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# STUDY OF THE KINEMATICS OF THE POSITIONING MECHANISMS OF WIDE-GRIP MACHINE-TRACTOR UNITS

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*As stated in the humanitarian food program "Grain from Ukraine", launched by the President of Ukraine Volodymyr Zelenskyi: "The demand for agricultural products in the world is growing by 2-3% every year, and the shortage of grain on the markets of Africa and Asia may reach 10-15% already in 2024". Therefore, the issue of production of agricultural products is relevant today. As the analysis of the market of agricultural machinery shows, one of the directions of the development of the design of agricultural units is to increase the width of grip of tools for soil cultivation and plant care. The use of agricultural units with an increased working width has certain advantages:*

*- reduction of the number of passes of equipment on the ground;*
*- reduction of fuel consumption;*
*- reducing the negative impact of the unit on soil density.*

*Among the disadvantages of this type of aggregates should be attributed the increase in turning radii, as a result of the increase in width. It is possible to eliminate this shortcoming by quickly transferring the unit from the working position to the transport position and vice versa. To fulfill this task, a review of known designs of wide-grip aggregates was carried out. Known hydraulic drives for raising and lowering and folding and unfolding sections are considered. A kinematic diagram of the mechanism of rotation of the extreme section of the cultivator relative to the frame has been developed. Based on this calculation scheme, the reactions in the hinges were determined and the graphical dependencies of the mutual location of individual elements with respect to the angle of rotation φ were calculated. The existing hydraulic positioning mechanisms, which are equipped with chainless wide-grip machines and tools with articulated frames, do not provide the necessary maneuvering properties of MTA during their movement on the turning lane. Equipping hydroficated positioning mechanisms with follower stabilizing devices is a promising direction for improving hydraulic drives of machine-tractor units, which create prerequisites for reducing the turnaround time on the turning lane by selecting the parameters of the specified equipment.*

*The most important factor affecting the stability of the positioning of the sections of wide-grip machine-tractor units is the variable load that occurs on the hydraulic cylinder rod at the moment of turning the sections around the horizontal hinges at an angle of more than 90◦, which causes inconsistency in the flows of the working fluid entering and leaving the hydraulic cylinder , which leads to the interruption of the flow in the cavity of the hydraulic cylinder, which is connected to the injection hydraulic line.*

***Key words:*** wide-grip unit, hydraulic drive, support, kinematics, reaction in the support.
***F. 30. Fig. 11. Ref. 14.***

## 1. Problem formulation

Analyzing trends in the development of the modern agricultural machinery market, one should note the tendency to increase the width of agricultural machinery. This design change is used by manufacturers to increase the productivity of agricultural units, reduce energy consumption and increase their productivity.

It is important to note that in the group of agricultural machines designed for the cultivation of row crops, there is no tendency to increase the working width. This is due to the technology of processing row crops, which requires the accuracy of matching the width of the seeder used during sowing to the width of the cultivator. It is also important to consider that the operator's attention is required to maintain the row, and as the working width increases, the complexity of the requirements for accuracy of sowing and subsequent processing of the row spacing increases, which can lead to premature fatigue of the operator. Taking into account the above, the working width of agricultural tools for such work is from 2.8 to 5.6 meters. [2].

The working width of soil tillage machines of another type can be up to 20 meters.

## 2. Analysis of recent research and publications

When designing wide-reaching aggregates, the question arises of ensuring their necessary maneuverability when making U-turns and moving between sections.

In the work of V. Sivolapov [1], it is stated that one of the main parameters that affect the productivity of machine-tractor units is the idle time associated with crossings, turns and starts.

The design features of the wide-grip machine-tractor unit mostly depend on the method of aggregation and transfer from the transport position to the working position and vice versa.

The width of the attachment structures, as a rule, does not exceed 6 meters. A further increase in the width of capture (more than 6 meters) for trailed planters is achieved by the use of trailing frames - hitches equipped with self-installing wheels and special hanging mechanisms. Transportation is carried out in the transverse direction due to the use of additional brackets and adjustable wheels located on one side of the agricultural unit. The other side of the implement rests on the tractor suspension. When transporting such weapons over long distances, they are installed on special carts.

Trailed solid-frame implements also have a grip width of up to 6 meters and are transported in the same way as trailed implements - in the transverse direction.

Sectional agricultural machine trailers with a working width of more than 6 meters are made according to an independent biaxial or dependent complex scheme.

An example of trailed agricultural machines, designed according to an independent two-axis scheme, are wide-reaching aggregates (Fig. 1.) of three or four cultivators assembled on the basis of SP-15 or SP-16 trailed hydraulic couplings for use with tractors of traction class 3...5 ( T-150, DT-75, etc.). When using the SP-16 hitch, which has all self-aligning wheels, and the row arrangement of tools, it is possible to implement a combined method of turning the wide-grip unit, which allows you to significantly reduce the size of the turning lanes.

