7), $kg \cdot h^{-1}$	G_n 8), $kg \cdot ha^{-1}$
HTZ – 120 + PLN-4-35 (0 + 4)	1,47	1,51	0,80	26,5 ±0,5	9,2	15,0	16,6	20,8
	1,69		0,92		11,2	17,8	20,7	22,5
HTZ-120 + PLN-5-35 (0 + 5)	1,39	1,78	0,89	26,7 ±0,2	14,0	–	24,2	27,2
	1,62		1,04		15,0	–	24,6	23,7
HTZ-120 + PLN-2-35+ PLN-4-35 (2 +4)	1,37	2,16	1,07	27,1 ±0,3	15,3	–	23,6	22,1
	1,61		1,25		16,6	–	24,8	19,8

1) – operating speed; 2) – operating width; 3) – productivity per hour of prime time;
4) – plowing depth; 5) – tractor slipping; 6) – plow pulling resistance;
7) – hourly fuel consumption; 8) – fuel consumption per hectare

During field experimental studies, it was found that a plowing machine-tractor unit, operating according to the scheme (2 + 4) has satisfactory trajectory indicators. So, the oscillations of the furrow trajectory, laid by this plowing machine-tractor unit, are rather low-frequency (Fig. 2.28). The main dispersion spectrum, which has a value of 148,44 cm^2, is concentrated in the frequency range of 0 ... 0,7 m^{-1}. When the speed of a plowing machine-tractor unit is 1.61 m·s^{-1} it is 0 ... 1,1 s^{-1} or just 0 ... 0,17 Hz. The length of the correlation relationship of fluctuations in the trajectory of the furrow of the plowing machine-tractor unit in this case is at least 8,5 m.

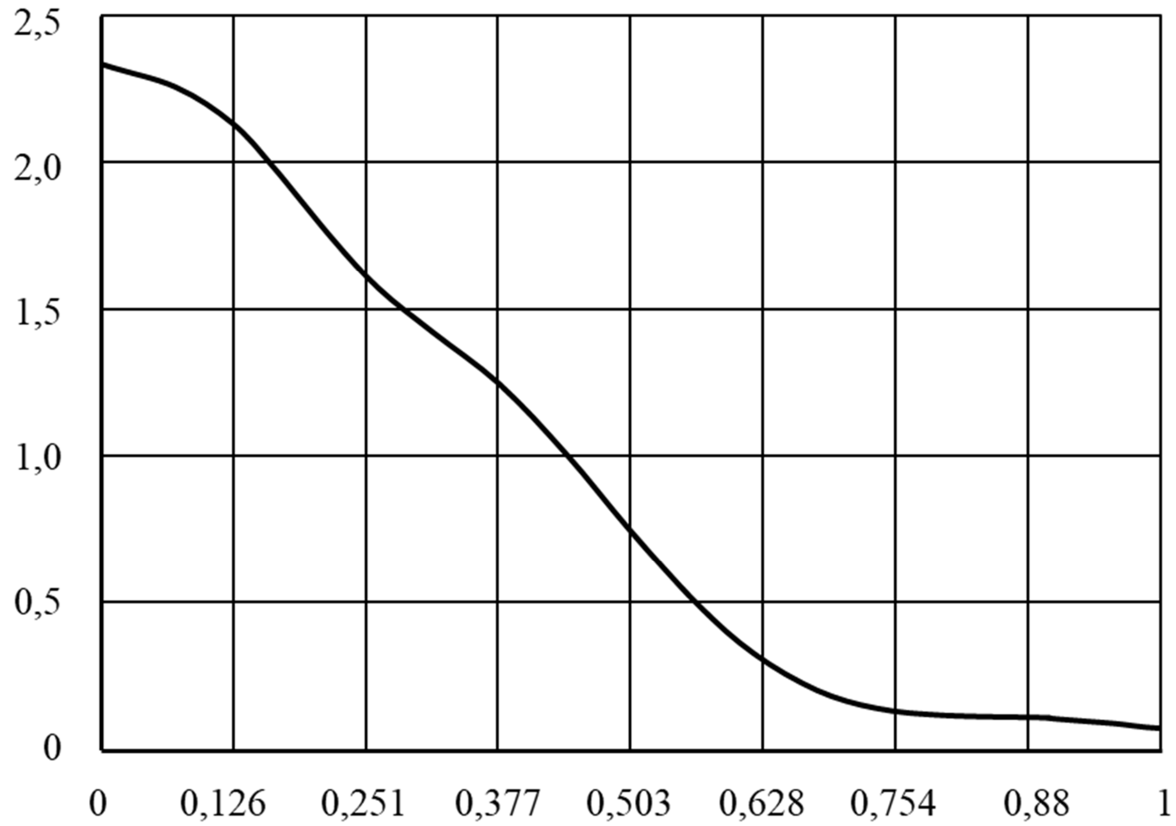


Fig. 2.28 – Normalized spectral density of oscillations of the trajectory of the furrow, laid by the plowing machine operating under the scheme (2 + 4)

2.10. Influence of the technological mode and layout of a plowing unit on the stability of its movement in the longitudinal-vertical plane

It is known that the stability of the plowing machine movement in the longitudinal-vertical plane determines the stability of the plowing depth. As shown above, the vertical movements of the power vehicle mainly depend not on the unevenness of the traction resistance of this vehicle, but on the fluctuations of the longitudinal profile of the cultivated agricultural background.

The influence of the latter on the stability of the front plowing tool can be traced in a certain way by analyzing the fluctuations in the angle of rotation (tilt) of the lower links of the front and/or rear linkage of the tractor. Thus, our experimental studies have shown that the mean square deviation of the rotation angle of the HTZ-120 wheeled aggregate tractor's front linkage lower links did not exceed 1 degree. The variance of the oscillations of this parameter was concentrated in the frequency range of 0 ... 6,0 m^{-1} (Fig. 2.29). With the forward speed of this plowing unit equal to 1,37 – 1,61 $m \cdot s^{-1}$, (see table 2.6 data) this amounted to 8,2 ... 9,7 s^{-1} or 1,3 ... 1,5 Hz.

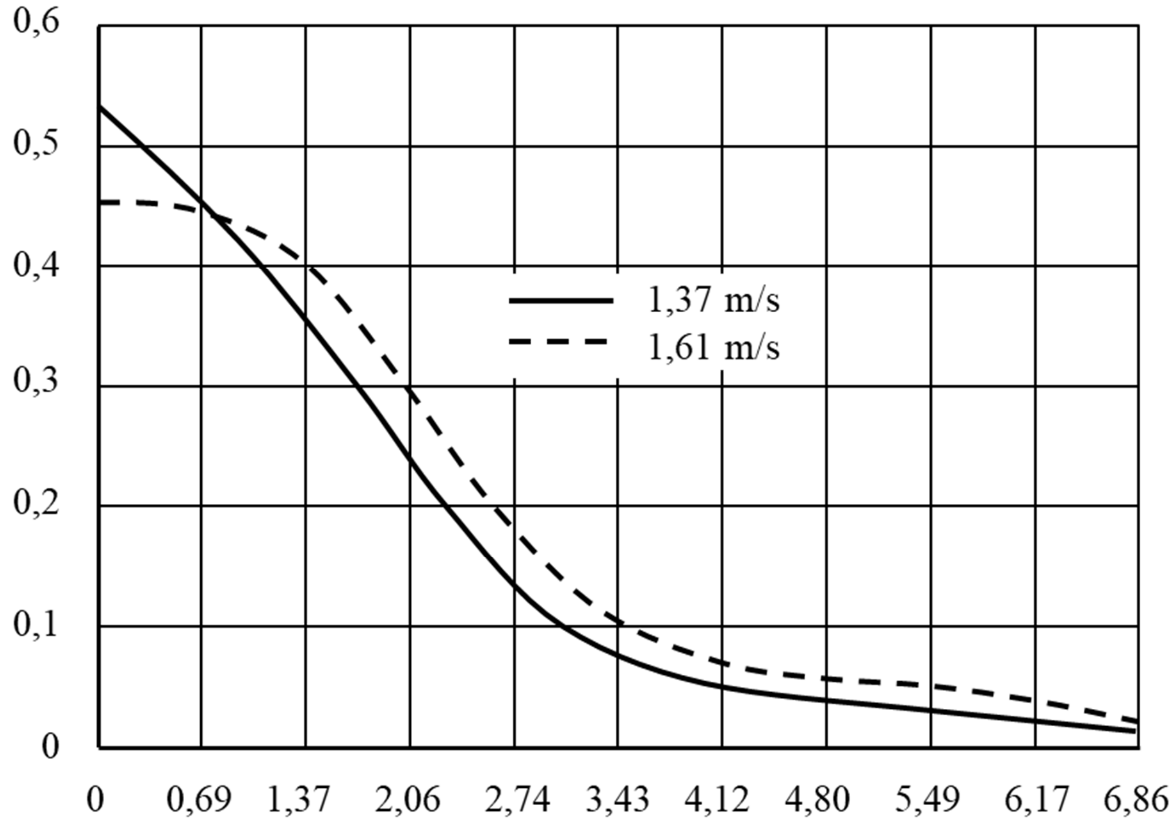


Fig. 2.29 – Normalized spectral densities of oscillations of lower links of front linkage of the inter-aggregate tractor HTZ-120 at different travel speeds of the plowing unit

The nature of the normalized correlation functions of the lower link rotation angle oscillations shows that the correlation length for this process is approximately 4,0 m or 2,5 s at a speed of the plowing machine-tractor unit of 1,61 m·s^{-1} (Fig. 2.30). At a working speed of 1,37 m·s^{-1}, the correlation length of the process in question increases to 4,4 m or 3,2 s.

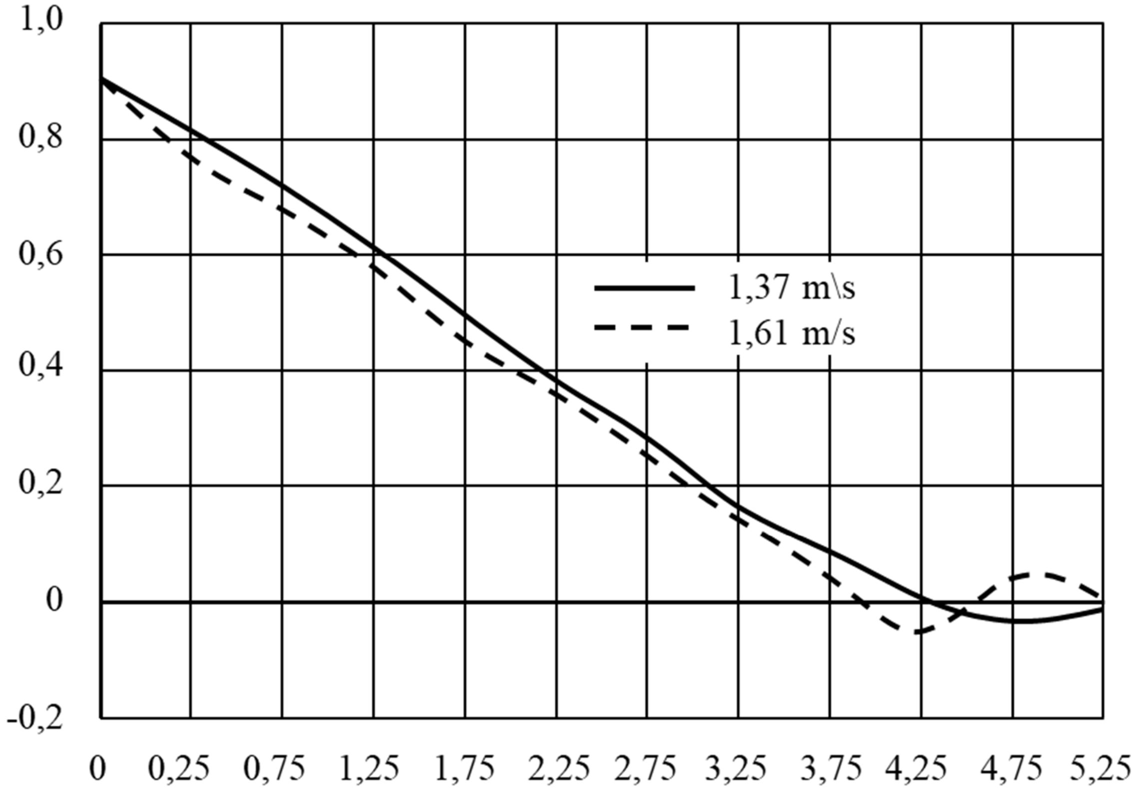


Fig. 2.30 – Normalized correlation functions of oscillations of HTZ-120 tractor's front linkage lower links at different speeds of the plowing unit

As emphasized above, the standard deviation of the rotation angle α_n of the of the HTZ-120 tractor's front linkage lower links did not exceed 0,017 rad. (i.e. actually 1°).

