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Study of the controlled motion process of an agricultural wide span vehicle fitted with an automatic driving device

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Abstract. The aim of research is to analyse the process of the wide span vehicle motion on the treads of the permanent process track with the use of the traction method of turn. The completed studies have proved that the plane-parallel motion of a wide span vehicle solely with the use of the traction method of turn on the treads of the permanent process track requires a significantly smaller difference between the moments applied to the wheels on the right side and on the left side of the vehicle, as compared to that of a crawler tractor. This difference is in proportion to the width of its wheel base that has a relatively smaller length. It has been established that the use of solely traction-based turn does not provide for the adjustment of the motion trajectory through the lateral (plane-parallel) displacement of the fore-and-aft axis of the agricultural wide span vehicle. The oscillations in the lateral displacement of the agricultural wide span vehicle in the process of its motion are low-frequency ones. The main variance spectrum of these oscillations is concentrated within the range of frequencies of 0–2 s^{-1}. At the same time, the maximum transverse displacement of the agricultural wide span vehicle equal to 1.3 10^{-5} m N^{-1}, occurs also at low frequencies. In view of the fact that the main variance spectrum of the oscillations of the tangential forces applied to the wheels on the left and right sides of the agricultural wide span vehicle is concentrated exactly within a low frequency range, only the high accuracy of the system using a laser beam can ensure its satisfactory steerability.

Key words: accuracy, automatic driving, controlled traffic farming, wide span vehicle.

INTRODUCTION

The future development of agricultural wide span vehicles suggests the automation of their plane-parallel motion on the treads of the permanent process track (Kingwell & Fuchsbichler, 2011). This aspect becomes especially topical, when wide span vehicles are turned using the traction method (drive wheels on the right and left sides have different rates of revolution).

Farming with the use of a permanent process track implies the allocation of special zones in the field area intended for the motion of agricultural machinery and segregated from the zones containing cultivated plants (Nevens & Reheul, 2003; Hamza & Anderson, 2005; Chan et al., 2006; Gasso et al., 2013; Pedersen et al., 2013; Antille et al., 2015; Chamen, 2015; Keller et al., 2019). With such a layout, high grip-and-traction characteristics of its running gear are achieved due to the latter's contact with the dry and hard surface of the soil. That is, in terms of its functional purpose, the field area is divided into the fertile (agronomic) and process zones (Raper, 2005; Kingswell & Fuchsbichler, 2011; Onal, 2012; Antille et al., 2019).

In terms of engineering, it is most feasible to implement the above-mentioned farming strategy by way of developing the new design layout of the tractor, so called wide span (gantry) vehicle (Onal, 2012; Bulgakov et al., 2019). Its principal difference from other types of power units is that it travels on the treads of the permanent process track. The distance between the treads is equal to the span of the tractor and all the agricultural implements are situated within this span (Bulgakov et al., 2018). The worldwide experience of the use of wide span vehicles has proved that they have the following advantages in comparison with conventional tractors: the power savings in soil tillage processes reach 55%; the cost of crop drilling operations is reduced by 40%; the tillage quality and the soil structure are improved; the loss of the field area under the travelling agricultural machinery is reduced to a minimum; high accuracy is achieved in the positioning of the tools; the field operations become highly automated; the yield of the cultivated plants increases by 7–10% (Bulgakov et al., 2017).

On the other hand, the tractor steering control issues come into importance in case of wide span vehicles. In view of the very wide transverse wheel base, it becomes difficult for the driver-operator to control the motion trajectories of the wheels on the left and right sides. At the same time, it is common knowledge that the steering systems of today's wheeled vehicles are usually tailored for manual control. Also, these systems can be divided into two classes: they use either kinematic or forced principle of turning. The kinematic type of turning implies the angular displacement of either the steer wheels (front wheels, rear wheels or both of them simultaneously) relative to the frame of the machine or one part of the mobile unit's frame relative to its other part in the horizontal plane (articulated frame). The turning by (traction) force is implemented by way of rotating the wheels on different sides of the machine at different rates. The steering systems based on the use of steer wheels are the most widely used ones. That said, when all the wheels of the machine are steerable, they can simultaneously turn on the front axle and the rear axle to the opposite sides or to the same side. Sometimes combined kinematic-and-traction layouts are implemented or proposed in order to improve the steerability of mobile machines (Chebrolu et al., 2017; Pascuzzi et al., 2020; Kuvachov et al., 2021). However, the design and functioning conditions of the agricultural wide span vehicles intended for controlled traffic farming systems differ to some extent from those of other state-of-the-art wheeled machines. The assumptions established in the classical theory of motion for mobile power units are not applicable to the discussed agricultural wide span vehicles.

At the same time, it ought to be noted that considerable developments have recently been observed in the implementation of self-driving systems for mobile power and transport units (Pascuzzi et al., 2020). Depending on the result of the measurement, an automatic action is taken, which is aimed at reducing the said difference down to an acceptably small value. The above-mentioned self-control systems can work on the basis of different principles of operation. In particular, they can use GPS systems, short- and long-range radars, short-, middle- and long-range distance finders based on visible-light and infrared lasers, video cameras that monitor the forward view panorama and analyse the obtained images, etc. (Ji & Zhou, 2014; Cerruto et al., 2018).

The problem of automating the process of driving mobile machines amounts to the problem of developing the set of devices that will control the motion of the agricultural wide span vehicle on the treads of the permanent process track without an operator (Yang et al., 2015). Solving this problem has proved to be a rather difficult task. First of all, due to the relatively limited possibility of manoeuvring in motion and the action of a great number of the random perturbing factors that create all kinds of possible situations, to which the system has to respond by putting in action the respective controls (steering gear, brakes, engine control etc.). However, the authors believe that the principle of turning by force (traction) can be efficient in the automated control of the gantry tractor travelling within the limiting dimensions of the treads of the permanent process track. As is known, this principle is implemented by rotating the wheels on different sides of the machine at different rates of revolution.

The use of GPS equipment for the automated steering of mobile agricultural integrated machines has gained great popularity. The technical excellence of the state-of-the-art instrumentation in terms of its accuracy in reproducing the pre-set motion trajectories of the mobile machines is very impressive (Griepentrog, 2009; Bakker et al., 2011; Longo & Muscato, 2013; Zhu et al., 2016; Guerrieri et al., 2019).

As it is known from the classical tractor theory, for any mobile power tool, the ratio of its longitudinal base to the transverse base - $(L \cdot K^{-1})$, called the relative support base, is an important parameter characterizing its stable turns during the technological process in agricultural production. The significant by magnitude and direction external impacts from the soil and the cultivated plants tend to constantly change the trajectory of the movement of the agricultural equipment. For most serial agricultural tractors and other power tools, used in agricultural production, industry and transport, the value of the relative support base, as a rule, is greater than 1. However, the layout scheme of any wide span agricultural vehicle, due to its wide track and small wheelbase, a priori assumes that this value is less than 1. Because of this, the value of the stabilizing moment in the onboard method of turning any skeleton agricultural vehicle, caused by the moment of the lateral forces of interaction of its wheels with the supporting surface, is much less than the value of the moment of the forces caused by the difference in the sum of the driving moments, supplied to the wheels of its left and right sides in a horizontal plane. The process of the onboard method of turning such power tools with a small relative support base is insufficiently studied. For a more complete understanding of the process of the onboard method of turning a wide span agricultural vehicle, theoretical and experimental investigations are required.