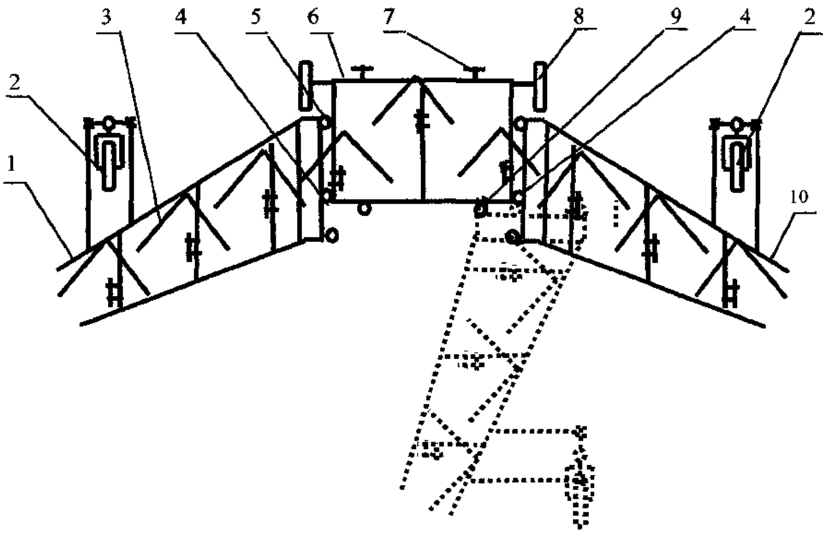



***Fig. 1. Schematic of a wide-reaching hitchless steam cultivator:***

***1 – extreme left section; 2 – self-installing wheel; 3 – flat cutting paw; 4, 5 – hinges; 6 – middle section; 7 – hanging; 8 – support wheel of the middle section; 9 – connecting (transport) hinge; 10 – extreme right section***

Before long-distance transportation, the unit is disassembled, the right and left wings of hitch 2 are brought back and fixed in this position, then the tool is attached to the hitch with a train.

It should be noted that such reconstructions require considerable expenditure of time, labor, and third-party assistance to the operator. In addition, row units with independent sections have increased material capacity, because for the use of sections as independent machines, all elements necessary for operation and transportation in a transverse position are included in the design of each section.

Agricultural machines with dependent complex sections have a more economical design, in which the support wheels are turned off on one side of each section. Hitchless cultivators have a specific metal capacity of up to 170 kg/m, while this indicator in units of two KPS-4 cultivators and a SP-11 hitch is about 240 kg/m [2].

In this case, the frame of the agricultural machine is made in the form of articulated sections, which allows to change the dimensions of the unit by turning and collapsing the sections relative to each other to ensure maneuverability in short and long-distance transport. Aggregates are folded both horizontally and vertically.

In the considered in fig. 2 structures for long-distance transportation on narrow roads, the extreme sections are wound back around the hinge 4 and fixed with the help of the connecting (transport) hinge 9. Moreover, the transfer of the unit from the working position to the transport position and back, as well as the connection and disconnection of the machine with the tractor are carried out one tractor driver.

There is a design of a wide-reaching hitchless sectional steam cultivator, which can be quickly transferred from the working position to the transport position and back in the vertical plane with the help of a hydraulic drive.

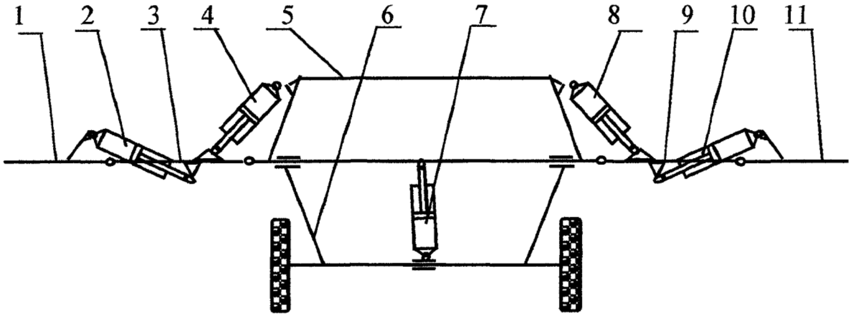



***Fig. 2. Schematic of a wide-grip hitchless steam cultivator:***

***1, 11 – extreme sections; 2, 10 – hydraulic cylinders for folding the extreme sections; 3, 9 – middle sections; 4, 8 – hydraulic cylinders for folding the middle sections; 5 – central section; 6 – lifting mechanism of the central section (moved to the transport position); 7 – hydraulic cylinder for lifting the central section***