Due to the small angle of rotation of the aggregating tractor's front linkage lower links, the angular oscillation of the front plow was practically absent. It turns out that the front plow during work made practically only vertical movements. In this case, the stability of the front plow's plowing depth can be theoretically estimated by vertical displacements of its "center of resistance" as follows. The distance from the latter to the attachment point of the front linkage lower links to the frame of the tractor HTZ-120 in the experimental vehicle was 1,05 m. The standard of the fluctuation angle α_n was equal to, ±0,017 rad, as mentioned above. It follows that the standard deviation of vertical oscillations of the front plow (adequate to the plowing depth) should be at the level of 1,83 cm. With actual value of this parameter 1,88 cm (which will be shown below) we have quite acceptable coincidence of theoretical and experimental data.

According to experimental data, the mean square deviation of the plowing depth of each comparable unit did not exceed the agricultural requirements (2 cm) and separately amounted to: for a plowing machine-tractor unit operating under the scheme (0 + 4) – 1,74 cm; for a plowing machine-tractor unit operating under the scheme (0 + 5) – 1,52 cm and for a plowing machine-tractor unit operating under the scheme (2 + 4) – 1,88 cm.

From the results of analysis of variance, it follows that at significance levels of 0,05 and even 0,01, the difference between these standards is statistically random, because according to Fisher's F-criterion [30] the zero-hypothesis of equality of the compared variances of plowing depth variations is not rejected.

The actual depth of plowing performed by the compared plowing machine-tractor units can also be considered almost the same (Table 2.6), because the zero-hypothesis of equality of the average values of this parameter at statistical significance levels of 0,05 and 0,01 is not rejected.

Normalized spectral densities (Fig. 2.31) and correlation functions (Fig. 2.32) of plowing depth fluctuations by the comparable plowing machine-tractor aggregates differ little from each other.

The main share of dispersion of the estimated parameter for all schemes of plowing machine-tractor units is concentrated in the frequency range of 0 ... 5,0

m^{-1} (see Fig. 2.31). This is somewhat more narrowed down in comparison with the normalized spectral densities of oscillation angle of the HTZ-120 tractor's front linkage lower links (see Fig. 2.30).

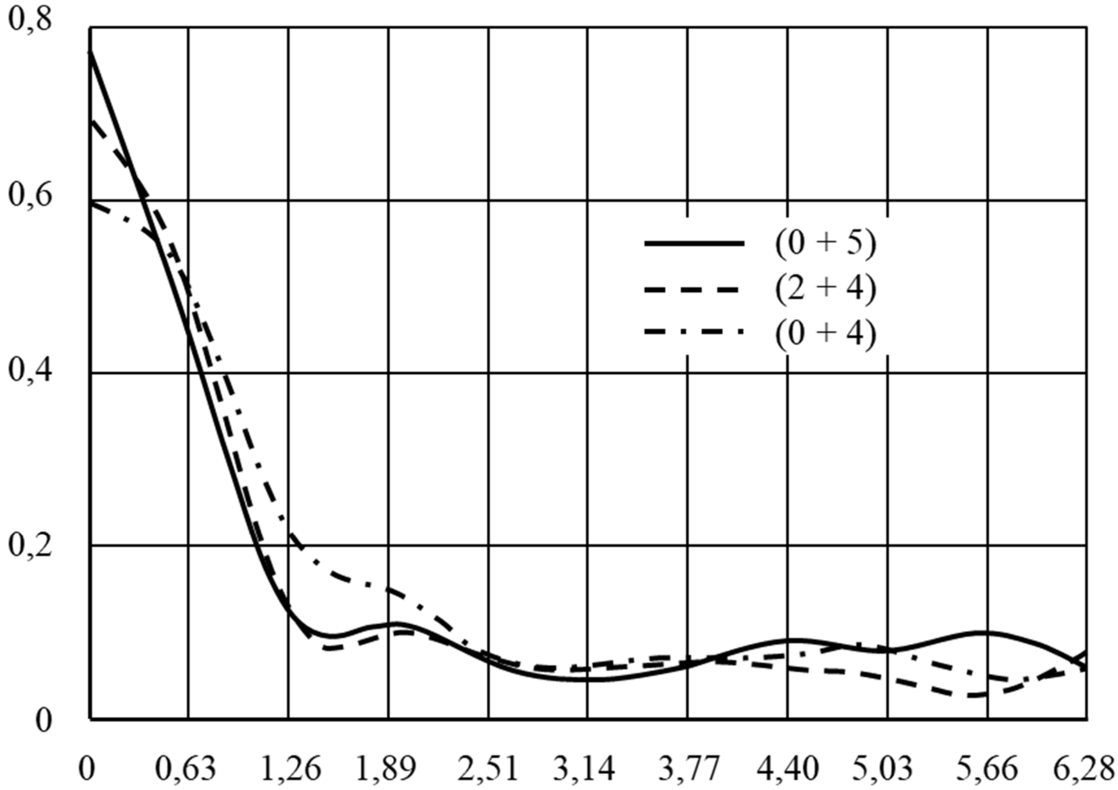


Fig. 2.31 – Normalized spectral densities of fluctuations in the depth of plowing by plowing units of different schemes on the basis of the tractor HTZ-120

Wider frequency range of angle α_n fluctuations can be explained by the fact that its change is due to the simultaneous effect of:

1) vertical displacements of the front axle of the aggregating tractor, to which the front plow is connected;

2) independent vertical oscillations of the front-mounted plowing tool.

In the normalized correlation functions of oscillations of both angle α_n, and the depth of plowing of the compared plowing machine-tractor units, there are no periodic components (see Fig. 2.30 and Fug. 2.32). And this indicates that the vibrations of the main source of vertical vibrations of the machine-tractor unit – the longitudinal profile of the track – do not have such components.

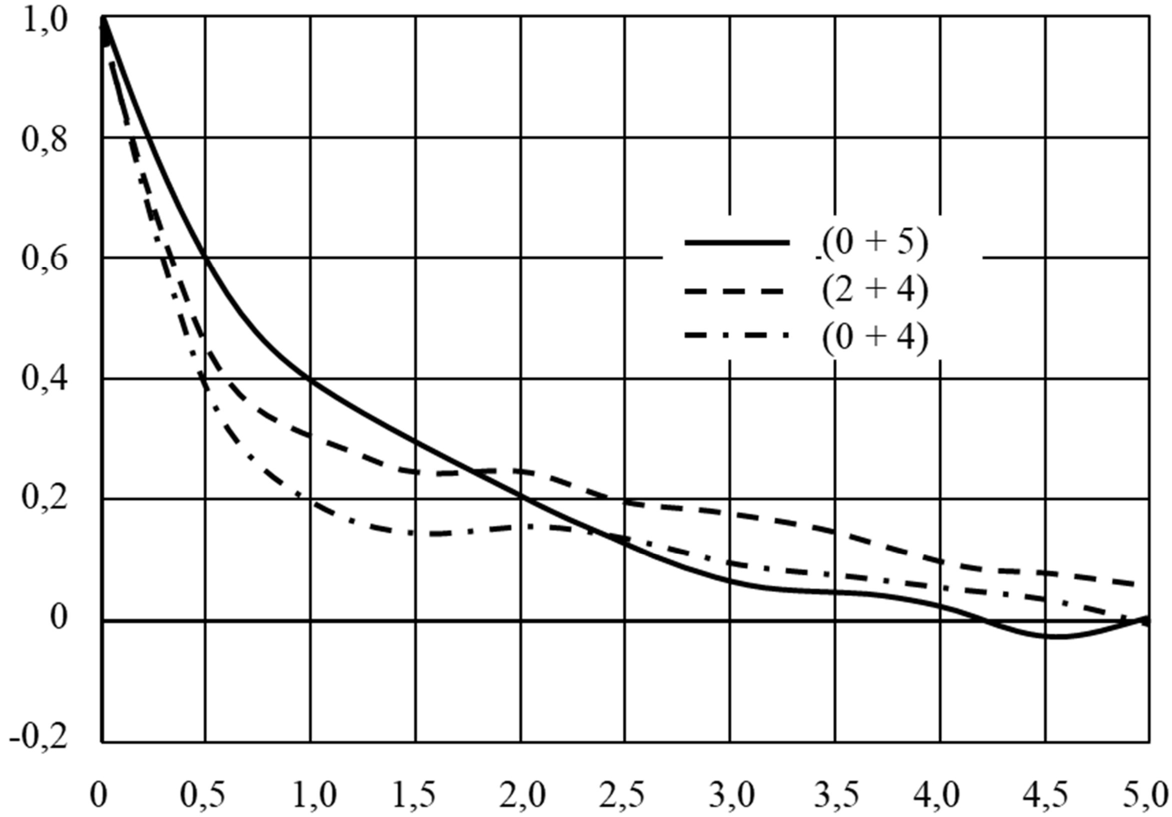


Fig. 2.32 – Normalized correlation functions of fluctuations in the depth of plowing by plowing units of different schemes on the basis of tractor HTZ-120

2.11. Operational and technological assessment of a "push-pull" plowing machine based on an aggregate tractor HTZ-120

One of the most important characteristics of any plowing machine and tractor unit is the complexity of its assembly, which lies in the approach of a power vehicle (aggregating tractor) to the agricultural implement and its subsequent connection.

When aggregating front and rear plows with the tractor HTZ-120, no significant problems were encountered. It was found that the order in which the plows are connected to a given power vehicle can be arbitrary. Analysis of the experimental data showed that two mechanics spent about 7 minutes to connect the front-mounted plow to tractor HTZ-120 (Table 2.7). Disconnecting the mentioned plowing tool took no more than 6 minutes.

Table 2.7 – Labor intensity of front plow aggregation

Name of operation	Labor intensity, men/hour
Aggregating the plow (2 mechanics)	
1. Approach of the tractor to the plow (from a distance of 5 m)	0,0028
2. Connecting the lower links of the tractor's front linkage	0,0444
3. Connecting the center link of the tractor's front linkage	0,0200
4. Setting the plow leg into transport position	0,0010
5. Locking the lower links of the tractor linkage	0,0500
TOTAL:	0,1182
Disconnecting the plow	
1. Unlocking the lower links of the tractor front linkage	0,0450
2. Setting the plow leg into working position	0,0010
3. Disconnecting the tractor center link	0,0200
4. Disconnecting the lower links of the tractor's front linkage	0,0300
TOTAL:	0,0960

It took the two mechanics almost as long to attach and detach the rear-mounted plow to this aggregating wheeled tractor.

In general, it turns out that for aggregating tractor HTZ-120 with front and rear plows two mechanics need not more than 15 minutes. It should be noted that in the presence of fast connecting automatic devices even one mechanic could cope for this amount of time.

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The operating procedure for the new plowing machine during field work was as follows. When, before entering the turn lane, the edge of the field board of the second body of the front plow came out on the reference line, the operator raised it into transport position and continued moving in a straight line until the edge of the field board of the last body of the rear plow came out on the reference line.

Before carrying out the operational and technological assessment of arable machine-tractor units, the field was divided into paddocks of a certain width. The optimal value of this parameter ($C_{opt.}$) was found according to the expression [31]:

$$C_{\text{опт.}} = \left(16 \cdot R_{\text{min.}}^2 + 2 \cdot B_p \cdot L_p\right)^{\frac{1}{2}},$$

where B_p – working width of a plowing machine;

L_p – field length;

$R_{\min}$ – permissible turning radius of a plowing machine.

For a plowing machine-tractor unit, working according to the scheme (2 + 4), we have B_p = 2,1 m. At L_p = 800 m we obtain: C_{opt} = 64 m.

At the same time, for a plowing machine-tractor unit working according to the scheme (0 + 5) – B_p = 1,75 m, and C_{opt} = 59 m.

Actual value of the pen width (C_∂) was chosen so that it would be at least $C_{opt.}$ and a multiple of the machine's double working width:

$$C_{ОПТ} \leq C_\partial = K.2.B_p.$$

where K – multiplicity coefficient.

For a plowing machine – tractor unit working according to the scheme (2 + 4), the coefficient $K = 16$, and for a unit operating according to the scheme (0 + 5), the value of this coefficient is $K = 17$. It follows that for the first machine-tractor unit $C_\partial = 67{,}2$ m, and for the second plowing machine-tractor unit – $C_\partial = 59{,}5$ m.

After arranging the working zones, temporary observations were made of the operation of the compared plowing machine-tractor units. Both of them worked in the background mode with the characteristics shown in Table 2.8.

Plow PLN-5-35 with the second remote body was used as a rear-mounted plowing tool of the new plowing machine-tractor unit. After installing the latter in its place and disconnecting the front plow from the HTZ-120 tractor, we got a basic plowing machine-tractor unit. The plows of the new and the basic plowing units were adjusted to a plowing depth of 27 cm. To exclude the influence of the subjective factor, the same machine operator worked on both plowing machines alternately.