The aim of this work was to theoretically analyze the dependencies of the dynamic and kinematic parameters that rule the optimal swerving of the developed agricultural wide span vehicle for changing its trajectory.

MATERIALS AND METHODS

An agricultural wide span vehicle (Fig. 1) has a number of features that significantly distinguish it from other types of power vehicles used in agriculture, industry and transport. This is, first of all, a significant by its dimensions transverse base (track) - K, formed by a frame, at the ends of which there are the left and the right sides. Since the sides of the vehicle are located at a considerable distance from each other, this makes it necessary to provide an individual drive for their running wheels installed in pairs at each side in the form of the front and the rear driving wheels, moving along the same track (Bulgakov et al., 2020a). The stability of the movement of this wide span agricultural vehicle must be ensured by appropriate dimensions of the longitudinal base - L, which is especially important since the working bodies of the agricultural machines are located on the frame between the two sides or strictly on the transverse axis of symmetry of the agricultural vehicle, or at some distance from this axis.

Figure 1. The agricultural wide span vehicle during the test (Bulgakov et al., 2020b).

A wide span agricultural vehicle makes a plane-parallel movement in a horizontal plane, parallel to the field surface, strictly following the traces of the technological track. However, in reality, in the process of the movement, under the influence of various external disturbances there are characteristic deviations of the wide span agricultural vehicle from the rectilinear movement along the traces of a constant technological track in a horizontal plane. They cause deviations from the given direction not different magnitude and direction. In this case the following takes place: Δx – its transverse deviation, $\Delta\varphi$ – its angular heading deviation. The nature of the disturbances fully depends on the design scheme of the wide span agricultural vehicle and its technological purpose.

To describe the plane-parallel movement of the wide span agricultural vehicle with an onboard method of its rotation, we will draw up an equivalent diagram in a horizontal plane that will simulate this type of its movement. In a simplified schematic representation of the wide span agricultural vehicle, we denote by the corresponding letters the characteristic points of the unit considered. These will be: point S_t – the center of mass of the agricultural vehicle; L_1 and L_2 – the centers of the front and rear wheels of the left side of the agricultural vehicle, respectively; R_1 and R_2 – the centers of the front and rear wheels on the starboard side, respectively; C – the point of attachment to the agricultural vehicle of the agricultural machine. The considered wide span agricultural vehicle has transverse and longitudinal axes of symmetry (Bulgakov et al., 2018), which, intersecting at S_t, form its center of mass (Fig. 2). Let us indicate on the equivalent diagram the design parameters of this wide span agricultural vehicle: K – the transverse base (track) of the agricultural vehicle (the distance between the longitudinal axes of the left and right sides); L – longitudinal base of the agricultural vehicle (distance between the centers of the front and rear wheels of the left and right sides; b – distance from the point of attachment C of the agricultural machine to the longitudinal axis of the left side; l_t – distance from the center of the rear wheel of the left and right sides to the transverse axis of symmetry of the agricultural vehicle passing through its center masses (point S_t).

Further, for an analytical description of the plane-parallel movement of the agricultural vehicle, we introduce the necessary coordinate systems xOy. First we will show the fixed Cartesian coordinate system xOy, rigidly connected with the surface of the field. Here axis Oy of the indicated coordinate system is directed towards the forward movement of the agricultural vehicle, axis Ox – to the right, in the direction of the movement of this wide span unit. We assume that the discussed agricultural vehicle, equipped with some kind of an agricultural machine, performs uniform forward movement at speed V_0 on the rut relative to the fixed horizontal plane xOy (which is connected with the plane of the field surface) (Fig. 2).

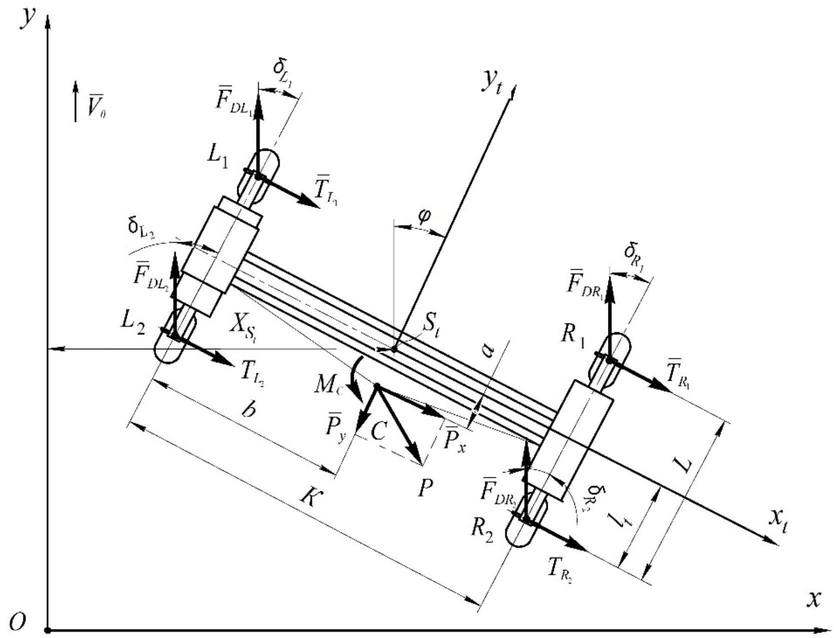

Figure 2. Equivalent diagram of a wide span agricultural vehicle with an onboard turning method during its plane-parallel movement in a horizontal plane.

We will also introduce an additional rectangular Cartesian coordinate system $x_tS_ty_t$, rigidly connected with the agricultural tool, the starting point of which is located at its center of mass (point S_t). In addition, we will direct axis S_ty_t of this coordinate system along the longitudinal axis of symmetry of the agricultural tool, axis S_tx_t will be directed to the right in the course of its movement along the transverse axis of symmetry of the agricultural device.