The cultivator is intended for aggregation with tractors of class 3 (30...50 kN) and consists of an even number of separate sections 1, hinged together and equipped with pneumatic wheels 8, supporting two-wheeled carriages 9, towing equipment 2 in the form of two rods, with one end attached to the carriages 9, and the other end to the tractor hitch 3. The middle sections in front and behind are connected to each other by spacers 4, which have horizontal hinges. A slider I is installed on the vertical rack of the carriage, which is capable of axial movement and is connected to the extreme and middle sections by means of hinges 12. A hydraulic cylinder 6 is fixed on the rack, the rod of which is hinged to the extreme and middle sections by means of rods 5, and it has lower 13 and upper 14 stops to limit the movement of the slider 11.

An example of a wide-grip unit, the design of which provides for the possibility of turning the extreme sections in the vertical plane can serve as a hydrofied ring-toothed roller KZK – 10.

Structurally, the roller consists of five main sections: two extreme sections - 1 and 11, two middle sections - 3 and 9, and a central section - 5. In the working position, all sections are deployed relative to the second in a horizontal position.

To transfer to the transport position, the central section is raised with the help of hydraulic cylinder 7. Then, with the help of hydraulic cylinders 2 and 10, sections 1, 11 are turned by 90° and hydraulic cylinders 4, 8 ensure the rotation of sections 1 and 11 by another 90° and at the same time the total rotation of sections 1, 11 and 3, 9 at 90°.

The roller hydraulic drive system ensures the transfer of sections of the roller to the transport and working position when aggregating this tool with the hydraulic system of the tractor hitch.

KSHU - 12 hitchless cultivator can also serve as a generalized example of a wide-grip tillage tool, the design of which, with relatively complex kinematics, allows changing the transport dimensions with a stationary unit by one operator in a short period of time.

Cultivator KSHU-12 consists of five sections: central, two middle and two extreme. Transferring the cultivator to the transport position is carried out in the following sequence. First, the working bodies are raised by increasing the height of the wheel support under the central section. Next, the extreme sections are turned in the vertical plane by 180° and fixed relative to the middle sections. Then the middle sections (along with the outer ones) are turned in the vertical plane by 90° and fixed relative to the central section. Transferring the cultivator to the working position is carried out in the reverse order.

The hydraulic system presented in Fig. 3.

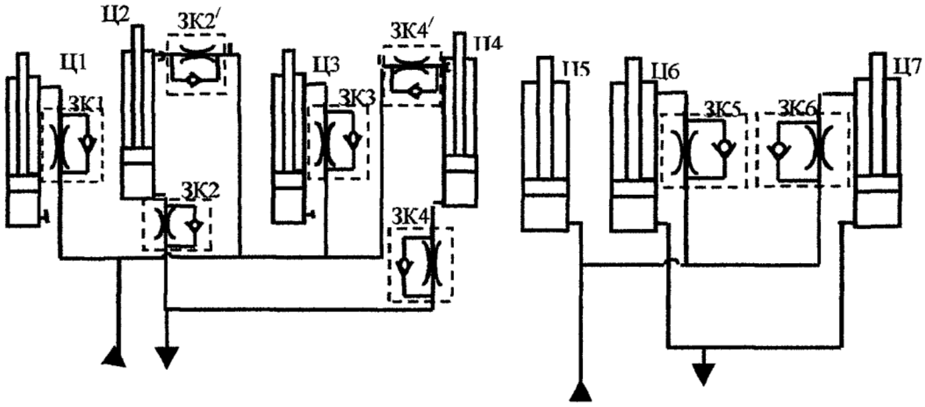



***Fig. 3. Basic hydraulic diagram of the KSHU-12 cultivator:***
***Ts1, Ts3 – hydraulic cylinders for turning the middle sections by 90°; Ts2, Ts4 – hydraulic cylinders for turning the extreme sections by 180°; Ts5, Ts7 – hydraulic cylinders for lifting working bodies; ЗК1-ЗК6 are retarding valves***

In hydraulic systems of KPZ-9.7, KSHU-12, KSHU-14, KChP-7.2 cultivators, the task of ensuring uniform movement of the section is solved by installing one-sided chokes with a constant cross-section or two-sided retarding chokes. This method is simple, but at the same time it significantly reduces the efficiency of the hydraulic system (for the KSHU-14, the efficiency during folding does not exceed 50%). This is due to the fact that the hydraulic resistance of each of the retarding throttles will be up to 7.2 MPa, which will lead to the loss of part of the pump's supply in the event of the inevitable activation of the protective valve-safety system. In addition, the use of chokes significantly affects the thermal balance of the system.