Analysis of the data obtained (Table 2.9) showed that the use of a plowing unit, working on the scheme (2 + 4), compared with a plowing unit working on the scheme (0 + 5), increased the working width of the machine by 20,7 %. The working speed of the new plowing machine - tractor unit was only 3 % lower. In the end, the basic (net) productivity of the new plowing machine was 17 % higher than that of the basic one.

The coefficient of utilization of shift time of the plowing machine-tractor unit consisting of an aggregate tractor HTZ-120 and plow PLN-5-35 was 2,4 % higher. This is reflected in the fact that the shift output advantage of the new unit has decreased to 14,4 %.

Table 2.8 – Conditions for operational and technological evaluation of the compared plowing machine-tractor units

Indicator	Value
Soil type	Dark brown,
	Residual saline
Terrain Microterrain	flat levelled
Agricultural background	Field after wheat harvesting
Soil humidity (%) in the layerв слое:	
0 … 5 cm	8,3 ... 8,8
5 … 10 cm	8,8 ... 11,0
10 … 15 cm	9,7 ... 14,5
15 … 20 cm	13,5 ... 15,2
20 … 25 cm	11,5 ... 13,8
Soil hardness (MPa) in the layer 0 … 25 cm	1,24 ... 1,28
Amount of weeds, no.$\cdot m^{-2}$	50 ... 60
$g \cdot m^{-2}$	150 ... 160
Presence of wheat stubble residues, $g \cdot m^{-2}$	60 ... 80

Time expenditures for elimination of technical failures in the two plowing machines compared are almost the same, as evidenced by the values of operating time utilization coefficients (see Table 2.9).

The average duration of one turn of the new plowing machine was only 4 % longer. The obtained practical result is close to the above calculations and equals 3 %.

Table 2.9 – Operational and technological performance of the compared plowing machine-tractor units

Indicator	Schematic diagram of a plowing machine-tractor unit	
	(2 + 4)	(0 + 5)
Composition of the plowing machine-tractor unit: tractor	HTZ-120	
Plowing machine	PLN-2-35+ PLN-4-35	PLN 5-35
Mode of work:		
– working width, m	2,5	1,78
– velocity of working movement, $m \cdot s^{-1}$	1,60	1,65
– set plowing depth, cm	27	27
– rut length, m	800	
Performance, $ha \cdot h^{-1}$:		
– main time	1,24	1,06
– shift time	0,90	1,03
– operating time	0,99	0,86
Specific fuel consumption, $kg \cdot ha^{-1}$	24,3	27,2
Operational and technological coefficients:		
– use of shift time	0,83	0,85
– use of operating time	0,80	0,81
– technological process reliability	0,98	0,97
– working strokes	0,88	0,90
The average duration of one		
“pear-shaped” turn, s	52	50
Width of the turning lane, m	32,4	30,3
Agricultural characteristics:		
– average plowing depth, cm	26,8	27,1
– uniformity of plowing depth, ±cm	1,90	2,00
– Uniformity of working width, cm	5,8	6,1
– flaws	Not present	

In addition to an increase in labor productivity, the use of the new plowing unit scheme reduces specific fuel consumption by an average of 10.7%.

As for the quality of work, it is almost the same in both compared plowing machine-tractor units.

CHAPTER 3
PLOW OPERATION
WITH MODULAR POWER VEHICLES

3.1. Prospects and efficiency of modular power vehicles

In their time, experts in the field of operation of machine-tractor units, found that for the successful agricultural activities in the most typical farms – collective, large farms and others, mobile power vehicles of several classes are needed. Manufacturers of mobile power vehicles (tractors) for agriculture offer a wide range of different classes, namely: class 0,2 – power tillers and mini-tractors; class 0,6 – self-propelled chassis tractors; class 1,4 – multipurpose row crop power vehicle; Class 2 – heavy-duty tractors for general purpose; Class 3 – general-purpose tractors; Class 5 – general-purpose power tractors such as K-700, K-701; class 6 – special purpose tractors [32]. At the same time for Ukraine, the greatest number of tractors required are from class 1,4 (type UMZ-6 and "Belarus"), class 3 – (type T-150K and T-150 – Kharkov tractor plant) and class 5 - tractors (type K-700, K-701).

Many years of practice show that the greatest volume of field work was and is performed by tractors traction class 1,4; 2; 3 and 5. Of these, Ukraine currently produces only models of class 1,4 (YMZ-80) and class 3 (tractors family HTZ).

Without powerful tractors of traction class 2, agricultural producers can somehow manage by, for example, implementing 6- and 8-row systems of cultivating row crops with inter-row of 70 cm on the basis of units consisting of domestic aggregate tractors YMZ-6 (YMZ-80) or the Belarusian tractors like MTZ-80.

At the same time, in the southern regions of the country 12- and even 18-row systems are more effective, the implementation of which requires tractors of traction class 2. This is facilitated by the fact that at present, domestic power vehicles of traction class 3 – tractors of the family HTZ-121/160 are quite suitable for this purpose [28].

As for tractors of traction class 5, their absence has more serious consequences. The fact is that in many farms a significant amount of arable land remains uncultivated in autumn. Spring primary tillage in most of Ukraine is quite problematic. In the southern regions it is generally unacceptable.

While we used to attribute these problems to fuel shortages or something else, recently most experts see the main reason as the lack of such highly productive tillage units, which in the past were designed on the basis of such tractors as K-700 and K-701. Despite the fact that the annual load of these vehicles did not always exceed 50 %, their widespread use only in the main tillage was justified.

Recently, the lack of such tractors in Ukraine significantly hinders the introduction of promising “No-till” technology [33].

To solve the problems associated with the lack of tractors of the necessary traction classes in Ukraine, we must either produce them ourselves or buy them abroad.

Practical implementation of the first option at domestic tractor plants without their significant capital re-equipment associated with significant financial costs is practically impossible. So, HTZ (not to mention Dnepropetrovsk YUMZ) cannot produce tractors of traction classes 2 or 5 in their own facilities.

As for the purchase of power vehicles (tractors) abroad, there are a number of problems. First, foreign tractors must be easily adaptable to domestic agricultural tools and machinery. Secondly, it is necessary to pay attention to the ratio of the power vehicle (tractor) weight to the power of its engine. If the power capacity of the tractor exceeds 20 $kW \cdot t^{-1}$, then there is the problem of its effective loading in a purely traction version.

Third, in addition to the allowable energy saturation, universal row crop tractors must have a wheel track and tires that allow them to fit into the accepted row spacing of cultivated crops, furrow widths, etc.

Fourth, it is the cost of oils and lubricants, some peculiarities of maintenance of foreign power vehicles and so on.

However, there is a way out of this difficult situation. It lies in the development of fundamentally new modular power vehicles in our country on the basis of those tractors that are produced. First of all, it concerns a general-purpose modular powervehicle of variable traction class 3-5.

The modular power vehicle consists of two modules: power and technological one. The power module is an energy-intensive tractor (preferably with two levels of its engine power), and the technological is an attachable (if necessary) additional axle with an active wheel drive, trailed mounting system and appropriate technological equipment [2].

For many years we have been testing a general-purpose modular power vehicle under the conventional brand MES-300 (Fig. 3.1). Its power module differed from the serial wheeled tractor T-150K by a synchronous PTO and a different engine (SMD-601), whose power was set at two levels - 125 and 162 kW.

The MES-300 technological module was mounted on the rear axle of the T-150K tractor. The active drive of its wheels was carried out from the shank of the synchronous PTO of the power module through the cardan shaft via the appropriate reducer. The vertical and horizontal articulated joints of the frame of the technological module allowed it to rotate relative to the energy module (i.e. tractor) by 30° in the horizontal and by ±15° – in the transverse-vertical plane, as well as to copy the track profile irregularities by its wheels. For aggregation with agricultural machines and tools, the technological module was equipped with a rear hydraulic trailing system.

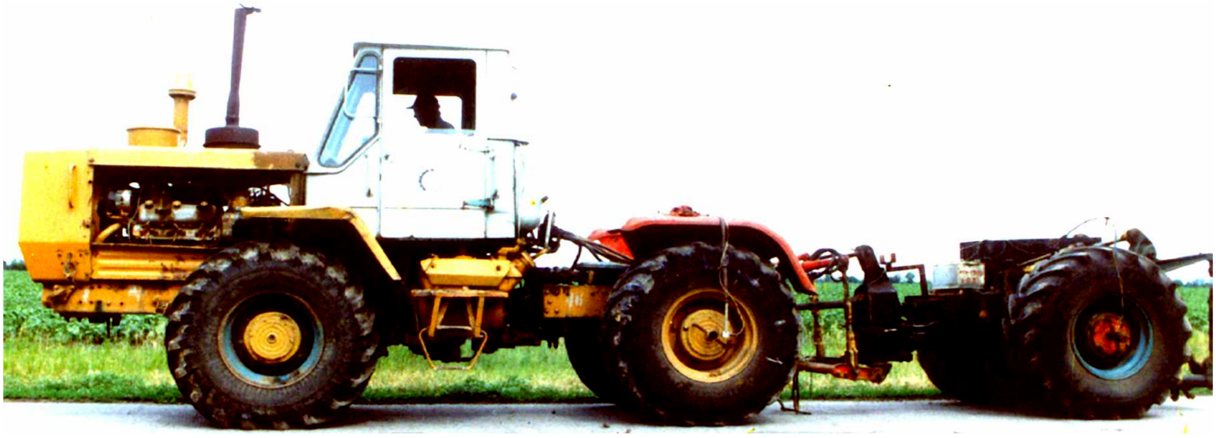

Fig. 3.1 – Modular power vehicle MES-300

Brief technical characteristics of MES-300:

Operating weight, kg	11900
incl. the power vehicle, kg	8200
the technological module, kg	3700
Engine power, kW	125/162
Energy capacity, kW·t^{-1}	19.8
Base, mm	5500
Track, mm	1680
Tire size:	21,3R24

The power module of the MES-300 power vehicle was used independently as a tractor of traction class 3 with a set of agricultural equipment and tools, designed mainly for aggregation with the T-150K wheeled tractor. In this case, the technological module was disconnected and the motor power switch was moved to a lower value (125 kW).

When connecting the technological module, the upper power limit of the engine installed on the MES-300 (162 kW) was set. The weight of the entire power vehicle was increased both by attaching the technological module and by the possibility of ballasting the latter with a weight of up to 1,5 tons.

Analysis of the experimental studies showed that two mechanics of average qualification connected the modules of the modular power vehicle in about 8 – 10 minutes [2].

Disconnecting the technological module from the energy module was carried out by one mechanic in 4 – 5 minutes. If there is a non-mechanical (electric, hydraulic or other flexible) wheel drive of the technological module, even less time will be required for its aggregation with the power module.

As the tests showed, the traction and power characteristics of the MES-300 modular power vehicle are about the same as those of the K-700 tractor. This means that the new power vehicle can be classified in at least two traction classes, i.e., 3 and 5. As a result, in practice, it can be used both in a loop with agricultural machinery, which are designed for aggregation with tractor type T-150K, and with power vehicles such as K-700 and K-701. For example, the operational and technical evaluation of units based on MES-300 showed that their main indicators correspond to the normative indicators of similar units based on the tractor K-700 (see Table 3.1).

To implement a reasonable choice of scheme and design parameters of a modular power vehicle of variable traction class 3 – 5, we have developed an appropriate scientific rationale, including the draft baseline requirements.

According to the results of theoretical and experimental studies, the specified modular power vehicle should have the following design parameters [2]:

Operating weight, kg	13500 … 14400
incl. power module, kg	8100 … 8650
Technological module, kg	5400 … 5750
Engine power, kW	200 … 250
Energy capacity of the energy module, $kW.t^{-1}$	23,1 … 30,9

Today, in Ukraine, new tractors of HTZ-170 (picture 3.2) and/or HTZ-160 (picture 3.3) can potentially be used as a power module of modular power vehicles of variable traction class 3 – 5. The rear axles of these power tools are the elemental base for the creation of appropriate technological modules on their basis.

Table 3.1 – Performance of plowing machine-tractor units based on the MES-300

Indicators	**Value of indicators**				
Tool brand:	KPSH-9	BDT-10	PCH-4,5	PNL-8-40	4KPS-4
Operating conditions:					
– working width, m	7,68	10,0	4,22	2,80	15,79
– tillage depth, cm	16,0	11,8	25,0	27,0	10,8
– speed, $km \cdot h^{-1}$	7,39	8,0	7,17	7,15	6,98
Productivity, $ha \cdot h^{-1}$					
– main	5,67	8,0	3,02	2,00	11,2
– operational	4,67	6,5	2,68	1,58	9,50
Fuel consumption, $kg \cdot ha^{-1}$	7,37	4,15	14,8	16,3	3,46
Utilization rates:					
– operational time	0,82	0,81	0,88	0,79	0,86
– working strokes	0,87	0,83	0,94	0,88	0,92

From the point of view of short-term forecasting of development and application of modular power vehicles on the basis of HTZ-160 and HTZ-170 tractors, they are not inferior to such foreign tractors of traction class 5 as K-701 and Case 4690 according to the technical level and competitiveness.