When performing an agricultural technological process, the skeleton of a wide span agricultural vehicle deviates from its original position under the impact of external random disturbances, receiving additional speeds. As a result, the forward transverse movement of the agricultural vehicle along axis Ox and its rotation around the center of mass (point S_t) at a certain angle φ (heading angle) relative to the coordinate system xOy in the horizontal plane begins. Therefore, the previously indicated coordinate system $x_t S_t y_t$ serves to describe the rotation of the agricultural tool around point S_t by angle φ in plane xOy, where φ is its heading angle, namely, the angle between axes Oy and $S_t y_t$ (Fig. 2). In addition, the origin of the coordinate system $x_t S_t y_t$ (point S_t), associated with the skeleton of the wide span agricultural vehicle, moves in a transverse direction (along axis Ox). The position of point S_t at an arbitrary point of time t relative to axis Oy is indicated by abscissa X_{S_t} (Fig. 2). This means that in the process of relative movement of the wide span agricultural vehicle, its center of mass S_t moves in the direction of axis xO, which is characterized by a change in the abscissa X_{S_t}. Thus the considered agricultural tool in relation to plane xOy has two degrees of freedom, which correspond to two generalized coordinates, namely: angle φ of its rotation around the center of mass, and a change in abscissa X_{S_t} of its center of mass S_t.

The external forces acting upon the agricultural vehicle during its plane-parallel movement in a horizontal plane (Fig. 2) include:

1) the driving tangential forces F_{DL_1}, F_{DL_2}, F_{DR_1} and F_{DR_2} of the wheels of the left and right sides of the agricultural vehicle, applied, respectively, the points L_1, L_2 and R_1, R_2 that form slip angles δ_{L_1}, δ_{L_2}, and δ_{R_1}, δ_{R_2}, with its longitudinal axis of symmetry;

2) the lateral forces T_{L_1}, T_{L_2}, T_{R_1} and T_{R_2}, applied, respectively, at points L_1, L_2, and R_1, R_2;

3) the longitudinal P_y and transverse P_x components of traction resistance, applied at point C, that constitute the main moment M_C of forces, from the side of the agricultural implements.

For a mathematical description of the lateral interaction of any mobile power tool with an agricultural background, the 'lateral slip' hypothesis in a linear interpretation is most often used. In this case, to determine the lateral horizontal forces at the points of contact of the wheels with the soil, the tire slip resistance coefficients are used (Bulgakov et al., 2020a and 2020b). Using this provision, for this bridge agricultural vehicle, we can write:

$$\begin{aligned} T_{L_1} &= k_{L_1} \cdot \delta_{L_1}, \\ T_{L_2} &= k_{L_2} \cdot \delta_{L_2}, \\ T_{R_1} &= k_{R_1} \cdot \delta_{R_1}, \\ T_{R_2} &= k_{R_2} \cdot \delta_{R_2}, \end{aligned} \tag{1}$$

where k_{L_1}, k_{L_2} and, k_{R_1}, k_{R_2} are the slip resistance coefficients of the left and the right sides of the agricultural vehicle, respectively.

The method for deriving expressions for calculating slip angles δ_{L_1}, δ_{L_2}, and δ_{R_1}, first of all involves determining the absolute values of the velocities of points L_1, L_2 and R_1, R_2. These speeds can be determined from the expressions (Fig. 3):

$$\begin{aligned} \overline{V}_{L_1} &= \overline{V}_{0} + \overline{V}_{S_t} + \overline{V}_{L_1 S_t}, \\ \overline{V}_{L_2} &= \overline{V}_{0} + \overline{V}_{S_t} + \overline{V}_{L_2 S_t}, \\ \overline{V}_{R_1} &= \overline{V}_{0} + \overline{V}_{S_t} + \overline{V}_{R_1 S_t}, \\ \overline{V}_{R_2} &= \overline{V}_{0} + \overline{V}_{S_t} + \overline{V}_{R_2 S_t}, \end{aligned} \tag{2}$$

where $\bar{V}_0$ is the vector of the linear forward speed of the movement of the wide span agricultural vehicle relative to plane xOy; $\bar{V}_{S_t}$ – the linear velocity vector of the center of mass of the agricultural vehicle relative to plane xOy; $\bar{V}_{L_1S_t}$, $\bar{V}_{L_2S_t}$ and $\bar{V}_{R_1S_t}$, $\bar{V}_{R_2S_t}$ are the linear velocity vectors of centers L_1, L_2 and R_1, R_2 of each wheel of the agricultural vehicle relative to its center of mass (point S_t) in a horizontal plane (Fig. 2).

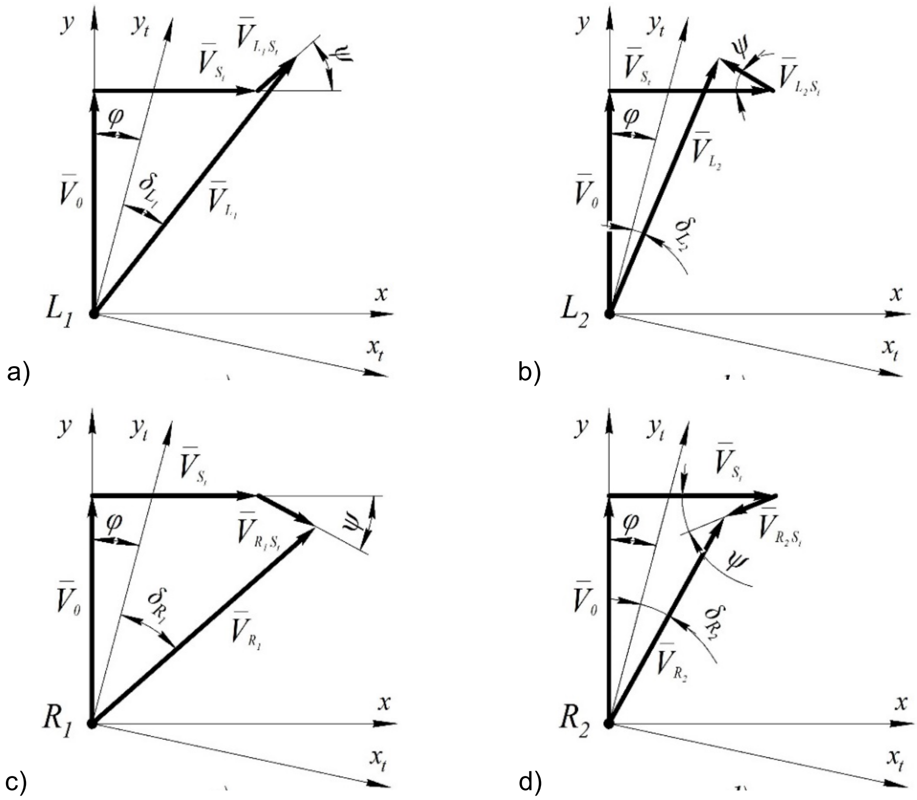

Figure 3. The vector sums of the speeds of the centers of the wheels of the wide span agricultural vehicle: a) – the front wheel of the left side with center L_1; b) – the rear wheel of the left side with center L_2; c) – the front wheel of the starboard side with center R_1; d) – the rear wheel of the starboard side with the center R_2.