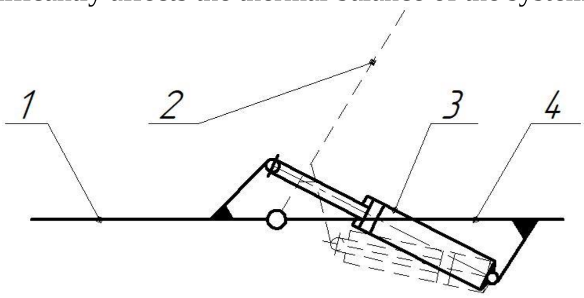



***Fig. 4. Kinematic diagram of folding of the extreme section of the KZK-10 roller:***
***1 – the folding section in the initial position; 2 – the section that is composed at the moment of switching the input signal; 3 – turning hydraulic cylinder; 4 – supporting structure***

A real reserve for increasing the productivity of wide-grip machine-tractor units is the improvement of their maneuvering properties.

The maneuverability of the unit with its modular structure can be significantly increased by turning (folding) the sections in the vertical plane.

In all known systems of spatial reorientation of sections of wide-reaching tillage machines in the vertical plane, a hydraulic drive is used as an executive mechanism.

When the sections are rotated in the vertical plane, the positional load perceived by the output link of the executive mechanism (hydraulic cylinder rod) changes in magnitude and can be directed both towards the movement of the output link (active load) and coincide with it (passive or secondary load).

## 3. Results of the researches

When identifying analytical dependencies that take into account the influence of kinematic parameters on the reaction force in the hinges, the rotary mechanism for positioning the extreme sections of the KSHU-12 cultivator was used as the basic design, since this scheme is the most typical for wide-grip tillage machines.

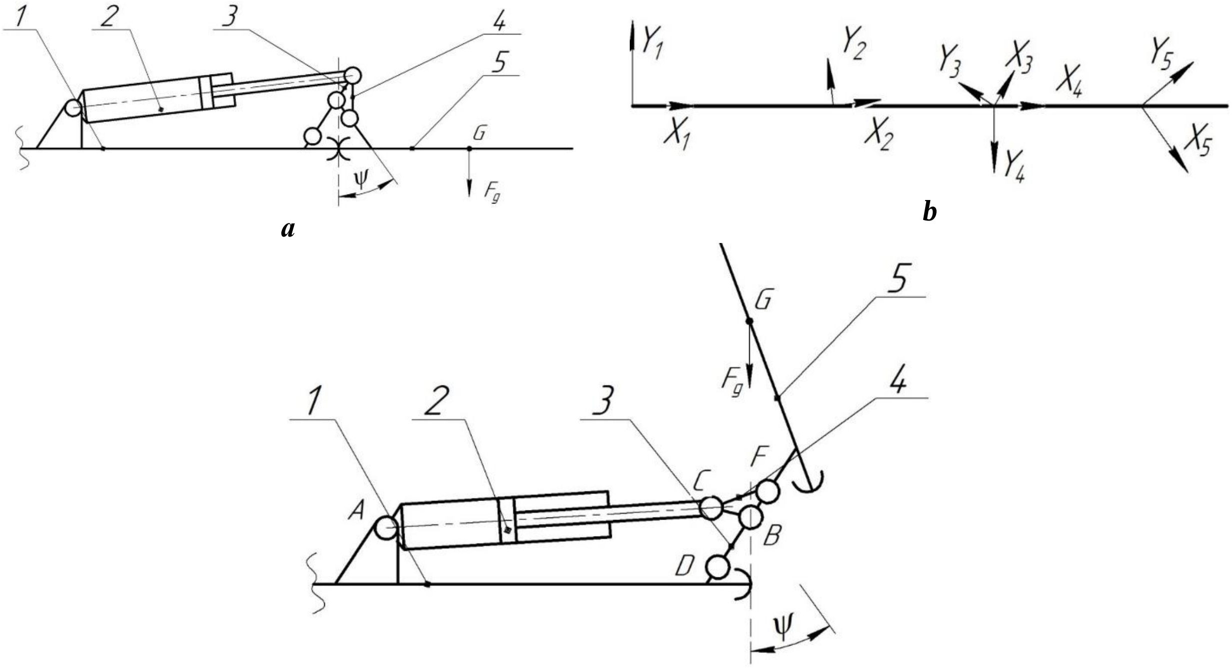



***Fig. 5. Scheme of the rotation mechanism of the extreme section of the KSHU-12 cultivator: a – initial position; b – diagram of the relative position of the accepted coordinate systems; c – current intermediate position; 1 – middle section; 2 – executive hydraulic cylinder; 3, 4 – traction; 5 – positionable section***

The following assumptions were made when compiling the model of the load change process:

- moments arising from the action of frictional forces were not taken into account, considering the frictional force in the hinge to be constant and independent of the angle of rotation of the section;
- the deflection of the links was not taken into account, assuming the structural elements to be solid, since the magnitude of the deflections is infinitesimally small compared to the movement during folding-unfolding;
- the forces of inertia were not taken into account, since the movement of the section at the moment of changing the sign of the load is carried out at a constant speed, and the mass of the moving rods (moving at a variable speed) is infinitely small compared to the mass of the section.