It is established that the class 3 – 5 modular power vehicle can be widely used in summer-autumn operations of basic tillage (plowing, flat-cutting cultivation, removing stubble of crops, etc.). In addition, this power vehicle can be used for soil preparation for sowing of late crops and fallow tillage. The duration of loading of such a power vehicle for each of these operations will be determined by the structure of areas of cultivated crops.

As calculations show, during the spring-summer period, a class 3 – 5 modular power vehicle can be busy with pre-sowing cultivation and fallow tillage for at least 300 hours. The loading of the new power vehicle in the summer-autumn period can be at least 670 hours. Moreover, its loading in the second half of the year is more even than that in the first half of the year.

Fig. 3.2 – Modular power vehicle of traction class 3 – 5
on the basis of HTZ-170 tractors

Fig. 3.3 – Modular power vehicle of traction class 3 – 5 on the basis of
HTZ-160 tractors

The total utilization of the modular power tool reaches 976 hours. The rest of the year the technological module cannot be used. But it is obvious that the loss from its downtime is much (5 - 7 times) less than the loss that can be incurred from the downtime of tractors of traction class 5.

Calculations show that to perform almost the entire range of agricultural work in the farm with an arable area of about 4500 hectares requires three power and two technological modules (i.e. two modular power tools of traction class 3 – 5 and one power module - tractor type HTZ-170 and/or HTZ-160).

In addition to saving labor resources by 25 %, the use of modular power vehicles allows you to reduce both fuel consumption by 23,5 %, and the duration of technological operations with all the positive consequences that follow from this.

As we can see, our country has all the facilities to create new modular power vehicles of variable traction class 3 – 5, the implementation of which provides a number of advantages.

Firstly, with tractors and technology modules at your disposal, you can completely do without the much-needed tractors of traction class 5.

Secondly, the introduction of modular power vehicles can be carried out using their own facilities and does not require, which is very important, the creation of fundamentally new designs of agricultural machinery and tools.

Thirdly, the annual workload of tractors of traction class 3 increases significantly due to their use with and without a traction module. For some time during the year technological modules may not be used, but the losses from their downtime are many times less than from the downtime of tractors of class 5.

Fourthly, the probability of quality mastering the tractor design of two brands (HTZ-160 and HTZ-170) instead of three or more - is much higher, which in itself will have a positive impact on the effectiveness of their maintenance and practical operation.

3.2. Specific features of combining the modular power vehicle with a plow

Depending on the ratio of the actual B_d and the desired B_t values of the power tool gauge, the plow can be connected symmetrically to the coupling tractor as well as with a right- or left-hand cross-shift $e_п$ [see expression (1.12)].

In the right-hand transverse displacement that we already discussed in the first chapter, the "center of resistance" of the plow (point *D'*, Fig. 3.4) is located to the right of the longitudinal axis of symmetry of the power vehicle.

Between this line and the line of traction there is an angle γ, which leads to the appearance of the lateral component T_y' of force T'. During plowing operation, the force T_y' is an additional load on the field boards, which leads to an increase in their friction force against the furrow wall and, ultimately, to an increase in the traction resistance of the plowing tool.

If the plow is fixed symmetrically, its "center of resistance" (point D_o, Fig. 3.4) is in the symmetry plane of the power vehicle. In this case, although the force T_y' does not disappear, its impact is minimized. The pressure on the field boards of the plow is mainly by the forces that act on the plowing tool (plow) from the soil moved to the side ($\sum P_y$, Fig. 3.4).

High traction characteristics and relatively narrow track of the modular power vehicle make it possible to attach a plow not only symmetrically, but also with a left-hand cross-shift. In the case of the location of the "center of resistance" of the plowing tool to the left of the longitudinal axis of symmetry of the power vehicle (point *D"*, Fig. 3.4) the lateral component T_y'' of traction force T'' contributes to reducing the load on the field boards.

The condition of plow equilibrium in this case is as follows:

$$T_x''.e_n - T_y''.d = 0, \tag{3.1}$$

where d, e_{n} – longitudinal coordinate and magnitude of the left lateral displacement of the "center of resistance" of the plow, respectively.

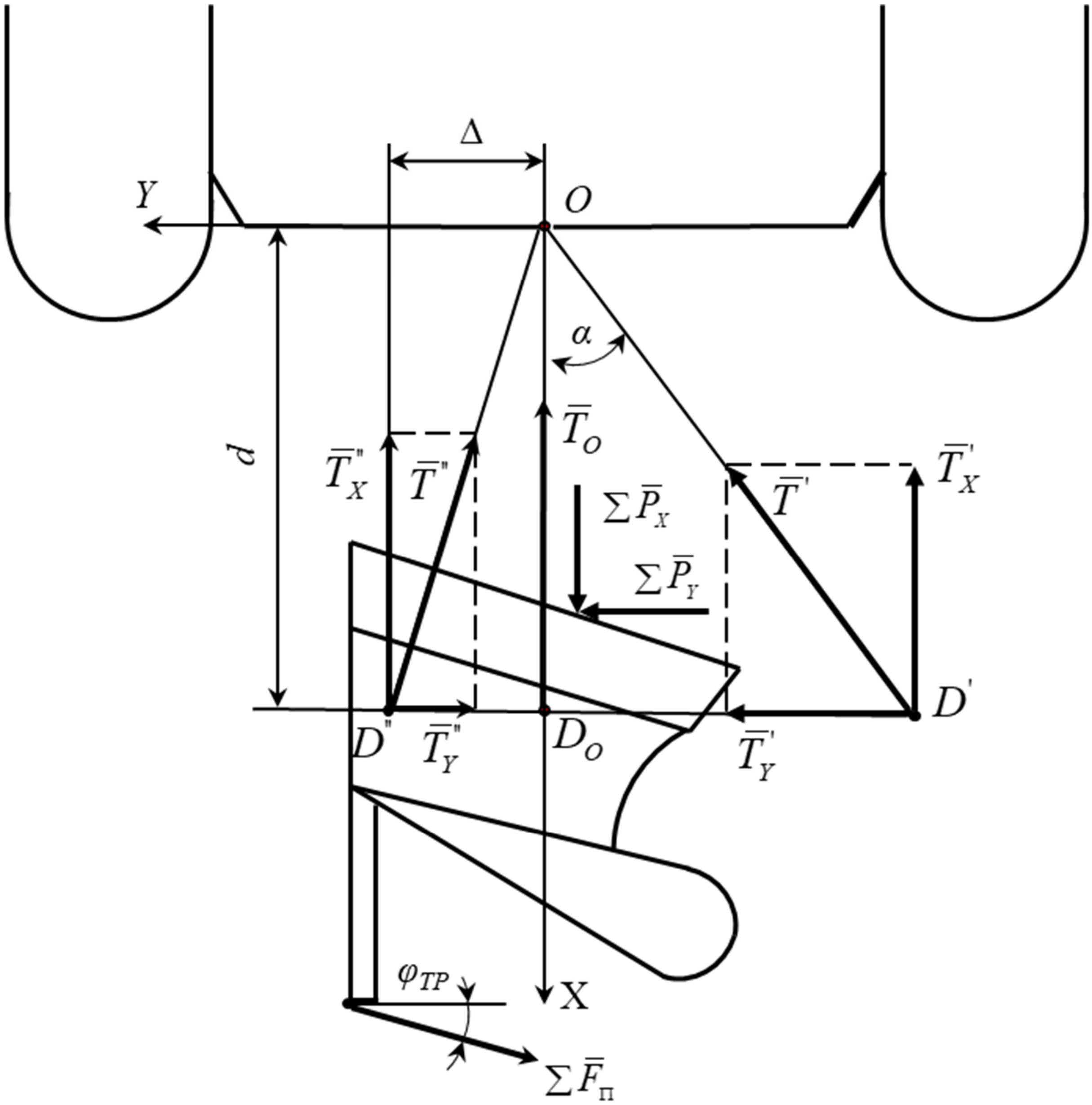

Fig. 3.4 – Diagram of forces acting on the plow at its various transverse displacements

In this case, as can be seen from the scheme shown in Fig. 3.4, it follows that:

$$T''_x = \sum P_x + \sum F_n . \sin\varphi_{tp} , \tag{3.2}$$

$$T_y'' = \sum P_y - \sum F_n . \sin \varphi_{tr} \qquad (3.3)$$

where $\sum P_x$, $\sum P_y$ – total forces of longitudinal and transverse components of soil resistance to the plow bodies, respectively;

$\sum F_П$ – total reaction of the furrow wall, which is deflected from the normal by the friction angle $\varphi_{тр}$ of soils on steel.

In its turn:

$$\sum F_П . \sin \varphi_{TP} = \sum T_{TP}, \qquad (3.4)$$

a

$$\sum F_П . \cos \varphi_{TP} = \sum T_{TP} . f^{-1}, \qquad (3.5)$$

where $\sum F_{тр}$ – total friction force of the field boards against the furrow wall;

f – coefficient of friction of steel on soil.

Taking into account (3.2), (3.3), (3.4) and (3.5), the initial equation (3.1) after transformations will take the following form:

$$\sum F_{TP} = \left(d . \sum P_y - e_П . \sum P_x . \right) \left(e_П + d . f^{-1} \right)^{-1}. \qquad (3.6)$$

As can be seen from expression (3.6), with increasing magnitude of the left lateral displacement of the plow $e_п$ the total friction force of the field boards against the furrow wall decreases according to the hyperbolic law. And this unambiguously leads to a corresponding decrease in traction resistance of the plowing tool.

To ensure the plow is stable in the horizontal plane, the maximum value of $e_п$ should be such that the minimum value of the total force $\sum F_{тр}$ remains more than zero. Otherwise, a plowing tool (plow) will deviate from the furrow wall, worsening or even making it impossible to plow. As it follows from expression (3.6), it is possible if:

$$d.\sum P_y - e_{\Pi}.\sum P_x > 0\,. \tag{3.7}$$

Whence:

$$e_{\Pi} < d.\left[\sum P_y.\left(\sum P_x\right)^{-1}\right]. \tag{3.8}$$

Taking into account that $\sum P_y = \sum P_x \cdot \mathrm{ctan}(\varphi_{\text{тр}} + \gamma_{\text{o}})$, we finally get:

$$e_{\Pi} < d.c\tan\left(\varphi_{TP} + \gamma_o\right), \tag{3.9}$$

where γ_o – angle formed by the blade with the furrow wall.

When determining the value of the left-hand lateral displacement of the plow for each particular plowing machine, the values obtained from expressions (1.12) and (3.9) are compared and for practical use, we take the smaller one. In this case, both the ratio between the track of the modular power tool and the working width of the plow, and the condition for stable operation of the latter are taken into account.

3.3. Influence of the technological mode and setting of the plowing unit on the performance of its trajectories

Quite often, the statistical characteristics of the adopted evaluation parameters are used to analyze the trajectory indicators of agricultural aggregates under practical operating conditions. At the same time, researchers sometimes limit themselves to considering their average values, standards, and coefficients of variation. However, such an assessment can lead to contradictory conclusions, since these characteristics do not carry any information about the internal structure of the process under study.

For a large class of stationary (and those that are reduced to stationary) processes, their internal properties and structure are determined by the correlation function - in the time domain, and the spectral density – in the frequency domain. Spectral density, as we know, gives no additional information about the random process compared to the correlation function. At the same time, for real random processes occurring during the operation of agricultural plowing machine-tractor units, it is this characteristic, which determines the dispersion spectrum of the process and its frequency composition, which is more physically perceptible and, therefore, more informative.

As a result of statistical processing of experimental data, we did not find that with a change in forward speed of the studied machine-tractor unit from 5,1 to 9,6 km.h^{-1}, i.e. at speeds of the plow, aggregated by MES-200 when working with five-bodied (PLN-5-35) and six-bodied (PLN-6-35) plows, average values and standard deviation of ordinates of angle of rotation α of steering wheels, rotation angle β of the plow, course angle φ and rotation angle ψ of the technological module технологического модуля are subject to some kind of law.

There is a certain regularity in the change in the normalized spectral density of these parameters. Thus, with an increase in the translational velocity, the spectrums of their dispersions are somewhat stretched, although they remain mainly low-frequency (the cut-off frequency of the spectrum ω are 1,5 ... 2,0 s^{-1}). The spectral density maximums are shifted towards higher frequencies with simultaneous decrease of their level (Fig. 3.5 – Fig. 3.10).

When analyzing the dependence of the variance spectrum of the initial values (angles φ and ψ) from the place where the hook load is applied, the following pattern is observed. The asymmetrical connection of the plow results in a constant torque. In this case, if there is angular mobility of the technological module relative to the power module, and an angle is set between their longitudinal axes, the value of which is greater the greater the torque. The latter is the product of the traction resistance of the plow by the value of the transverse displacement of its point of application.