To determine the tangents of the slip angles of the wheels (points L_1, L_2, and R_1), it is necessary to project the sums of vectors (Fig. 3) on and axes S_tx_t and S_ty_t, and take the ratio of the obtained sums of projections. As it is evident from Fig. 3, the sums of the projections of the velocity vectors, shown in Fig. 3, a; 3, b; 3, c and 3, d will be respectively equal:

$$\begin{aligned}
V_{L_1x_t} &= -V_0 \cdot \sin\varphi + V_{S_t} \cdot \cos\varphi + V_{L_1S_t} \cdot \cos(\varphi+\psi),\\
V_{L_2x_t} &= -V_0 \cdot \sin\varphi + V_{S_t} \cdot \cos\varphi - V_{L_2S_t} \cdot \cos(\varphi-\psi),\\
V_{R_1x_t} &= -V_0 \cdot \sin\varphi + V_{S_t} \cdot \cos\varphi + V_{R_1S_t} \cdot \cos(\varphi-\psi),\\
V_{R_2x_t} &= -V_0 \cdot \sin\varphi + V_{S_t} \cdot \cos\varphi - V_{R_2S_t} \cdot \cos(\varphi+\psi),
\end{aligned} \tag{3}$$

where ψ is the angle between the velocity vectors $\bar{V}_{L_1S_t}$, $\bar{V}_{L_2S_t}$, $\bar{V}_{R_1S_t}$, and $\bar{V}_{R_2S_t}$, and the horizontal axis.

In addition, taking into account the smallness of angles φ and ψ, we assume that $\sin\varphi \approx \varphi$, $\cos\varphi \approx 1$, and $\cos(\varphi + \psi) \approx 1$. Therefore, expressions (3) can be written in a more simplified form. Namely:

$$\begin{aligned} V_{L_1x_t} &= -V_\text{o} \cdot \varphi + V_{S_t} + V_{L_1S_t}, \\ V_{L_2x_t} &= -V_\text{o} \cdot \varphi + V_{S_t} - V_{L_2S_t}, \\ V_{R_1x_t} &= -V_\text{o} \cdot \varphi + V_{S_t} + V_{R_1S_t}, \\ V_{R_2x_t} &= -V_\text{o} \cdot \varphi + V_{S_t} - V_{R_2S_t}. \end{aligned} \tag{4}$$

Further, it is obvious that the projection of the velocity vector $\bar{V}_{L_1y_t}$ of center L_1 of the front wheel of the left side of the agricultural vehicle on axle S_ty_t will be equal to:

$$V_{L_1y_t} = V_\text{o} \cdot \cos\varphi + \dot{X}_{S_t} \cdot \sin\varphi \tag{5}$$

Considering that in the second term on the right side of expression (5) the product of two small quantities is a small quantity of a higher order, there is every reason to write that $\bar{V}_{L_1y_1} \approx V_0$.

In a similar way we have:

$$V_{L_1y_t} = V_{L_2y_t} = V_{R_1y_t} = V_{R_2y_t} = V_\text{o} \tag{6}$$

Next, we determine the values (modules) of the vectors included in expression (2). It's obvious that:

$$\left|\bar{V}_\text{o}\right| = V_\text{o}, \; \left|\bar{V}_{S_t}\right| = \dot{X}_{S_t}. \tag{7}$$

To determine the values of the linear speeds of rotation of centers L_1, L_2, R_1, and R_2 of the wheels of the agricultural vehicle relative to its center of mass (point S_t), we find the radii of the tuns of points L_1, L_2, R_1, and R_2 relative to point S_t. As it evident from Fig. 3, the specified radii will be equal to:

$$\begin{aligned} R_{L_1} &= R_{R_1} = \frac{1}{2}\sqrt{4 \cdot (L - l_t)^2 + K^2}, \\ R_{L_2} &= R_{R_2} = \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2}. \end{aligned} \tag{8}$$

Then, taking into account expression (8), we obtain:

$$\begin{aligned} \left|V_{L_1S_t}\right| &= \left|V_{R_1S_t}\right| = \frac{1}{2}\sqrt{4 \cdot (L - l_t)^2 + K^2} \cdot \dot{\varphi}, \\ \left|V_{L_2S_t}\right| &= \left|V_{R_2S_t}\right| = \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2} \cdot \dot{\varphi}. \end{aligned} \tag{9}$$

After substitution of expressions (7) and (9) into expression (4), we will have:

$$\begin{aligned} \left|V_{L_1x_t}\right| &= \left|V_{R_1x_t}\right| = -V_\text{o} \cdot \varphi + \dot{X}_{S_t} + \frac{1}{2}\sqrt{4 \cdot (L - l_t)^2 + K^2} \cdot \dot{\varphi}, \\ \left|V_{L_2x_t}\right| &= \left|V_{R_2x_t}\right| = -V_\text{o} \cdot \varphi + \dot{X}_{S_t} - \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2} \cdot \dot{\varphi}. \end{aligned} \tag{10}$$

Thus, on the basis of the obtained expressions (6) and (10), it is possible to determine the values of the tangents of the slip angles δ_{L_1}, δ_{L_2}, and δ_{R_1}, δ_{R_2}, and, considering the smallness of these angles, and the slip angles themselves of all the four wheels of the considered agricultural vehicle. Namely:

$$\tan\delta_{L_1} = \tan\delta_{R_1} \approx \delta_{L_1} \approx \delta_{R_1} = \frac{V_{L_1 x_t}}{V_{L_1 y_t}} = \frac{V_{R_1 x_t}}{V_{R_1 y_t}} =$$
$$= \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\cdot\left(L-l_t\right)^2 + K^2}\cdot\dot{\varphi} - V_o\cdot\varphi}{V_o},$$
$$\tan\delta_{L_2} = \tan\delta_{R_2} \approx \delta_{L_2} \approx \delta_{R_2} = \frac{V_{L_2 x_t}}{V_{L_2 y_t}} = \frac{V_{R_2 x_t}}{V_{R_2 y_t}} =$$
$$= \frac{\dot{X}_{S_1} - \frac{1}{2}\sqrt{4\cdot l_t^2 + K^2}\cdot\dot{\varphi} - V_o\cdot\varphi}{V_o}. \quad (11)$$

Since the slip angles δ_{L_1}, δ_{L_2}, and δ_{R_1}, δ_{R_2}, are directed in the other direction with respect to the positive direction of change of the heading angle φ (Fig. 2), their values must be negative. With this in mind, we finally obtain:

$$\delta_{L_1} = \delta_{R_1} = \varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\cdot\left(L-l_t\right)^2 + K^2}\cdot\dot{\varphi}}{V_o},$$
$$\delta_{L_2} = \delta_{R_2} = \varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4\cdot l_t^2 + K^2}\cdot\dot{\varphi}}{V_o}. \quad (12)$$

Having determined the slip angles δ_{L_1}, δ_{L_2}, and δ_{R_1}, δ_{R_2}, we find the lateral forces:

$$T_{L_1} = k_{L_1}\cdot\left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\cdot\left(L-l_t\right)^2 + K^2}\cdot\dot{\varphi}}{V_o}\right],$$
$$T_{L_2} = k_{L_2}\cdot\left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4\cdot l_t^2 + K^2}\cdot\dot{\varphi}}{V_o}\right],$$
$$T_{R_1} = k_{R_1}\cdot\left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\cdot\left(L-l_t\right)^2 + K^2}\cdot\dot{\varphi}}{V_o}\right],$$
$$T_{R_2} = k_{R_2}\cdot\left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4\cdot l_t^2 + K^2}\cdot\dot{\varphi}}{V_o}\right]. \quad (13)$$

From the study of automatic driving and driving wheels (Bulgakov et al., 2021), it is known that different turning torques M_{ki} must be applied, which will provide different values of the tangential forces F_{Di}:

$$F_{Di} = \frac{(M_{ki} - M_{fi})}{r_i}, \quad (14)$$

where M_{ki} is the turning torque applied to the i-th wheel; r_i is the dynamic radius of the i-th wheel; M_{fi} is the rolling resistance moment of the i-th wheel.