To describe the load law of the output link of the executive element, and in this case the force perceived by the hydraulic cylinder rod, the reaction forces in the hinges of the rotary mechanism should be determined.

In Fig. 7 presents simplified diagrams of the rotary mechanism of the KSHU-12 cultivator at the initial (a) and current (b) moments of time.

We isolate link 5 by introducing an additional reaction of the $R_4$ bond from link 4.

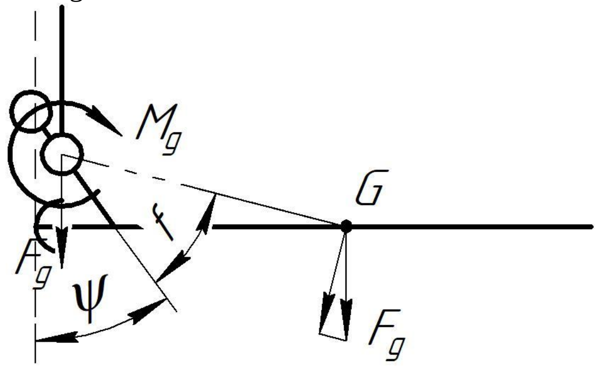



***Fig. 6. Diagram of transfer of the point of application of force Fg.***

After entering the coordinate system and moving the point of application of the force $F_g$ to the hinge $F$, we get the load diagram. When moving the point of application of the force $F_g$, the appearance of an additional moment should be taken into account $M_g$.

$$M_g = |FG| F_g \cdot \sin(\psi + \varphi + f) \tag{1}$$

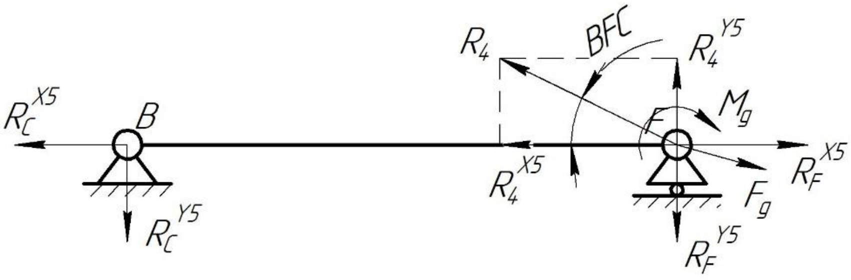



***Fig. 7. Calculation scheme of link 5***

Then the reaction of the connection $R_4$ can be determined from the dependence:

$$R_4 = \frac{R_F^{Y5}}{\sin(BFC)} \tag{2}$$

To determine the reaction forces of the support in the hinge $F$, we will compile a system of equations:

$$\text{Axis 0X5: } F_g \cdot \cos(\psi + \varphi) - R_C^{X5} + R_F^{X5} = 0 \tag{3}$$

$$\text{Axis OY5: } -F_g \cdot \sin(\psi + \varphi) - R_C^{Y5} + R_F^{Y5} = 0 \tag{4}$$

Consider the balance of moments with respect to the hinge $F$.

$$\text{MF: } M_g - R_C^{Y5} \cdot |BF| = 0 \tag{5}$$

Then from (4) and (5) we have

$$R_C^{Y5} = \frac{M_g}{|BF|} + F_g \cdot \sin(\psi + \varphi) \tag{6}$$

So, from equation (2) taking into account (1), we obtain:

$$R_4 = F_g \cdot \left( \frac{|FG| \cdot \sin(\psi + \varphi + f)}{|BF| \cdot \sin(BFC)} + \frac{\sin(\psi + \varphi)}{\sin(BFC)} \right) \tag{7}$$

To explain further calculations in Fig. 7 and Fig. 8 shows the load diagram of link 3, on which the reactions of the connection $R_2$ and $R_4$ with links 2 and 4 are entered, respectively.