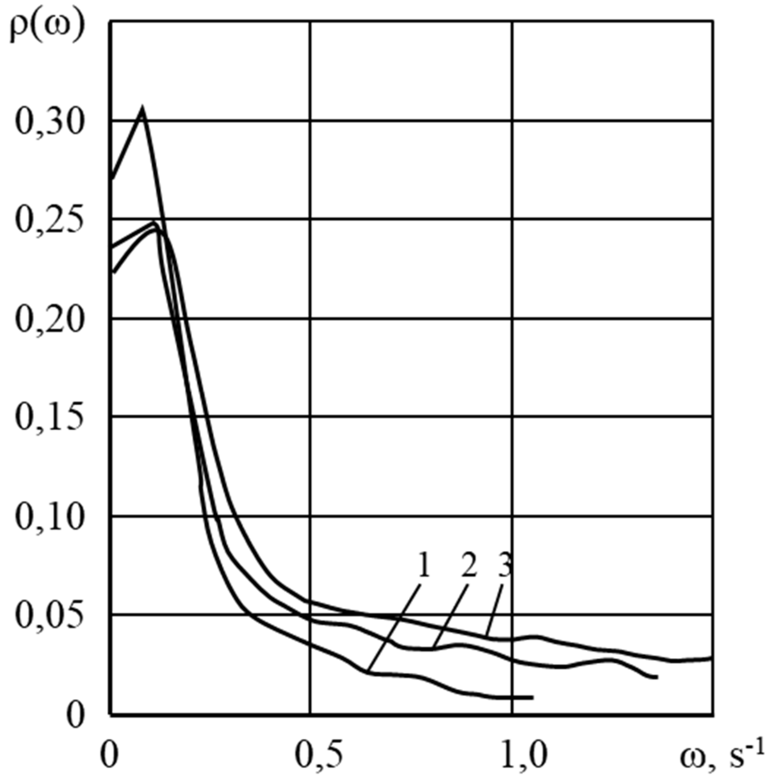

Fig. 3.5 – Normalized spectral density of course angle φ realizations of the power module at a plow cross displacement of 100 mm and travel speeds of a plowing machine 1 – 6,2 km.h^{-1}; 2 – 6,5 km.h^{-1}; 3 – 8,0 km.h^{-1}

It should be noted that the power and technological modules respond differently to changes in torque. Thus, the dispersion spectrum of the course angle φ with increase of transversal rightward displacement of plow from 100 to 300 mm slightly widens (Fig. 3.5, Fig. 3.6).

The variance spectrum of the technological module rotation angleψ , meanwhile, on the contrary, on the contrary, is somewhat narrowed (Fig. 3.7, Fig. 3.8), because an increase in the angle between the axes of the power and technological modules limits the angular mobility of the latter.

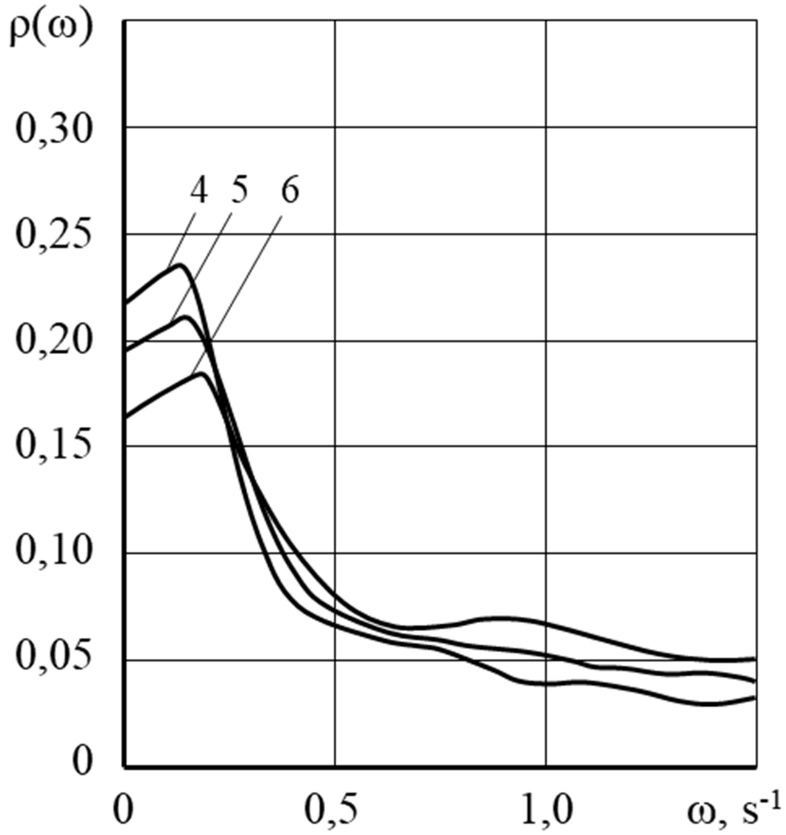

Fig. 3.6 – Normalized spectral density of course angle φ realizations of the power module at a plow cross displacement of 300 mm and travel speeds of a plowing machine 4 – 7,7 km.h^{-1}; 2 – 9,1 km.h^{-1}; 3 – 9,4 km.h^{-1}

Spectral analysis also shows that, compared to the controlled (rotation angle α of the steering wheels, Fig. 3.9) and perturbing (rotation angle β of the plow, Fig. 3.10), the output value dispersion spectra are more affected by (course angle φ, Fig. 3.5 and Fig. 3.6 and rotation angle ψ of the technological module, Fig. 3.7 and Fig. 3.8), i.e., the fluctuations φ and ψ are higher frequency. At the same time, it is known [4], that when a tracking dynamic system is working out the control action, it is desirable that the dispersion spectra of input and output quantities in the range of working (essential) frequencies should be approximately the same. Ideally, they would coincide.

When working out the disturbance (i.e. obstacle), it is desirable to have a spectrum of dispersion of the output quantity in the working frequency range as narrow as possible, compared with the same spectrum for the input signal.

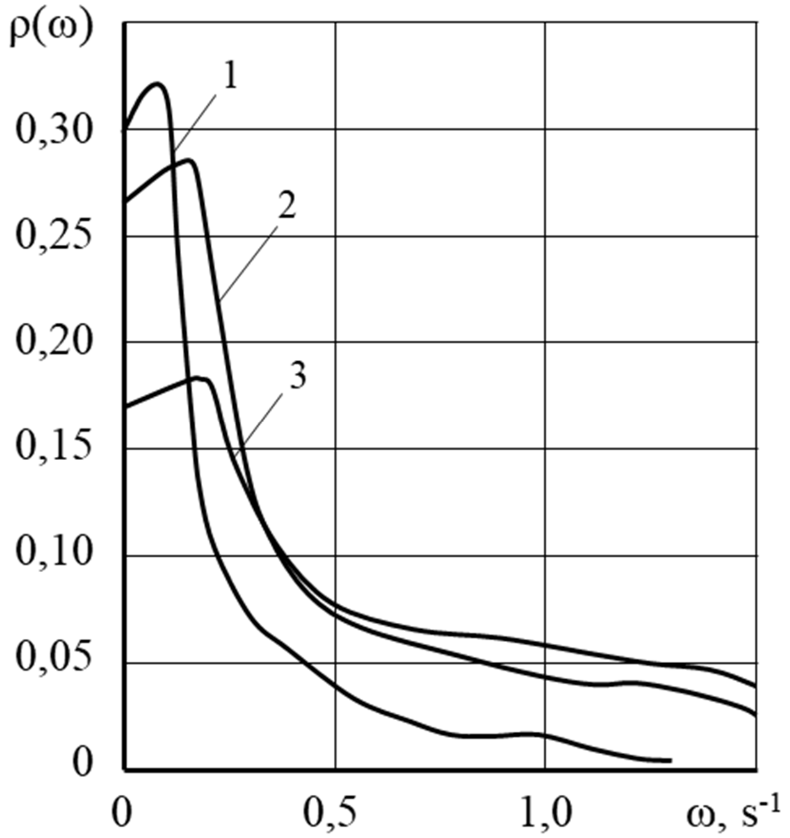

Fig. 3.7 – Normalized spectral densities of realizations of the technological module rotation angle ψ when the plow is shifted laterally by 100 mm and the speeds of the plowing machine and tractor unit are: 1 – 7,9 km.h^{-1}; 2 – 9,0 km.h^{-1}; 3 – 9,4 km.h^{-1}

It was found experimentally that when aggregating a modular power tool with a plow with an unlocked vertical joint of the technological module, the specified desirable conditions are not met. Due to differences in the internal structures of input and output processes, the amplitude-frequency characteristics of plowing machine-tractor units do not match the optimum by controlling and disturbing influences.

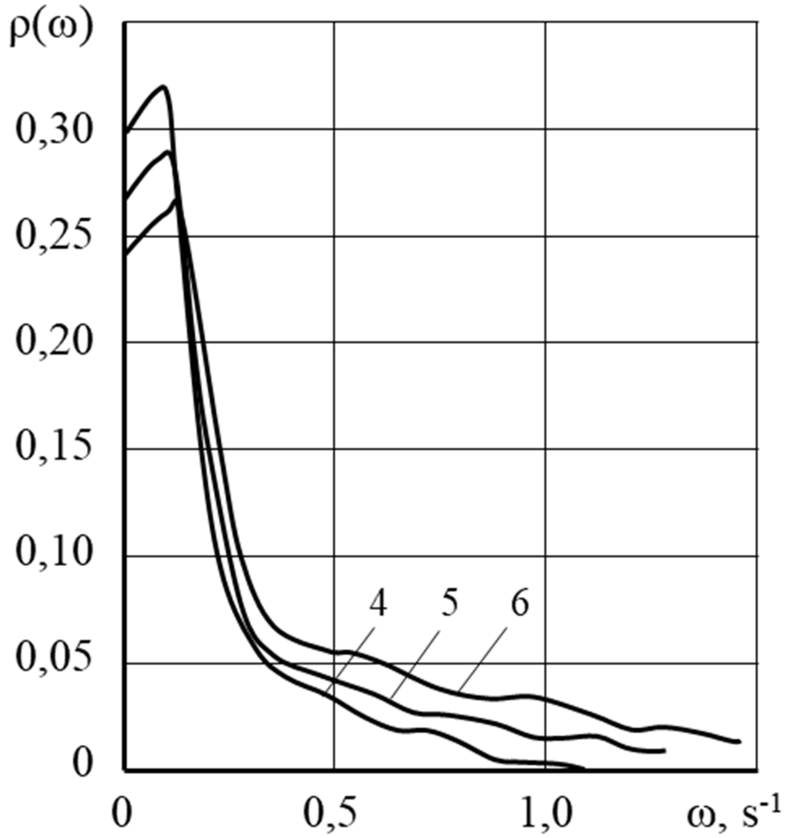

Fig. 3.8 – Normalized spectral densities of realizations of the technological module rotation angle ψ when the plow is shifted laterally by 300 mm and the speeds of the plowing machine and tractor unit are: 1 – 6.0 km·h^{-1}; 2 – 6.6 km·h^{-1}; 3 – 9.0 km·h^{-1}

Only when the specified vertical hinge of the technological module is locked, the range of dispersion of the course angle φ, for example, narrows (Fig. 3.10, Fig. 3.12), and its characteristics take on values approximately equal to those for the control action variance spectrum (Fig. 3.9):

– spectral density maximums occur at frequencies equal to 0,08 ... 0,12 s^{-1};

– the main part of the dispersion spectrums is concentrated in such an interval of 0 ... 0,8 s^{-1};

– the spectrum cut-off frequency is 0,6 ... 0,8 s^{-1}.

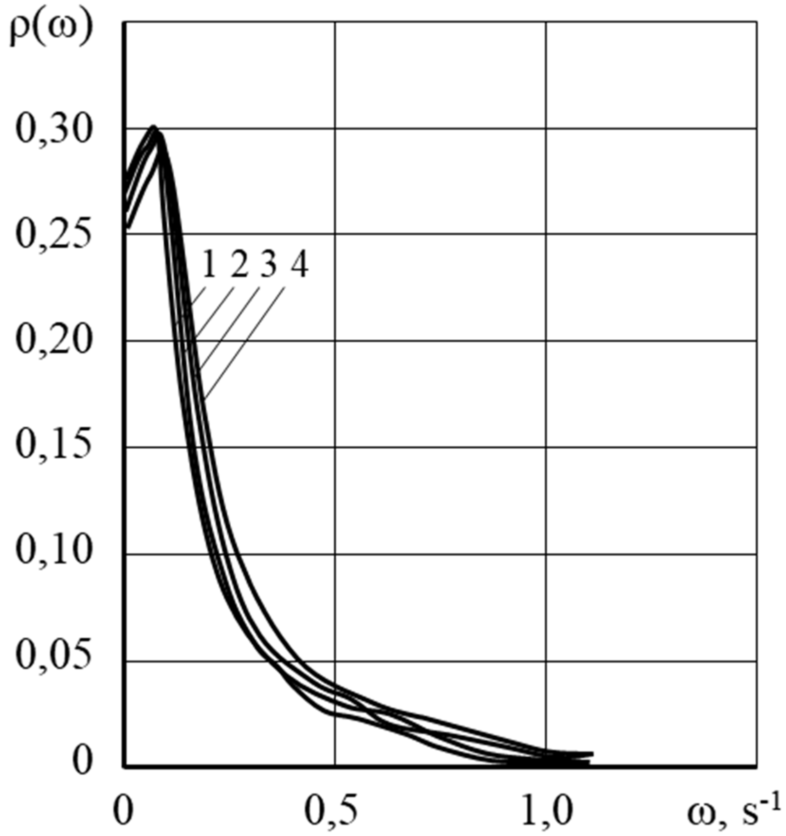

Fig. 3.9 – Normalized spectral densities of the angle of rotation α of the power module steering wheels when the plow is shifted laterally by 100 mm and the speeds of the plowing machine and tractor unit are: 1 – 5,8 km.h^{-1}; 2 – 6,8 km.h^{-1}; 3 – 7,7 km.h^{-1}; 4 – 9,3 km.h^{-1}

In relation to the perturbing influence, the spectrum of dispersion of the course angle φ of the power module for both plowing aggregates is narrower.