Due to the difference in the sum of the driving moments M_k applied to the left and right wheels of the agricultural wide span vehicle, a torque M_p appears in a horizontal plane, given by Bulgakov et al. (2021):

$$M_p = \frac{1}{2}K\left(F_{DL_1} + F_{DL_2} - F_{Dr_1} - F_{Dr_2}\right) \neq 0, \quad (15)$$

where F_{DL_1}, F_{DL_2} and F_{DR_1}, F_{DR_2} are the tangential forces applied, respectively, to the front and rear wheels of the left and right sides of the agricultural wide span vehicle.

The appearance of the torque M_p causes a change in the direction of the motion of the agricultural wide span vehicle. The resistance to turning of the machine causes a moment of lateral interaction forces of the wheels with the supporting surface M_T, which can be considered a stabilizing moment, the main moment M_C of forces acting from the side of the agricultural implements, the moment of the transverse component P_x of the traction resistance of agricultural implements and the moment of inertia forces M_J:

$$M_p = M_T + M_C + M_J = \frac{1}{2}L\cdot\left(T_{L_1} + T_{R_1} - T_{L_2} - T_{R_2}\right) + M_C + P_x \cdot a + J_m \cdot \ddot{\varphi}\,, \quad (16)$$

where M_T is the torque caused by the lateral interaction forces of the wheels with the supporting surface; M_J is the moment of inertia; T_{L_1}, T_{L_2}, and T_{R_1}, T_{R_2} are the lateral forces applied, respectively, to the front and the rear wheels of the left and right sides of the agricultural wide span vehicle; M_C – the main moment of forces acting upon from the side of the agricultural machine; J_m – is the moment of inertia of the agricultural wide span vehicle in a horizontal plane; $\ddot{\varphi}$ is the angular acceleration of the agricultural wide span vehicle in a horizontal plane.

When the turn is stable, the following equality is satisfied:

$$\frac{1}{2}K\cdot\left(F_{DL_1} + F_{DL_2} - F_{DR_1} - F_{DR_2}\right) = \frac{1}{2}L\cdot\left(T_{L1} + T_{R1} - T_{L2} - T_{R_2}\right) + M_C + P_x \cdot a + J_m \cdot \ddot{\varphi}\,. \quad (17)$$

By substituting expressions (13) into expression (17), after some transformations, the following differential equation is obtained:

$$J_m \cdot \ddot{\varphi} = \frac{1}{2}K\cdot\left(F_{DL_1} + F_{DL_2} - F_{DR_1} - F_{DR_2}\right) - \frac{1}{2}L\cdot\left\{\left(k_{L_1} + k_{R_1}\right)\cdot\left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{\left(L-l_t\right)^2 + K^2}\cdot\dot{\varphi}}{V_o}\right] - \right.$$
$$\left. -\left(k_{L_2} + k_{R_2}\right)\cdot\left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{l_t^2 + K^2}\cdot\dot{\varphi}}{V_o}\right]\right\} - M_C - P_x \cdot a. \quad (18)$$

Eq. (18) is a differential equation for the turn of the wide span agricultural vehicle around its center of mass (point S_t) in its plane-parallel motion.

Further we will compose a differential equation for the transverse displacement X_{S_t} of the center of mass S_t of the considered agricultural vehicle, using the basic law of dynamics, which, in this case, will be written in the following form:

$$M \cdot \ddot{X}_{S_t} = \sum_{k=1}^{n} F_{kx}\,, \quad (19)$$

where M is the mass of the agricultural vehicle; $\ddot{X}_{S_t}$ – acceleration of the center of mass S_t

of the agricultural vehicle in the direction of axis Ox; $\sum_{k=1}^{n} F_{kx}$ – the sum of the projections of all the external forces acting upon the unit under consideration, on axis Ox.

Using the equivalent circuit, shown in Fig. 3, we can find the sum of the projections of all the external forces on axis Ox. We have:

$$\begin{aligned}\sum_{k=1}^{n} F_{kx} &= -F_{DL_1} \cdot \sin\left(\delta_{L_1} - \varphi\right) - F_{DL_2} \cdot \sin\left(\delta_{L_2} - \varphi\right) - \\ &- F_{DR_1} \cdot \sin\left(\delta_{R_1} - \varphi\right) - F_{DR_2} \cdot \sin\left(\delta_{R_2} - \varphi\right) + \\ &+ \left(T_{L_1} + T_{L_2} + T_{R_1} + T_{R_2}\right)\cos\varphi + P_x \cdot \cos\varphi - P_y \cdot \sin\varphi .\end{aligned} \tag{20}$$

Taking into account the smallness of angles φ, δ_{L_1}, δ_{L_2}, δ_{R_1} and δ_{R_2}, expression (20) can be written in a more simplified form:

$$\begin{aligned}\sum_{k=1}^{n} F_{kx} &= -F_{DL_1} \cdot \left(\delta_{L_1} - \varphi\right) - F_{DL_2} \cdot \left(\delta_{L_2} - \varphi\right) - F_{DR_1} \cdot \left(\delta_{R_1} - \varphi\right) - \\ &- F_{DR_2} \cdot \left(\delta_{R_2} - \varphi\right) + T_{L_1} + T_{L_2} + T_{R_1} + T_{R_2} + P_x - P_y \cdot \varphi .\end{aligned} \tag{21}$$

By substituting expressions (12) and (13) into expression (21) and performing a series of transformations, we will obtain the final expression for the sum of the projections of all external forces, acting upon the discussed agricultural vehicle, to axis Ox:

$$\begin{aligned}\sum_{k=1}^{n} F_{kx} &= \left(F_{DL_1} + F_{DR_1}\right) \cdot \left[\frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L - l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \\ &+ \left(F_{DL_2} + F_{DR_2}\right) \cdot \left[\frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \\ &+ \left(k_{L_1} + k_{R_1}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L - l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \\ &+ \left(k_{L_2} + k_{R_2}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + P_x - P_y \cdot \varphi .\end{aligned} \tag{22}$$