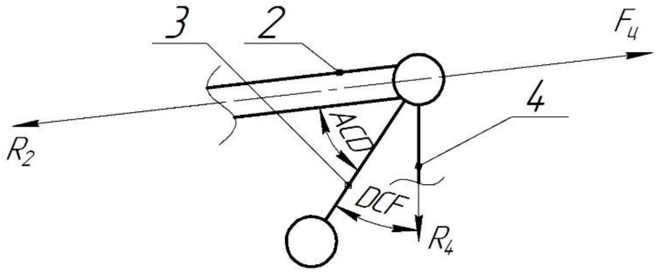



***Fig. 8. Explanatory scheme for the reduction of bond reactions***

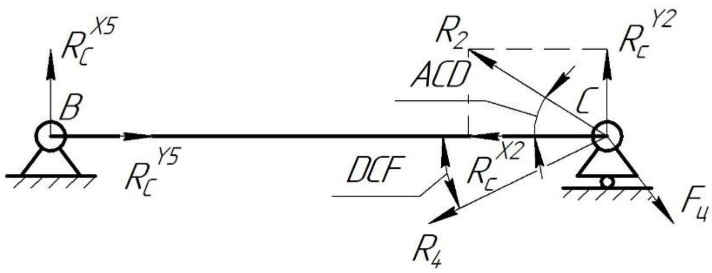



***Fig. 9. Calculation diagram of link 3***

In accordance with the scheme, the system of equilibrium equations of force projections on the coordinate axes have the form:

on the axis 0X2: $R_D^{X2} - R_C^{X2} - R_4^{X2} = 0$, (8)

on the axis 0B2: $R_D^{Y2} + R_C^{Y2} - R_4^{Y2} = 0$. (9)

Equilibrium of moments relative to the hinge 3:

$$\text{Mc: } R_D^{Y2} \cdot DC = 0, \tag{10}$$

So,

$$R_C^{Y2} = R_4^{Y2}. \tag{11}$$

The projection of the R4 force on the $Y_2$ axis is described by the dependence:

$$R_4^{Y2} = R_4 \cdot \sin(DCF). \tag{12}$$

In turn, the bond reaction can be found from the expression:

$$R_2 = \frac{R_C^{Y2}}{\sin(ACD)}, \tag{13}$$

$$R_2 = \frac{R_4 \cdot \sin(DCF)}{\sin(ACD)}. \tag{14}$$

Since this reaction of the connection is perceived only by the rod of the hydraulic cylinder, it can be written down:

$$F_ц = R_2 = \frac{F_g \sin(DCF)\left(|FG|\sin(\psi+\varphi+f)+|BF|\sin(\psi+\varphi)\right)}{|BF|\sin(BFC)\sin(ACD)}. \tag{15}$$

To find the angles DCF, BFC, ACD, we use the following ratios:

from a triangle DCF

$$|DF|^2 = |CD|^2 + |CF|^2 - 2|CD||CF|\cos(DCF), \tag{16}$$

from a triangle DBF

$$|DF|^2 = |BD|^2 + |BF|^2 - 2|BD||BF|\cos(DBF). \tag{17}$$

Moreover, the angle DBF is the sum of two angles:

$$DBF = 2\psi + \varphi. \tag{18}$$

Equating the right-hand sides of equations (16) and (17) and rearranging with respect to the DCF angle, we write

$$DCF = \arccos\left(\frac{|CD|^2 + |CF|^2 - |BD|^2 - |BF|^2 + 2|BD||BF|\cos(2\psi+\varphi)}{2|CD||CF|}\right).$$

The angle BFC can be found from the difference between the angles DFC and DFB:

$$BFC = DFC - DFB. \tag{19}$$

Where:

$$DFC = \arccos\left(\frac{|CF|^2 + |DF|^2 - |CD|^2}{2|CF||FD|}\right), \tag{20}$$

$$DFB = \arccos\left(\frac{|BF|^2 + |DF|^2 - |BD|^2}{2|BF||FD|}\right). \tag{21}$$

To find the angle ACD, we use the following relations:

$$ACD = \arccos\left(\frac{|AC|^2 + |CD|^2 - |AD|^2}{2|AC||CD|}\right). \tag{22}$$

In turn, the current distance of the AS is described by the ratio:

$$|AC| = \sqrt{|AD|^2 + |CD|^2 - 2|AD||CD|\cos(CDA)} \tag{23}$$

The angle *CDA* can be found from the difference between the angles *ADF* and *CDF*:

$$CDA = ADF - CDF\,. \tag{24}$$

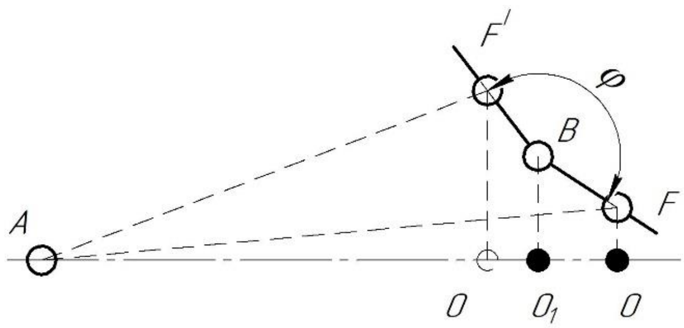
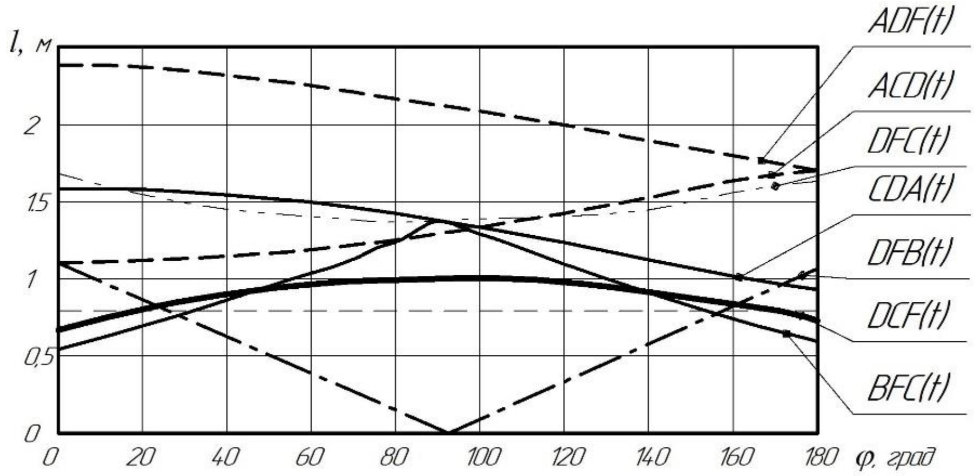
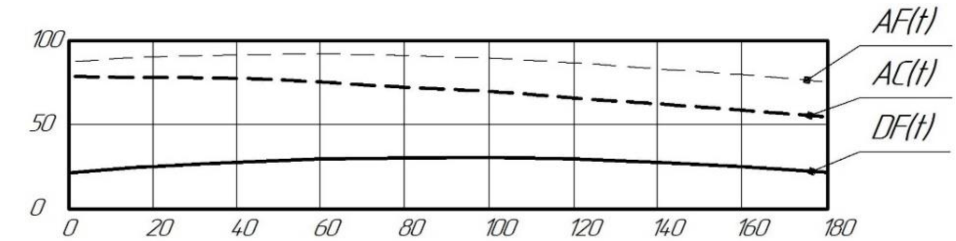


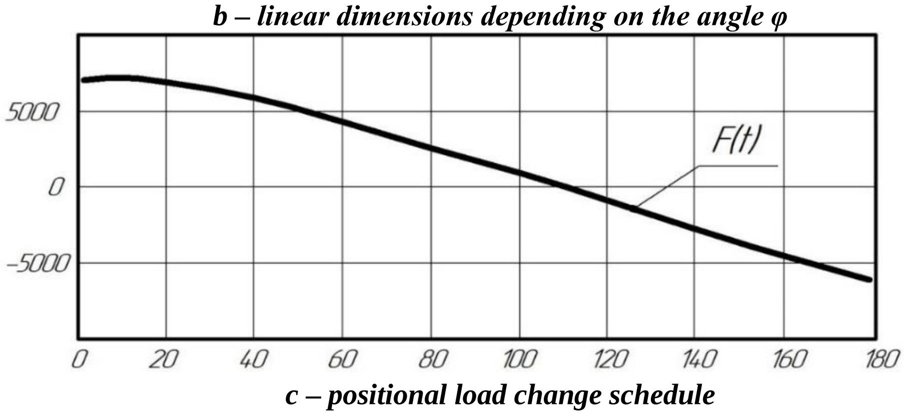

***Fig. 10. Scheme of conventional symbols for calculations of kinematic dependencies***




***a – relative location of individual elements depending on the angle φ***




***b – linear dimensions depending on the angle φ***




***c – positional load change schedule***

***Fig. 11. Graphs of changes in kinematic parameters depending on the angle of rotation of the section***

We find the angle *ADF* from the triangle *ADF*:

$$ADF = \arccos\left(\frac{|AD|^2 + |DF|^2 - |AF|^2}{2|AD||DF|}\right), \tag{25}$$

and the angle *CDF* can be found from the triangle *DCF*:

$$CDF = 180° - DCF - DFC\,. \tag{26}$$

The variable value AF included in expression (26) can be described by the following relationship:

$$|AF| = \sqrt{|AO|^2 - |FO|^2}\,. \tag{27}$$

where – $AO_1$ is a leg of a right-angled triangle *AOF* lying on the horizontal line *AO* (Fig. 8), the length of which is determined by the ratio