From all the above it follows that to improve controllability and stability of motion of a plowing unit on the basis of a modular power vehicle for universal plowing purpose, angular mobility in the horizontal plane of the technological module relative to the power module should be excluded.

As for plowing machine-tractor units based on general-purpose modular power vehicles, the requirement of rigid connection of the power and technological modules during the working process is fair in terms of stability of its movement.

However, most likely this is true only in theory, since the perturbing motion of such a unit is absolutely stable even in the free state of the vertical joint of the technological module.

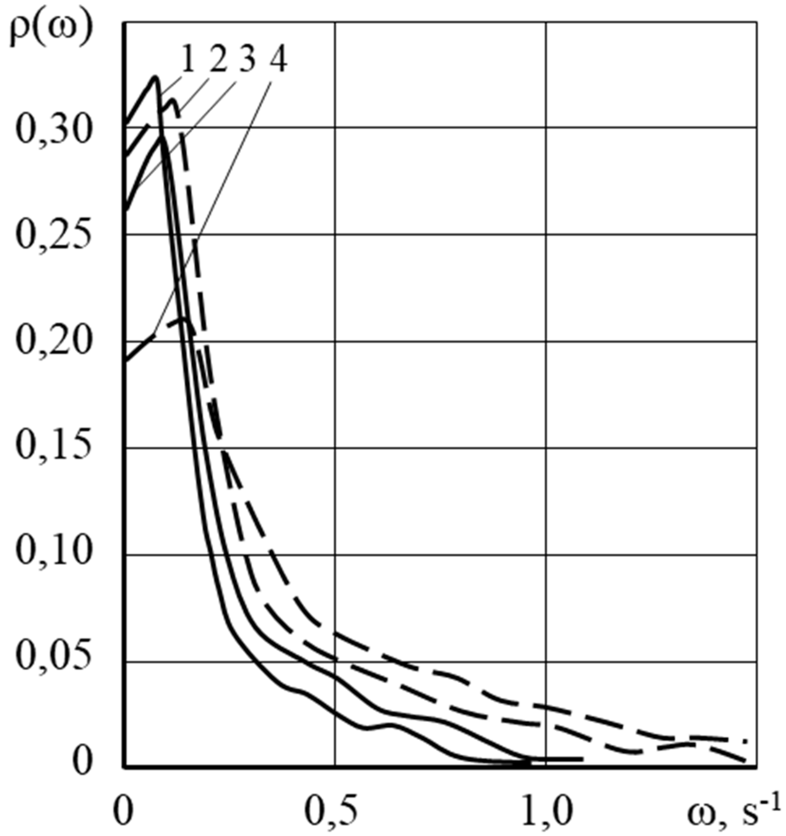

Fig. 3.10 – Normalized spectral densities of the angle of rotation β of five-bodied (—) and six-bodied (- - -) plows when the plow is shifted laterally by 300 mm and the speeds are: 1 – 6,0 km.h^{-1}; 2 – 5,5 km.h^{-1}; 3 – 7,5 km.h^{-1}; 4 – 6,4 km.h^{-1}

In the process of experimental studies it was found that the mean square deviation of the course angle of MES-300 power module front half-frame during work with PNL-8-40 plow is practically by one order of magnitude less than the value of this statistical parameter for disturbing influence, i.e. that of the plow rotation angle (Table 3.2). For example, the mean square deviation mentioned is approximately an order of magnitude less in the absence of a technological module, i.e. during operation of only one energy module MES-300 in the unit with PLN-5-35 plow.

The nature of the response of a general-purpose modular power vehicle to the control action of locking the vertical hinge of the technological module when working with almost any agricultural tool has a negative impact. The most acceptable variant in this respect is the synchronous rotation of the half-frames of the power module and the frame of the technological module.

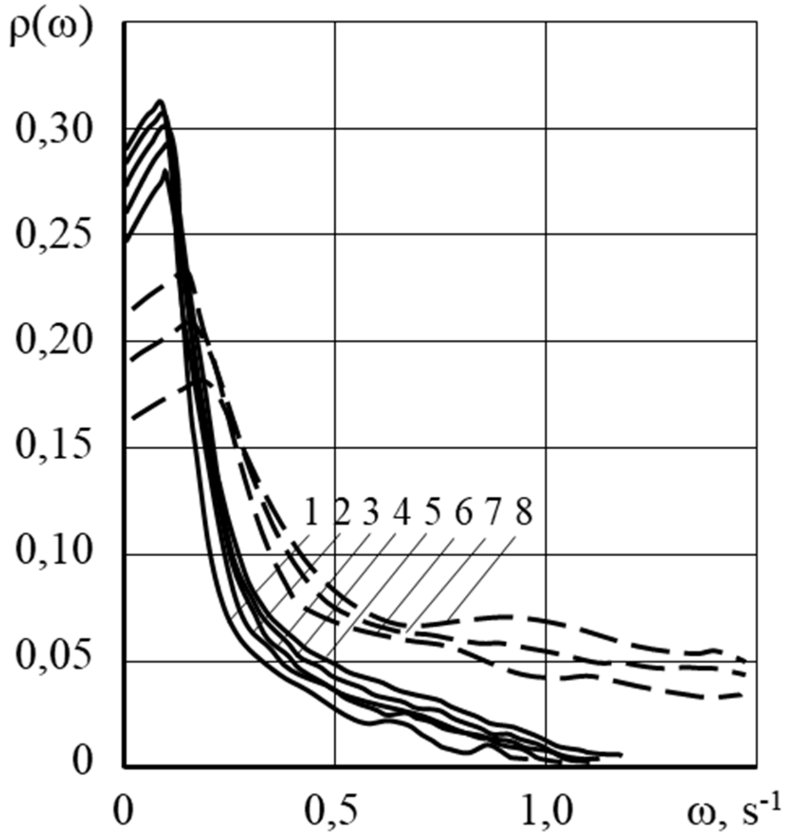

Fig. 3.11 – Normalized spectral densities of course angle φ of the power module when the modular power vehicle works with a five-bodied plow in locked (-----) and unlocked (- - -) states of the vertical hinge of the technological module: 1 – 6,0 km.h^{-1}; 2 – 6,4 km.h^{-1}; 3 – 7,3 km.h^{-1}; 4 – 8,5 km.h^{-1}; 5 – 9,2 km.h^{-1}; 6 – 7,7 km.h^{-1}; 7 – 9,1 km.h^{-1}; 8 – 9,3 km.h^{-1}

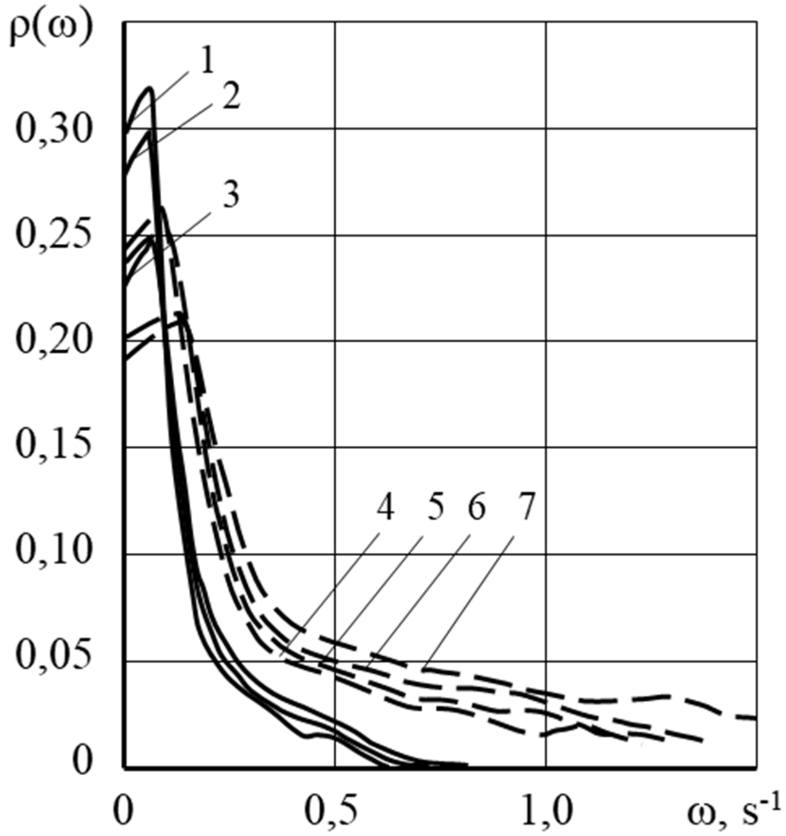

Fig. 3.12 – Normalized spectral densities of course angle φ of the power module when the modular power vehicle works with a six-bodied plow in locked (------) and unlocked (- - -) states of the vertical hinge of the technological module:
1 – 5,1 km.h^{-1}; 2 – 5,5 km.h^{-1}; 3 – 6,2 km.h^{-1}; 4 – 4,5 km.h^{-1};
5 – 5,6 km.h^{-1}; 6 – 6,0 km.h^{-1}; 7 – 6,4 km.h^{-1}

When the vertical joint of the technological module of the modular power tool is blocked, the plow must be connected to it in a pivoting manner. As to the general-purpose modular power vehicle, because in this variant the vehicle's own angular vibrations in the horizontal plane are transmitted to the process module to a lesser degree.

The articulated connection of the mounted plow can be made by adjusting the rear attachment of the technological module in a two-point or three-point pattern. Reconfiguration of the linkage is quite a time-consuming operation, so it is advisable to choose a single scheme with a subsequent significant simplification of the design of the power vehicle's rear linkage.

Table 3.2 – Movement parameters of MES-300
for wheat stubble plowing with PNL-8-40 plow [34]

Parameter	**Experiment**					
	1	**2**	**3**	**4**	**5**	**6**
Load at the hook, kN	45,8	47,7	57,1	58,9	61,9	73,3
Speed of movement, km·h^{-1}	7,20	7,40	7,16	7,12	6,50	5,30
Root-mean-square deviation of the course angle of the front half-frame of the energy module, rad.	0,0040	0,0039	0,0043	0,0040	0,0038	0,0049
Root-mean-square deviation of the angle of rotation of the technological module to the energy module, rad.	0,0071	0,0099	0,0068	0,0081	0,0071	0,0093
Root-mean-square deviation of plow rotation relative to the technological module, rad.	0,0147	0,0138	0,0130	0,0130	0,0134	0,0077

Actually, the two-point scheme provides a large angular mobility of the tool, as a result of which the stability of its movement in the horizontal plane depends little on the turns of the power vehicle. However, with a certain ratio of design parameters of the rear linkage it is possible to obtain the necessary angular mobility of the plow even with the three-point scheme.

In the course of experimental studies we found [35] that, in the case of connection of the modular power vehicle with the plow according to the three-point scheme, the stability of the vehicle is not violated when the convergence angle (α, Fig. 1.3) of the lower links of the technological module's rear linkage mechanism is no less than 0.4 rad. In this case, the internal structure of the plow's rotational angle change β (Fig. 3.13) is close to that of the two-point attachment (Fig. 3.10).