Substituting expression (22) into Eq. (19), we obtain the desired differential equation for the transverse movement X_{S_t} of the considered agricultural tool in the direction of axis Ox:

$$M \cdot \ddot{X}_{S_t} = \left(F_{DL_1} + F_{DR_1}\right) \cdot \left[\frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L-l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] +$$
$$+\left(F_{DL_2} + F_{DR_2}\right) \cdot \left[\frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] +$$
$$+\left(k_{L_1} + k_{R_1}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L-l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \tag{23}$$
$$+\left(k_{L_2} + k_{R_2}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + P_x - P_y \cdot \varphi .$$

Combining differential Eqs (23) and (18) into one system of differential equations, we obtain a mathematical model of a plane-parallel movement of the wide span agricultural vehicle with an onboard method of its turning. We have:

$$\left.\begin{aligned}
&M \cdot \ddot{X}_{S_t} = \left(F_{DL_1} + F_{DR_1}\right) \cdot \left[\frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L-l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \left(F_{DL_2} + F_{DR_2}\right) \cdot \left[\frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 \cdot l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \\
&+\left(k_{L_1} + k_{R_1}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{4\left(L-l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + \left(k_{L_2} + k_{R_2}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{4 l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] + P_x - P_y \cdot \varphi , \\
&J_m \cdot \ddot{\varphi} = \frac{1}{2} K \cdot \left(F_{DL_1} + F_{DL_2} - F_{DR_1} - F_{DR_2}\right) - \frac{1}{2} L \cdot \left\{\left(k_{L_1} + k_{R_1}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} + \frac{1}{2}\sqrt{\left(L-l_t\right)^2 + K^2} \cdot \dot{\varphi}}{V_o}\right] - \right. \\
&\left. -\left(k_{L_2} + k_{R_2}\right) \cdot \left[\varphi - \frac{\dot{X}_{S_t} - \frac{1}{2}\sqrt{l_t^2 + K^2} \cdot \dot{\varphi}}{V_o}\right]\right\} - M_C - P_x \cdot a .
\end{aligned}\right\} \tag{24}$$

If we consider that the slip resistance coefficients of the front and the rear wheels of the left and the right sides of the agricultural vehicle are equal, then we can assume as a first approximation that:

$$k_{L_1} = k_{L_2} = k_L \text{ and } k_{R_1} = k_{R_2} = k_R , \tag{25}$$

then, substituting values (25) into the system of differential equations (24), after a series of transformations, we obtain a system of differential equations of the following form:

$$
\left.
\begin{aligned}
&M\cdot\ddot{X}_{S_t}=\left(F_{DL_1}+F_{DR_1}\right)\cdot\left[\frac{\dot{X}_{S_t}+\frac{1}{2}\sqrt{4\left(L-l_t\right)^2+K^2}\cdot\dot{\varphi}}{V_o}\right]+\left(F_{DL_2}+F_{DR_2}\right)\cdot\left[\frac{\dot{X}_{S_t}-\frac{1}{2}\sqrt{4\cdot l_t^2+K^2}\cdot\dot{\varphi}}{V_o}\right]+\\
&+2\left(k_L+k_R\right)\cdot\left(\varphi-\frac{\dot{X}_{S_t}}{V_o}\right)+\frac{1}{2}\cdot\left(k_L+k_R\right)\cdot\left[\frac{\sqrt{4\left(L-l_t\right)^2+K^2}-\sqrt{4l_t^2+K^2}}{V_o}\right]\cdot\dot{\varphi}+P_x-P_y\cdot\varphi,\\
&J_m\cdot\ddot{\varphi}=\frac{1}{2}K\cdot\left(F_{DL_1}+F_{DL_2}-F_{DR_1}-F_{DR_2}\right)+\frac{1}{2}\cdot\frac{\left(k_L+k_R\right)\cdot L^2}{V_o}\cdot\dot{\varphi}-M_C-P_x\cdot a.
\end{aligned}
\right\}\quad(26)
$$

For the case when the front and the rear wheels of the right and the left sides of the agricultural vehicle are at the same distance from its transverse axis of symmetry, which finds its reflection in the design of the wide span agricultural vehicle, developed by us (see Fig. 1), then we can assume that $l_t = L^{-2}$. Then the system of differential equations (26) takes the final form:

$$
\left.
\begin{aligned}
&M\cdot\ddot{X}_{S_t}=\left(F_{DL_1}+F_{DR_1}\right)\cdot\left[\frac{\dot{X}_{S_t}+\frac{1}{2}\sqrt{L^2+K^2}\cdot\dot{\varphi}}{V_o}\right]+\left(F_{DL_2}+F_{DR_2}\right)\cdot\left[\frac{\dot{X}_{S_t}-\frac{1}{2}\sqrt{L^2+K^2}\cdot\dot{\varphi}}{V_o}\right]+\\
&+2\left(k_L+k_R\right)\cdot\left(\varphi-\frac{\dot{X}_{S_t}}{V_o}\right)+P_x-P_y\cdot\varphi,\\
&J_m\cdot\ddot{\varphi}=\frac{1}{2}K\cdot\left(F_{DL_1}+F_{DL_2}-F_{DR_1}-F_{DR_2}\right)+\frac{1}{2}\cdot\frac{\left(k_L+k_R\right)\cdot L^2}{V_o}\cdot\dot{\varphi}-M_C-P_x\cdot a.
\end{aligned}
\right\}\quad(27)
$$

Numerical calculation of the system of equations (27) shows that already during the first 0.6 s from the moment of the created difference of the moments applied to the wheels of the left and the right sides of the wide scan agricultural vehicle in a horizontal plane, its heading angle φ changes to 6 degrees. And also further this value increases exponentially. But at the same time it should be taken into account that after some time of the side turn it is necessary to return the axis of the wide span agricultural vehicle into an opposite direction (exit the turn). That is, in order to return the parallel-displaced agricultural vehicle to the preset trajectory, it is necessary to introduce an additional orientation error in the form of a turn of its skeleton when entering the turn, which should be eliminated at the exit from the turn.

Therefore, manual correction of the trajectory of the movement of a bridge agricultural vehicle with a side turn will only be accompanied by its worse orientation relative to the preset trajectory. Hence it follows that by its onboard turn it is practically impossible to carry out controlled movement with its transverse displacement without an additional turn. It means that the use of only a side turn does not allow to make correction of the trajectory of the movement by a lateral (plane-parallel) displacement of the longitudinal axis of the wide span agricultural vehicle. In case a displacement of the longitudinal axis of the wide span agricultural vehicle from the preset trajectory arises without turning it (for example, as result of its movement on a transverse slope), restoration of the preset trajectory of the movement will be accompanied by turning of its longitudinal axis, and this will require next correction by turning in the opposite direction.

RESULTS AND DISCUSSION

The experimental amplitude-frequency characteristic of oscillations of the heading angle φ (Fig. 2) of the wide span agricultural vehicle showed (Fig. 4) that in the operating range frequencyof 0–2 s^{-1}, the highest value that the amplitude reaches is 2.6 10^{-5} rad N^{-1}, applied, respectively, to the wheels of its left and right sides, which is observed at low frequencies ω. At frequencies of oscillations of the control action of about 2 s^{-1}, the amplitude of oscillations of the heading angle φ of the agricultural tool does not exceed 1.0 10^{-5} rad N^{-1}.