$$|AO| = |AO_1| + |O_1O|, \tag{28}$$

where $AO_1$ – design parameter (constant value); $O_1O$ – projection of the segment *BF* (lever 4, see Fig. 8) onto the horizontal line *AO*:

$$|O_1O| = |BF| \cdot \sin(\psi + \varphi)\,. \tag{29}$$

The segment *FO* is determined by the ratio:

$$|FO| = |O_1B| - |BF| \cdot \cos(\psi + \varphi)\,. \tag{30}$$

Calculations were performed on the basis of the obtained dependencies, and graphs of changes in kinematic parameters depending on the angle of rotation were obtained (Fig. 8).

## 5. Conclusions

The results obtained in the process of research made it possible to draw the following conclusions.

The existing hydraulic positioning mechanisms, which are equipped with chainless wide-grip machines and tools with articulated frames, do not provide the necessary maneuvering properties of MTA during their movement on the turning lane. Equipping the hydroficated positioning mechanisms of the follower with stabilizing devices is a promising direction for the improvement of hydraulic drives of the MTA, which create prerequisites for reducing the turnaround time on the turning lane by selecting the parameters of the specified equipment.

The most important factor affecting the stability of the positioning of the sections of wide-grip MTA is the variable load that occurs on the hydraulic cylinder rod at the moment of turning the sections around the horizontal hinges at an angle of more than 90, which causes inconsistency of the flows of the working fluid entering and leaving the hydraulic cylinder, which leads to to the interruption of the flow in the cavity of the HC, connected to the injection hydraulic line. In the presence of retarder valves, flow coordination is achieved by increasing the energy consumption of the pump drive.

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## ДОСЛІДЖЕННЯ КІНЕМАТИКИ МЕХАНІЗМІВ ПОЗИЦІОНУВАННЯ ШИРОКОЗАХВАТНИХ МАШИННО-ТРАКТОРНИХ АГРЕГАТІВ.

*Як зазначено у гуманітарній продовольчій програмі «Grain from Ukraine», що започаткована президентом України Володимиром Зеленським: «Потреба в сільськогосподарській продукції у світі щороку зростає на 2-3%, а дефіцит зерна на ринках Африки та Азії може сягнути 10-15% вже у 2024». Отже питання виробництва сільськогосподарської продукції на сьогодні є актуальним. Як свідчить аналіз ринку сільськогосподарської техніки, одним із напрямків розвитку конструкції сільськогосподарських агрегатів, є збільшення ширини захвату знарядь для обробітку грунту та догляду за рослинами. Застосування сільськогосподарських агрегатів із збільшеною шириною захвату, має певні переваги:*

- *зменшення кількості проходів техніки по грунту;*
- *зменшення витрати палива;*
- *зменшення негативного впливу агрегату на щільність грунту.*

*До недоліків даного виду агрегатів, слід віднести збільшення радіусів повороту, у наслідок збільшення ширини. Усунути даний недолік, можливо за допомогою швидкого переведення агрегату із робочого положення у транспортне і навпаки. Для виконання даної задачі проведено огляд відомих конструкцій широкозахватних агрегатів. Розглянуто відомі гідравлічні приводи підйому та опускання і згортання та розгортання секцій. Розроблено кінематичну схему механізму повороту крайньої секції культиватора відносно рами. Базуючись на даній розрахунковій схемі визначені реакції у шарнірах та розраховані графічні залежності взаємного розташування окремих елементів відносно кута повороту φ. Існуючі гідрофіковані механізми позиціонування, якими оснащують безланцюгові широкозахватні машини і знаряддя із шарнірно-зчленованими рамами, не забезпечують необхідні маневрові властивості МТА під час їх руху на поворотній смузі. Оснащення гідрофікованих механізмів позиціонування слідкуючим стабілізуючими пристроями є перспективним напрямком* *удосконалювання гідроприводів машинно тракторних агрегатів, що створюють передумови для зниження часу виконання розвороту на поворотній смузі добором параметрів зазначених обладнань.*

*Найважливішим фактором, що впливають на стабільність виконання позиціонування секцій широкозахватних машинно-тракторних агрегатів є знакозмінне навантаження, що виникає на штоку гідроциліндра в момент повороту секцій навколо горизонтальних шарнірів на кут більш 90˚, що викликає неузгодженість потоків робочої рідини, які входять і виходять з гідроциліндра, що й приводить до розриву потоку в порожнині гідравлічного циліндра, яка з'єднана з нагнітальною гідролінією.*

***Ключові слова:*** *широкозахватний агрегат, гідравлічний привод, опора, кінематика, реакція у опорі.*

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