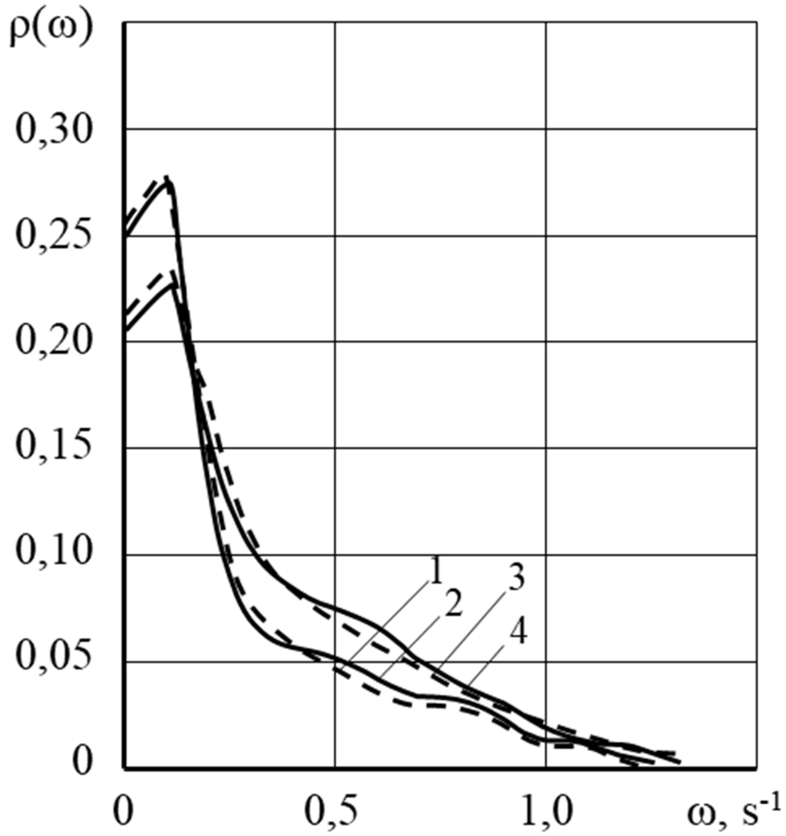

Fig. 3.13 – Normalized spectral densities of the rotational angle β of the plow, connected to MES-200 by the three-point scheme with blocked (----------) and free (- - - -) states of the vertical joint of the technological module and different movement speeds of the plowing unit:
1, 2 – 5,3 km.h^{-1}; 3, 4 – 7,2 km.h^{-1}

3.4. Ways to reduce the energy costs of a plowing machine-tractor unit based on a modular power vehicle

It was emphasized earlier that the level of energy consumption by a plowing unit is largely determined by the size and direction of the transverse movement of the "center of resistance" of the plow relative to the longitudinal axis of symmetry of the power vehicle.

In the course of experimental studies, it was found that the right-hand movement of the plow relative to its own link unambiguously leads to an increase in energy consumption by the plowing unit. For example, when moving the five-point plowing tool PLN-5-35 to the right by 100 mm, the traction resistance of the latter increased by 9 %, and the hourly fuel consumption of MES-200 increased by 5,6 % [36].

Transverse displacement of the plow together with the connecting device relative to the technological module does not lead to an increase in the traction resistance of the tool, although it increases the torque acting on the power vehicle in the horizontal plane. However, for a modular power vehicle, which has an increased longitudinal base, it is not so dangerous. Therefore, if for one reason or another it is not possible to connect the five-bodied plow to the modular power vehicle for universal plowing symmetrically, then its right side displacement should be carried out first by shifting along the hinged mechanism of the technological module together with the connecting device and only then (if the first displacement is not sufficient) – relative to its own connecting device.

Based on the analysis of expression (1.13), the value of the right-side transverse displacement of the coordinates of the "center of resistance" of the tool can be reduced mainly by changing the sum of the first two summands This is explained by the fact that with the design width of the plow's body the maximum value of $b_{к.}(n+1)$ is limited by the traction properties of the power vehicle, which, in turn, deteriorate with a decrease in the width b of its propellers. Because of this mutually opposing influence on each other, changing the third and fourth summands, unlike the first two, does not produce the desired effect.

From the above, we can conclude that to reduce energy consumption during plowing, the power vehicle, which is aggregated with the plow, should:

– have a narrow track;
– be characterized by high traction properties;
– move at the shortest possible distance from the furrow wall.

For practical implementation of the latter requirement, a sufficiently high stability of the plowing machine and tractor unit movement is required. Otherwise it is possible for the power vehicle to "slip" into the furrow. To prevent the latter, and to avoid crumbling of the furrow wall, it is recommended to choose the distance from it to the outer edge of the tractor engine to be equal to the plowing depth [1]. The results of many years of experimental studies have shown that plowing machine-tractor units based on serial power vehicles class 3 – 5, configured according to this recommendation, are significantly more energy-intensive (fuel consumption) than units based on the new modular power vehicles MES-200 and MES-300.

In order to explain such result let us study parameters of aggregation calculated by formula (1.13): T-150, T-150K and MES-200 with PLN-6-35 plow (Table 3.3), and K-701 and MES-300 with PTK-9-35 plow (Table 3.5).

With almost the same nominal pulling force as the T-150K tractor, the MES-200 has a narrower track (Table 3.3). The increased longitudinal base provides the new power vehicle with sufficiently high stability in the horizontal plane. This allows the MES-200 in combination with the plow to move within 100 mm of the furrow wall without breaking it or “sliding” into it.

Under typical conditions for arid environment i.e. low moisture and hardness of soil at the moment of plowing (8 ... 12 % and 2 ... 3 MPa accordingly) the traction resistance of a six-bodied plow even at its symmetric connection can reach such values that slipping of drawbar elements of traction class 3 used with this tool exceeds the agrotechnically permitted 20 % level. The right side lateral shift of the “center of resistance” of the plow in such soil conditions further complicates the situation, resulting in slippage of the wheel tractor T-150K engines, which can reach the maximum allowable value (30%) or even exceed it. For example, in the process of operational and technological evaluation of plowing units, the work of T-150K tractor with PLP-6-35 plow was impossible with soil moisture in 5 ... 15 cm layer – 6,5 ... 12,5 % and soil hardness – 1,00 ... 2,96 MPa. In this regard, the specified power vehicle was aggregated with the five-bodied tool.

Table 3.3 – Parameters of aggregation of power vehicles of class 3 with a PLP-6-35 plow

Indicator	**Power vehicle**		
	T-150	**T-150K**	**MES-200**
Longitudinal base, mm	1800	2860	5300
Track, mm	1435	1680	1400
Propeller width, mm	420	540	430
Recommended distance from the furrow wall to the outer edge of the propeller (A), mm	240	300	100
Number of plow bodies	6 – 5	6 – 5	6 – 5
Displacement of "center of resistance" of plow at A, mm	-57 – 117	185 – 360	-210 – -35
Distance from the furrow wall to the outer edge of the propeller with symmetrical plow attachment, mm	300	120 – 115	310 – 135

In similar soil conditions MES-200 could work with six-crop PLP-6-35 plow only because of the left-hand cross movement of the "center of resistance" of the tool by 100 mm. The pulling resistance of the plow was reduced by at least 5 %, and the hourly fuel consumption by 6 % or more (Table 3.4).

Removing the five field boards resulted in further reductions in traction resistance and hourly fuel consumption by 9,3 and 3,7 %, respectively.

Table 3.4 – Energy performance of a plowing unit based on MES-200 with a PLP-6-35 plow at different options of its attachment

Indicator	**Average value of the index at the value of left lateral displacement $e_п$ of the plow, mm**		
	$e_п$ = 0	**$e_п$ = 100**	**$e_п$ = 100 and without boards**
Pulling resistance of the plow, kN	47,7	45,2	41,0
Pulling resistance of the plow, kN	28,5	26,7	25,7
Slipping of propellers, %	24,8	20,0	18,5

Compared to the symmetrical ($e_п$ = 0) connection, the transverse left shift of a six-bodied plow by 100 mm and the removal of five field boards in total allowed to reduce the traction resistance of the tool by 14,0 % and hourly fuel consumption by 9,8 %. The MES-200 propeller slippage did not exceed 18,5 %.

Under favorable soil conditions, the symmetric connection of a PLP-6-35 plow to T-150K tractor is possible only when the distance from the furrow wall to the outer edge of its right wheels will be not more than 115 mm. However, as practice has shown, the process of movement of this plowing unit is characterized in this case by quite frequent “sliding” of the power vehicle into the furrow. An attempt to correct the situation by shifting the tractor to the left leads to poor mating of the adjacent machine-tractor passes.

As for caterpillar tractor T-150, its aggregation with PLP-6-35 plow, condition (1.13) is completely fulfilled even when working with recommended value A (Table 3.3). Symmetrical attachment of a five-bodied plowing tool is possible if the distance from the furrow wall to the edge of the right track will not be more than 120 mm. But in practice it is not always possible, because the tractor T-150, although close to MES-200 in traction properties and parameters of the undercarriage, is significantly inferior to it in the longitudinal stability.

A symmetrical connection of the plow is possible by putting the propellers of the right side of the power vehicle in the furrow. In foreign countries such a scheme of plowing unit is used quite often, although it has a number of serious disadvantages.

This is confirmed by laboratory and field studies of a plowing machine-tractor unit based on a modular power vehicle of universal-purpose plowing, which moved the wheels of the right side in the furrow and beyond its limits. When driving in a furrow (hereinafter – the first option), its traction force at 12% slip was 27,2 kN against 31,8 kN when working outside the furrow (the second option) with the same slip. The implementation of the first option of plowing machine-tractor unit operation is characterized by approximately the same (14 ... 15 %) reduction of traction force of the modular power vehicle in the whole investigated range of plow resistance (Fig. 3.14).

Reduced traction properties of the modular power vehicle in the first option of work had a corresponding effect on its fuel and energy performance. Thus, if the maximum conditional traction coefficient of efficiency η_y when the power vehicle was moving in the furrow was 51,5 %, then, when moving behind the furrow, this figure increased to 56,5 %. At the same value of η_y hourly fuel consumption of the modular power vehicle in the first option is 4,5 – 10,4 % higher than in the second option. In addition, during the movement of the modular power vehicle in the furrow, the bottom of the latter compacts, and it becomes wider. As a result, this leads to poor connection of adjacent arable unit passes. To eliminate this disadvantage, it is necessary to shift the plow transversely to the right side, which leads to the operation of the tool with incomplete width of its first body. And this has a corresponding effect on the performance of the machine, since the working width of the plow is reduced by 10 – 12 cm.

Like the MES-200 modular power vehicle, the MES-300 general-purpose modular power vehicle can be more effectively combined with a plow than an equal traction class wheeled tractor (e.g., K-701). Thus, if at the recommended value of A the nine-crop plough can be connected to both power tools with transverse left-hand shift, then the eight-crop tool with K-701 (unlike MES-300) can be connected only with right-hand shift (Table 3.5).

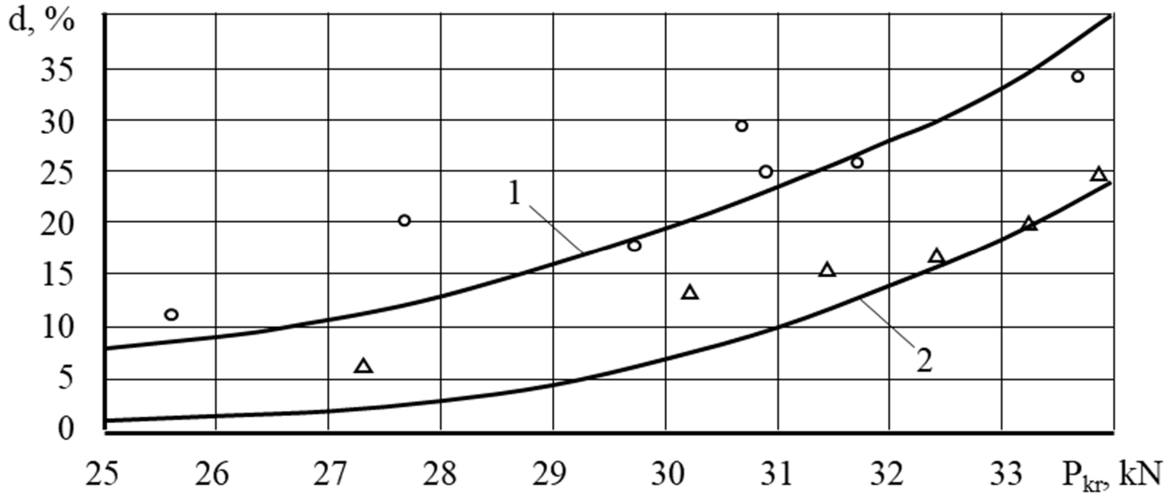

Fig. 3.14 – Slipping curves of the modular power vehicle when working with the plow in the furrow (1) and outside the furrow (2)

Table 3.5 – Parameters of aggregation of class 5 power vehicles with PTK-9-35 plow

Characteristic	MES-300	K-701
Longitudinal base, mm	5870	3200
Track, mm	1680	2115
Propeller width, mm	540	660
Recommended distance from the wall of the furrow to the outer edge of the propeller (A), mm	150	300
Number of plow bodies	9 – 8	9 – 8
Displacement of “center of resistance” of the plow at A, mm	-490 – -315	-62 – 112
Distance from the furrow wall to the outer edge of the wheel edge with symmetrical plow connection, mm	640 – 465	360 – 190

When plowing with an eight-bodied plow at a depth of 26 cm, the fuel consumption by the unit based on the MES-300 was 16,7 $kg.ha^{-1}$, and with a unit based on the K-701 wheeled tractor – 20,8 $kg.ha^{-1}$.