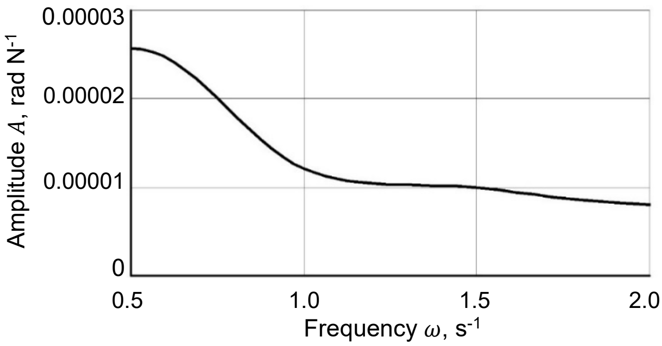

Figure 4. Experimental amplitude-frequency characteristic of fluctuations of the heading angle φ when the agricultural vehicle is working out the difference in tangential forces, applied, respectively, to the wheels of its left and right. Sides.

Oscillations of the transverse displacement of the wide span agricultural vehicle during its movement are also of low frequency (Fig. 5). The main dispersion spectrum of these oscillations is also concentrated in the frequency range 0–2 s^{-1}. The standard of oscillations of this parameter is ± 0.05 m.

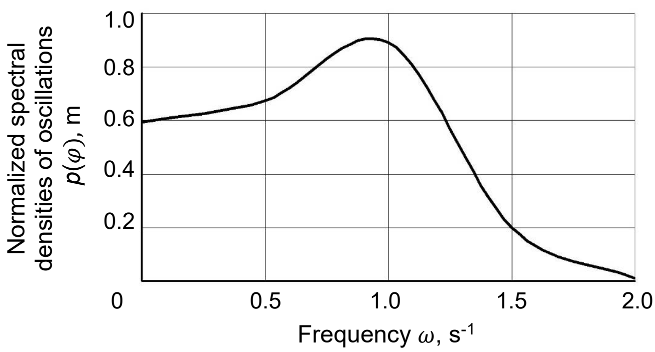

Figure 5. Normalized spectral densities of oscillations of the linear transverse displacement *Xs* of agricultural tool.

The experimental amplitude-frequency characteristic of oscillations of the linear transverse displacement *Xs* of the wide span agricultural vehicle during working out of the control action (Fig. 6) showed that in the operating frequency range of 0–2 s^{-1} oscillations of the input signal its maximum displacement, equal to $1.3 \cdot 10^{-5}$ m N^{-1}, is also observed at low frequencies.

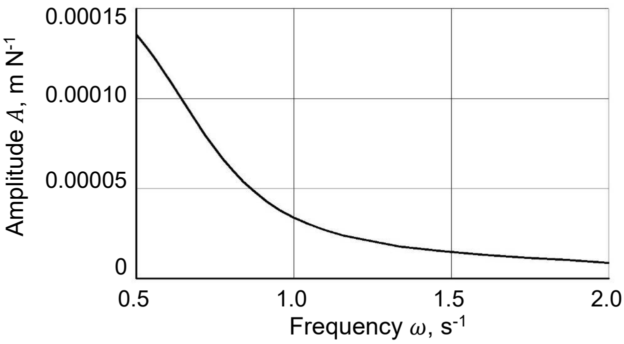

Figure 6. Experimental amplitude-frequency characteristic of oscillations of the linear transverse displacement *Xs* when working out the control action by the wide span agricultural vehicle.

Since the main range of dispersions of oscillations of the tangential forces, applied, respectively, to the wheels of the left and the right sides of the wide span agricultural vehicle, is concentrated exactly at low frequencies, therefore, only the high accuracy of the automated system for driving it along the tracks of a constant tramline, using a laser beam, will allow it to have satisfactory controllability.

CONCLUSIONS

The completed research has proved that the plane-parallel steering of the agricultural wide span vehicle with the use of solely the traction (forced) method of its turning on the treads of the permanent process track requires a significantly smaller difference in the moments applied to the wheels on the right and left sides, as compared to the conventional crawler tractor. The said difference is in proportion to the width of its wheel base at its relatively smaller length.

The smallest power consumption in the process of turning an agricultural wide span vehicle with the use of the traction (forced) method is achieved, when the lateral deformations of the tyres on the wheels of its left and right sides are equal in absolute magnitude, but have opposite directions. To ensure such type of deformations, the traction (forced) turning of the agricultural machine has to be done in such a way that its instantaneous centre of turn is on the axis of transverse symmetry of its chassis.

It has been proved theoretically that already in the first 0.6 s after a difference in the moments applied to the wheels on the right and left sides of the agricultural wide span vehicle has arisen, the heading angle of the vehicle in the horizontal plane changes by 6 deg. Thereafter, this value increases exponentially. In practical terms, that means that it would hardly be possible to perform the controlled motion with a lateral displacement without an additional angular displacement. Hence, the use of solely the traction turning does not provide for the adjustment of the motion trajectory with a lateral (plane-parallel) displacement of the agricultural wide span vehicle's fore-and-aft axis. The oscillations of its transverse displacement in the course of its motion are low-frequency ones. The main variance spectrum of these oscillations is concentrated within the frequency range of 0–2 s^{-1}. The standard oscillations of this parameter are within ± 0.05 m.

Within the operating frequency range of 0–2 s^{-1} of the control action, the maximum lateral displacement of the agricultural wide span vehicle equal to 1.3 10^{-5} m N^{-1} occurs also at lower frequencies.

In view of the fact that the main variance spectrum of the oscillations of the tangential forces applied to the wheels on the left and right sides of the agricultural wide span vehicle is concentrated exactly within a low frequency range, only the high accuracy of the system using a laser beam can ensure its satisfactory steerability.

REFERENCES

Antille, D.L., Chamen, W.C.T., Tullberg, J.N. & Lal, R. 2015. The potential of controlled traffic farming to mitigate greenhouse gas emissions and enhance carbon sequestration in arable land: a critical review. - In *Transactions of the ASABE* **58**(3), 707–731. doi: 10.13031/trans.58.11049

Antille, D.L., Peets, S., Galambošová, J., Botta, G.F., Rataj, V., Macak, M., Tullberg, J.N., Chamen, W.C.T., White, D.R., Misiewicz, P.A., Hargreaves, P.R., Bienvenido, J.F. & Godwin, R.J. 2019. Review: Soil compaction and controlled traffic farming in arable and grass cropping systems. *Agronomy Research* **17**(3), 653–683. doi: 10.15159/AR.19.133

Bakker, T., Van, A.K. & Bontsema, J. 2011. Autonomous navigation using a robot platform in a sugar beet field. *Biosystems Engineering* **109**(4), 357–368.