3.5. Influence of the scheme and parameters of a plowing tractor-machine unit on its operational, technological and agricultural performance

One of the main parameters affecting the operational and technological performance of a unit is its forward speed. In this regard, it is advisable to determine the degree of influence on changes in this indicator of the adjustment scheme and parameters of a plowing machine-tractor unit based on a modular power vehicle. First of all, it should be noted that all solutions, both schematic and design, which cause a reduction in energy consumption by the unit, usually lead to an increase in its speed and, therefore, in labor productivity. It is established that when reducing the right side cross displacement of a five-bodied plow relative to its own hitch by only 100 mm, the speed of the unit (due to reduction of traction resistance of the tool) increases by 2 – 4 %. In the case of displacement of the six-bodied plow to the left by 100 mm and removal of five field boards from it, the increase in speed of the plowing machine-tractor unit is 10,7 %. Only by blocking the vertical joint of the technological module, the speed of the modular power vehicle MES-200 in the unit with PLN-5-35 plow increased by 9,5 %. As a result, the shift productivity increased by 7,3 %, and the specific fuel consumption decreased by 3,7 % (Table 3.6).

The choice of optimal plowing unit composition has a significant impact on the production and technological performance of the unit. Thus, MES-200 aggregation with a five-bodied plow is reasonable when the traction resistance of the latter is less or equal to the nominal traction of the former. Otherwise, it is more advantageous to operate this power vehicle with a six-bodied plow. This is explained in the following way. If, for example, the resistance of a five-bodied plow is 32 kN, then the operating speed of a plowing machine and tractor unit at gear II-4 will be 9,1 km.h^{-1}, and productivity per 1 hour of regular time (with a working width of 1,75 m) – 1,59 hectares. Operation of MES-200 with a six-bodied plow at its minimum traction of 39 - 40 kN is possible only at the II-3 gear. The speed of a plowing machine-tractor unit is 7,2 km·h^{-1}, and productivity per hour of regular time (at a working width of 2,1 m) – 1,51 ha, which is 5 % less than in the first option. If the traction resistance of the five-crop plow is, say, 35 kN, the plowing unit cannot be operated with the II-4 transmission because of overloading. But when switching to gear II-3, the productivity of the plowing machine-tractor unit per hour of regular time will be 1,3 ha, which is 13,9 % less than when using MES-200 with a five-bodied plow.

Table 3.6 – Operational and technological performance of a plowing unit at different states of the vertical pivot of the technological module

Indicator	**Condition of the hinge of the technological module**	
	free	**blocked**
Composition of the unit:	MES-200 + PLN-6-35	
Terms and conditions of work:		
– operating speed, km·h^{-1}	7,00	7,67
– working width, m	2,14	2,10
– processing depth, cm	25 … 27	25 … 27
Productivity per 1 hour of work, ha		
– regular time	1,50	1,61
– shift time	1,29	1,31
Specific fuel consumption, kg·ha^{-1}	16,2	15,6

The above considerations are also fully valid with regard to the selection of the optimal composition of a plowing unit on the basis of a modular general-purpose power vehicle. When the resistance of the tool exceeds the nominal traction of the power vehicle (50 kN), it is advisable to use a nine-bodied (PTK-9-35) or eight-bodied (PNL-8-40) plow with a working width $B_п$ 3,15 and 3,20 m, respectively. Otherwise, one body should be subtracted from each of these plowing tools ($B_п$ = 2,8 m).

The above research results were used in the creation of plowing units based on mock samples of modular power vehicles to determine their operational and technological performance. During the tests, MES-200 was coupled with the PLP-6-35 plow with the first five field boards removed. The left lateral displacement of the latter relative to its own hitch was 100 mm, and the distance from the toe of the first body to the outer part of the wheel of the technological module was 125 mm. The attachment device of the technological module was set on the three-point scheme, and the locking of its vertical joint when driving on the paddock and unlocking on the turn lane was carried out by a special automatic device.

At the same time MES-300 was aggregated with PNL-8-40 plow, converted to semi-mounted version. The vertical pivot of the technological module of this power vehicle was not blocked.

The operating conditions of the machine-tractor unit were extreme: during the entire period of field tests, the average moisture content of the soil in the 0 - 15 cm layer was in the range of 8 - 12 %, and the hardness – 1,3 - 2,0 MPa.

The operational and technological indicators of the new plowing machine-tractor units were compared with similar indicators of basic machine-tractor units based on serial tractors T-150 and T-150K (Table 3.7).

Table 3.7 – Operational and technological indicators of plowing machine-tractor units

Plowing machine-tractor units parameters	MES-200 PLP-6-35	T-150 PLN-5-35	T-150K PLN-5-35	MES-300 PLN-8-40
Working conditions:				
– working width, m	2,14	1,75	1,80	2,80
– speed, $km \cdot h^{-1}$	7,01	6,80	6,55	7,15
Productivity, $ha \cdot h^{-1}$				
– regular	1,50	1,19	1,15	2,00
– shift	1,29	0,95	0,94	1,58
Fuel consumption, $kg \cdot ha^{-1}$	16,9	20,4	29,1	16,3
Utilization rate:				
– shift time	0,86	0,80	0,82	0,79
– working strokes	0,89	0,91	0,93	0,88
Width of turning lane, m	11,0	7,0	10,5	14,0
Agrotechnical parameters:				
– plowing depth, cm	28,3	26,6	27,5	28,8
– standard, ± cm	2,8	3,4	4,0	2,6
– variation coefficient, %	9,9	13,0	14,5	9,0

The latter were aggregated with a PLN-5-35 plow, as because of their insufficient traction properties, stable work with a six-bodied tool in the above-mentioned soil conditions was impossible.

The analysis of the obtained results showed that in comparison with the basic plowing machine-tractor units, the use of MES-200 allows increasing the shift productivity of the new unit by 35 – 37 %, and reducing fuel costs by 17 % (see Table 3.7).

When using the MES-300 power vehicle for plowing, the shift productivity of the new unit is 66,3 % higher compared with the plowing machine-tractor units based on the T-150 crawler tractor and 68,0 % higher compared with a unit based on the T-150K wheeled power vehicle. The main productivity in MES-300 is higher by 68,0 and 73,9 %, respectively. This is provided both due to larger (1,6 times) working width and higher (5 ... 9 %) travel speed.

As for fuel consumption per hectare, the plowing machine-tractor unit based on MES-300 has significantly lower fuel consumption. Compared to a unit based on a crawler tractor, the saving was 20,1 %, and for a wheeled tractor – 43,9 %.

In terms of shift productivity, a plowing machine-tractor unit based on MES-300 corresponds to a unit based on K-700A tractor, and in terms of specific fuel consumption it is more economical. The estimated reduction of this indicator is 20 %.

The main agro-technical indicators of the compared units are practically the same. The exception is the uniformity of plowing depth. According to agrotechnical requirements, fluctuations of this indicator, as it is known, should not exceed ±2 cm. This requirement was not met by any of the compared plowing machine-tractor units. However, the uniformity of the plowing depth is much higher for plowing machines based on a modular power vehicle. The reason for this result is explained by the minor relationship between the vertical oscillations of the power module and the technology module, which contributes to better copying of the longitudinal profile of the field surface by the modular power vehicle.

It was found experimentally that locking the vertical joint of the technological module, increases the controllability and stability of movement of the plowing machine-tractor unit, thereby improving the quality of its work. Only due to elimination of mutual angular mobility of power and technological modules in the horizontal plane, the mean square deviation of the width of a five-bodied plow decreased by 1,38 times, and of a six-bodied plow – by 2.1 times (Table 3.8).

Table 3.8 – Influence of the condition of the vertical joint of a technological module on the uniformity of the working width of a plowing machine-tractor unit

Indicator	MES-200 aggregated with a plow			
	PLN-5-35		PLP-6-35	
	state of the vertical hinge of the TM			
	free	blocked	free	blocked
Operating speed, $km \cdot h^{-1}$	9.0	9.0	6.3	6.4
Gripping width:				
– mean value, cm	176	176	212	213
– standard ± cm	5.4	3.9	6.7	3.2
– dispersion, cm^2	29.2	15.2	44.9	10.2

A noteworthy fact is that the operation of plow PLP-6-35 with one field board at its left-hand cross-shift has almost no effect on the uniformity of the working width of the plowing machine-tractor unit. Thus, the variance of this indicator was 17,64 cm^2, whereas when this plow is symmetrically connected to all field boards, it is 15,24 cm^2. Assessment of the significance of the degree of variation in the width of the arable unit, carried out using the Fisher F-criterion for the 5 % level of significance, shows that the zero hypothesis of equality of the compared dispersions is not rejected.

As a result of experimental studies, it was also found that the adjustment scheme of the rear attachment of the technological module does not significantly affect such a parameter as the uniformity of the working width of the plowing unit. This is confirmed by the fact that the zero hypothesis of equality of compared working width dispersions is not rejected by the example of aggregating the MES-200 modular power vehicle with PLN-5-35 connected by both two- and three-point schemes (Table 3.9).

Table 3.9 – Influence of the rear attachment adjustment scheme of a technological module on the uniformity of working width of an arable machine-tractor unit

Indicator	**Diagram for setting the rear linkage of the process module**	
	Two-point	**Three-point**
Operational speed, km·h^{-1}	9.0	8.9
Gripping width:		
– mean value, cm	176	177
– standard, ± cm	5.4	5.0
– dispersion, cm^2	29.16	25.0
– variation coefficient %	3.1	2.8

a b

Fig. 3.15 – Technological module MES-80:
a – view from the connection side of the power module;
b – view of rear linkage for plow attachment

Technical specifications of the MES-80

Operating weight (kg):	MTZ-80 tractor	3640
	technological module	2560
	MES-80	6200
Engine power, kW		62,0
Tractor power capacity, $kW \cdot t^{-1}$		17,0
MES-80 power capacity, $kW \cdot t^{-1}$		10,0
Distance between the rear wheels of the tractor and the wheels of the technological module, mm		2400
MES-80 undercarriage, mm		4770
Track of the undercarriage, mm		1450
Dimensions of the technological module tires		16,9 × 38

On the basis of universal tractor MTZ-80 we created a modular power vehicle of traction class 1,4 – 3 under the conventional brand MES-80. The technological module of this modular power vehicle in the front part has a coupling device by means of which it is connected to the rear tractor hitch (Fig. 3.15).

For aggregating with agricultural tools, the auxiliary axle is equipped with a hydraulic hitch system, PTO, fifth-wheel coupling, braking system, etc. During travel on the turning lane and when copying the track profile in transverse-vertical planes, the tractor and the technological module pivot are ensured by vertical and horizontal hinges. The adjustment of circular speeds of the wheels of the technological module and the rear wheels of the tractor is performed by a special reducer located on the frame of the additional axle.

The labor intensity of the technological module of the modular power vehicle connection with the tractor is 0,2 man-hours. Experimentally it was found that two mechanics spend no more than 6 minutes for this operation.

Table 3.10 – Operational and technological indicators
of plowing machine-tractor units

Indicator	Indicator value	
Composition of machine and tractor units: tractor plowing machine	 MEZ-80 PLN-4-35	 MTZ-80 PLN-3-35
Working conditions:		
– рабочая ширина захвата, m	1,45	1,05[1]
– plowing depth, cm	22,4	20 … 22[2]
– speed, $km.h^{-1}$	6,66	
Productivity, $ha.h^{-1}$		
– regular	0,97	
– shift	0,79	0,60[2]
– operational	0,79	
Costs:		
– labour, $man\text{-}h.ha^{-1}$	1,26	1,67[3]
– fuel, $kg.ha^{-1}$	14,35	17,8[2]
– metal, $t\text{-}h.ha^{-1}$	8,2	6,7[3]
Utilization coefficients:		
– shift time	0,82	
Operational time	0,82	
– reliability of the technological process	1,00	
– working strokes	0,84	
Average time of one turn, s	70,4	

[1]– constructive width of plow grip; [2]– regulatory indicators; [3]– estimated data.

Operational and technological tests of the plowing machine-tractor unit based on MES-80 were conducted in accordance with the procedure described in GOST 24055-88. The modular power unit was aggregated with PLN-4-35 plow (Fig. 3.16).

Fig. 3.16 – Modular power vehicle MES-80
in aggregate with PLN-4-35 plow

In the working paddock MES-80 moved with the right wheels outside the furrow. The distance from its wall to the right – side propellers did not exceed 15 cm. The mean square deviation of plowing depth, with its moisture in the layer 0 – 15 cm – 20 %, was 2.8, and the working width of the plowing machine – 3,4 cm.

The analysis of operational and technological data showed that the shift productivity of the new plowing machine-tractor unit is 31,7 % higher compared to the analogous normative index for the basic plowing machine-tractor unit consisting of MTZ-80 tractor and PLN-3-35 plow (Table 3.10).

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