Bulgakov, V., Adamchuk, V., Kuvachov, V., Arak, M. & Olt, J. 2017. Study into movement of wide span tractors (vehicles) used in controlled traffic farming. *Proceedings of the 28th DAAAM International Symposium "28th DAAAM International Symposium on Intelligent Manufacturing and Automation"* (Ed. B. Katalinic), 08–11th November 2017, Zadar, Croatia, EU. Vienna, Austria: DAAAM International Vienna, pp. 0199−0208, doi: 10.2507/28th.daaam.proceedings.027.

Bulgakov, V., Kuvachov, V. & Olt, J. 2019. Theoretical Study on Power Performance of Agricultural Gantry Systems. *Proceedings of the 30th International DAAAM Symposium 'Intelligent Manufacturing & Automation'*, Zadar, 23.–26.10.2019. Ed. B. Katalinic. Vienna, Austria: DAAAM International, **28**(1), 0167−0175, doi: 10.2507/30th.daaam.proceedings.022.

Bulgakov, V., Melnik, V., Kuvachov, V. & Olt, J. 2018. Theoretical Study on Linkage Unit of Wide Span Tractor. *Proceedings of the 29th International DAAAM Symposium "Intelligent Manufacturing & Automation"* (Ed. B. Katalinic). 2018 October 24–27, Zadar, Croatia. Vienna, Austria: DAAAM International, 29(1):0180−0189. doi: 10.2507/29th. daaam.proceedings.026.

Bulgakov, V., Olt, J., Kuvachov, V., Chernovol, M., Sviren, M. & Kolomiiets, S. 2020a. A study of the interaction between soil and the pneumatic wheels of agricultural gantry systems. *Agraarteadus / Journal of Agricultural Science* **31**(1), 3–9. doi: 10.15159/jas.20.03

Bulgakov, V., Olt, J., Kuvachov, V. & Smolinskyi, S. 2020b. A theoretical and experimental study of the traction properties of agricultural gantry systems. *Agraarteadus / Journal of Agricultural Science* **31**(1), 10–16. doi: 10.15159/jas.20.08

Bulgakov, V., Pascuzzi, S., Ivanovs, S., Kuvachov, V., Postol, Y., Santoro, F. & Melnyk, V. 2021. Study of the steering of a wide span vehicle controlled by a local positioning system. *Journal of Agricultural Engineering* **52**(3), LII:1144. doi: 10.4081/jae.2021.1144

Cerruto, E., Manetto, G., Santoro, F. & Pascuzzi, S. 2018. Operator Dermal Exposure to Pesticides in Tomato and Strawberry Greenhouses from Hand-Held Sprayers. *Sustainability* **10**, 2273, doi:10.3390/su10072273

Chamen, T. 2015. Controlled traffic farming – from world wide research to adoption in Europe and its future prospects. *Acta Technologica Agriculturae Nitra* **3**, 64–73. doi: 10.1515/ata-2015-0014

Chan, K.Y., Oates, A., Swan, A.D., Hayes, R.C., Dear, B.S. & Peoples, M.B. 2006. Agronomic consequences of tractor wheel compaction on a clay soil. *Soil & Tillage Research* **89**, 13–21.

Chebrolu, N., Lottes, P., Schaefer, A., Winterhalter, W., Burgard, W. & Stachniss, C. 2017. Agricultural robot dataset for plant classification, localization and mapping on sugar beet fields. *The International Journal of Robotics Research* **36**, 1045–1052. doi: 10.1177/0278364917720510

Gasso, V., Sørensen, C.A.G. & Oudshoorn, F.W. 2013. Controlled traffic farming: A review of the environmental impacts. *European Journal of Agronomy* **48**, 66–73.

Griepentrog, H.W. 2009. Safe and reliable: Further development of a field robot. *Precision Agriculture* 9, 857–866.

Guerrieri, A.S., Anifantis, A.S., Santoro, F. & Pascuzzi, S. 2019. Study of a Large Square Baler with Innovative Technological Systems that Optimize the Baling Effectiveness. *Agriculture (Switzerland)* **9**(5), 86. doi: 10.3390/agriculture9050086

Hamza, M.A. & Anderson, W.K. 2005. Soil compaction in cropping systems - a review of the nature, causes and possible solutions. *Soil & Tillage Research* **82**, 121–145.

Ji, C. & Zhou, J. 2014. Current situation of navigation technologies for agricultural machinery. *Transactions of the Chinese Society for Agricultural Machinery* **45**(9), 44–54.

Keller, T., Sandin, M.; Colombi, T., Horn, R. & Or, D. 2019. Historical increase in agricultural machinery weights enhanced soil stress levels and adversely affected soil functioning. *Soil & Tillage Research* **194**, 104293. doi: 10.1016/j.still.2019.104293

Kingwell, R. & Fuchsbichler, 2011. A.The whole-farm benefits of controlled traffic farming: An Australian appraisal. *Agricultural Systems* **104**(7), 513–521.

Kuvachov, V., Bulgakov, V., Adamchuk, V., Kaminskiy, V., Melnik, V. & Olt, J. 2021. Experimental research into new harrowing unit based on gantry agricultural implement carrier. *Agronomy Research* **19**(1), 126–135. doi: 10.15159/AR.20.239

Longo, D. & Muscato, G. 2013. Design and simulation of two robotic systems for automatic artichoke harvesting. *Robotics* **2**(4), 217–230. doi: 10.3390/robotics2040217

Nevens, F. & Reheul, D. 2003. The consequences of wheel-induced soil compaction and subsoiling for silage maize on a sandy loam soil in Belgium. *Soil & Tillage Research* **70**, 175–184.

Onal, I. 2012. Controlled traffic farming and wide span tractors. *Journal of Agricultural Machinery Science* **8**(4), 353–364.

Pascuzzi, S., Anifantis, A.S. & Santoro, F. 2020. The concept of a compact profile agricultural tractor suitable for use on specialised tree crops. *Agriculture (Switzerland)* **10**, 123. doi:10.3390/agriculture10040123

Pedersen, H.H., Sorensen, C.G., Oudshoorn, F.W. & McPhee, J.E. 2013. User requirements for a wide span tractor for controlled traffic farming. – International Commission of Agricultural and Biological Engineers, Section V. CIOSTA XXXV Conference 'From Effective to Intelligent Agriculture and Forestry', Billund, Denmark, 3–5 July 2013

Raper, R.L. 2005. Agricultural traffic impacts on soil. *Journal of Terramechanics* **42**, 259–280.

Yang, L., Luo. T., Cheng, X., Li, J. & Song. Y. 2015. Universal autopilot system of tractor based on Raspberry Pi. *Transactions of the Chinese Society of Agricultural Engineering* **31**(21), 109–115.

Zhu, Q., Gao, G. & Niu, W. 2016. The automatic navigation driving system of agricultural machinery. *Modern Agriculture* **5**, 65–67.